ASHRAE POCKET GUIDE

for Air Conditioning, Heating, Ventilation, Refrigeration

I-P

9th Edition

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PREFACE

The ASHRAE Pocket Guide was developed to serve as a ready, offline reference for engineers without easy access to complete ASHRAE Handbook volumes.

This ninth edition has been revised for 2017 to include updates from current editions of the ASHRAE Handbook series as well as from various ASHRAE standards. This edition also features a renewed emphasis in basic design aids: content on more specialized system types has been replaced by an appendix containing climatic design data for selected worldwide locations.

This edition of the ASHRAE Pocket Guide, which was first published in 1987, was compiled by ASHRAE staff editors; previous major contributors were Carl W. MacPhee, Griffith C. Burr, Jr., Harry E. Rountree, and Frederick H. Kohloss.

Throughout this Pocket Guide, original sources of figures and tables are indicated where applicable. For space concerns, a shorthand for ASHRAE publications has been adopted. ASHRAE sources are noted after figure captions or table titles in brackets using the following abbreviations:

Fig	Figure
Tbl	Table
Ch	Chapter
Std	ASHRAE Standard
2017F, 2013F, etc	*ASHRAE Handbook—Fundamentals*
2016S, 2012S, etc.	*ASHRAE Handbook—HVAC Systems and Equipment*
2015A, 2011A, etc.	*ASHRAE Handbook—HVAC Applications*
2014R, 2010R, etc.	*ASHRAE Handbook—Refrigeration*

Complete entries for all references cited in tables and figures are available in the original source publications.

1. AIR HANDLING AND PSYCHROMETRICS

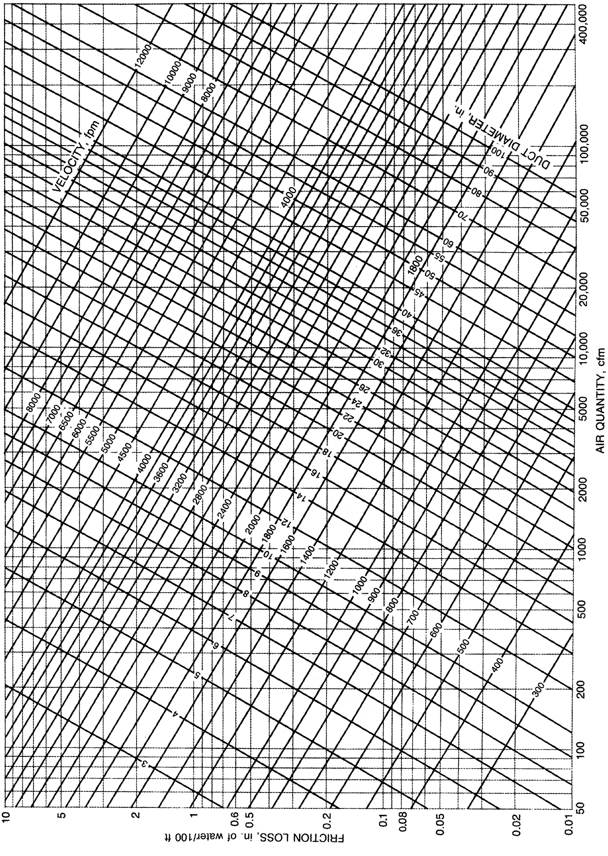


Figure 1.1 Friction Chart for Round Duct (ρ = 0.075 lb_m/ft^3 and ε = 0.0003 ft)
[2017F, Ch 21, Fig 10]

Table 1.1 Velocities vs. Velocity Pressures

Velocity V, fpm	Velocity Pressure P_v, in. H_2O
300	0.01
400	0.01
500	0.02
600	0.02
700	0.03
800	0.04
900	0.05
1000	0.06
1100	0.08
1200	0.09
1300	0.11
1400	0.12
1500	0.14
1600	0.16
1700	0.18
1800	0.20
1900	0.22
2000	0.25
2100	0.27
2200	0.30
2300	0.33
2400	0.36
2500	0.39

$P_v = (V/4005)^2$

Noncircular Ducts

Hydraulic diameter $D_h = 4A/P$, where A = duct area (in.^{2}) and P = perimeter (in.). Ducts having the same hydraulic diameter will have approximately the same fluid resistance at equal velocities.

Fittings

Resistance to flow through fittings can be expressed by fitting loss coefficients C. The friction loss in a fitting in inches of water is CP_v. The more radically the airflow is changed in direction or velocity, the greater the fitting loss coefficient. See *ASHRAE Duct Fitting Database* for a complete list. 90° mitered elbows with vanes will usually have C between 0.11 and 0.33.

Round Flexible Ducts

Nonmetallic flexible ducts fully extended have friction losses approximately three times that of galvanized steel ducts. This rises rapidly for unextended ducts by a correction factor of 4 if 70% extended, 3 if 80% extended, and 2 if 90% extended. For centerline bend radius ratio to diameter of 1 to 4, the approximate loss coefficient is between 0.82 and 0.87.

Table 1.2 Duct Leakage Classification[a]

Duct Type	Predicted Leakage Class C_L	
	Sealed[b,c]	Unsealed[c]
Metal (flexible excluded)		
Round and flat oval	3	30 (6 to 70)
Rectangular		
$\leq$ 2 in. of water (both positive and negative pressures)	12	48 (12 to 110)
> 2 and $\leq$ 10 in. of water (both positive and negative pressures)	6	48 (12 to 110)[c]
Flexible		
Metal, aluminum	8	30 (12 to 54)
Nonmetal	12	30 (4 to 54)
Fibrous glass		
Round	3	na
Rectangular	6	na

[a] The leakage classes listed in this table are averages based on tests conducted by AISI/SMACNA (1972), ASHRAE/SMACNA/TIMA (1985), and Swim and Griggs (1995).
[b] The leakage classes listed in the sealed category are based on the assumptions that for metal ducts, all transverse joints, seams, and openings in the duct wall are sealed at pressures over 3 in. of water, that transverse joints and longitudinal seams are sealed at 2 and 3 in. of water, and that transverse joints are sealed below 2 in. of water. Lower leakage classes are obtained by careful selection of joints and sealing methods.
[c] Leakage classes assigned anticipate about 25 joints per 100 linear feet of duct. For systems with a high fitting to straight duct ratio, greater leakage occurs in both the sealed and unsealed conditions.

Table 1.3 Recommended Ductwork Leakage Class by Duct Type

Duct Type	Leakage Class C_L, cfm/100 ft^2 at 1 in. of water
Metal (flexible excluded)	
Round	3
Flat oval	3
Rectangular	6
Flexible	6
Fibrous glass	
Round	3
Rectangular	6

$$\text{Leakage Class } C_L = Q/\Delta P_S^{0.65} \tag{1.1}$$

where

Q = leakage rate, cfm/100 ft^2 surface area

ΔP_s = static pressure difference, inches of water between inside and outside of duct

Table 1.4 Duct Sealing Requirement Levels

Duct Seal Level	Sealing Requirements[a]
A	All transverse joints, longitudinal seams, and duct wall penetrations
B	All transverse joints and longitudinal seams
C	Transverse joints only

[a] Transverse joints are connections of two duct or fitting elements oriented perpendicular to flow. Longitudinal seams are joints oriented in the direction of airflow. Duct wall penetrations are openings made by screws, non-self-sealing fasteners, pipe, tubing, rods, and wire. Round and flat oval spiral lock seams need not be sealed prior to assembly, but may be coated after assembly to reduce leakage. All other connections are considered transverse joints, including but not limited to spin-ins, taps and other branch connections, access door frames, and duct connections to equipment.

Table 1.5 Duct Sealing Recommendations

Recommended Duct Seal Levels	Duct Type			
	Supply			
Duct Location	≤2 in. of water	>2 in. of water	Exhaust	Return
Outdoors	A	A	A	A
Unconditioned spaces	B	A	B	B
Conditioned spaces (concealed ductwork)	C	B	B	C
Conditioned spaces (exposed ductwork)	A	A	B	B

Table 1.6 Duct Leakage per Unit Length

Unsealed Longitudinal Seam Leakage, Metal Ducts		Leakage, cfm per ft Seam Length at 1 in. Water Pressure	
Type of Duct/Seam		Range	Average
Rectangular	Pittsburgh lock		
	26 gage	0.01 to 0.02	0.0164
	22 gage	0.001 to 0.002	0.0016
	Button punch snaplock		
	26 gage	0.03 to 0.15	0.0795
	22 gage	NA (1 test)	0.0032
Round	Spiral (26 gage)	NA (1 test)	0.015
	Snaplock	0.04 to 0.14	0.11
	Grooved	0.11 to 0.18	0.12

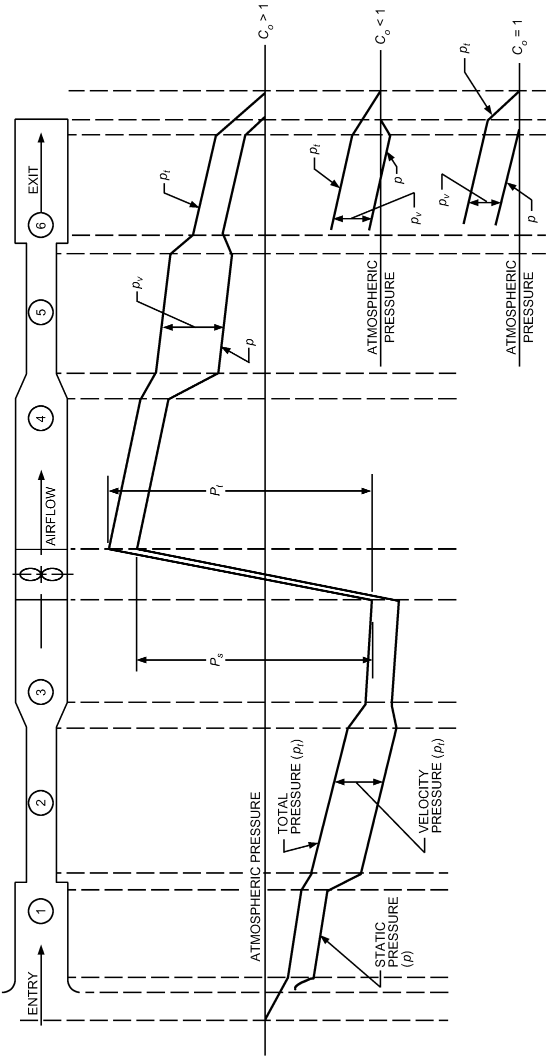


Figure 1.2 At Exit, the Fitting Coefficient C_o Affects ρ_t Loss [2017F, Ch 21, Fig 7]

Table 1.7 Circular Equivalents of Rectangular Duct for Equal Friction and Capacity[a]

Lgth. Adj.[b]	Length of One Side of Rectangular Duct (*a*), in.																
	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5	8.0	9.0	10.0	11.0	12.0	13.0	14.0	15.0	16.0
3.0	3.8	4.0	4.2	4.4	4.6	4.7	4.9	5.1	5.2	5.5	5.7	6.0	6.2	6.4	6.6	6.8	7.0
4.0	4.4	4.6	4.9	5.1	5.3	5.5	5.7	5.8	6.1	6.4	6.7	7.0	7.3	7.6	7.8	8.0	8.3
5.0	4.9	5.2	5.5	5.7	6.0	6.2	6.4	6.7	6.9	7.3	7.6	8.0	8.3	8.6	8.9	9.1	9.4

Lgth. Adj.[b]	Length of One Side of Rectangular Duct (*a*), in.																			Lgth. Adj.[b]
	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	22	24	26	28	30	
6	6.6																				6
8	7.6	8.2	8.7																		8
10	8.4	9.1	9.8	10.4	10.9																10
12	9.1	9.9	10.7	11.3	12.0	12.6	13.1														12
14	9.8	10.8	11.4	12.2	12.9	13.5	14.2	14.7	15.3												14
16	10.4	11.3	12.2	13.0	13.7	14.4	15.1	15.7	16.4	16.9	17.5										16
18	11.0	11.9	12.9	13.7	14.5	15.3	16.0	16.7	17.3	17.9	18.5	19.1	19.7								18
20	11.5	12.6	13.5	14.4	15.2	16.0	16.8	17.5	18.2	18.9	19.5	20.1	20.7	21.3	21.9						20
24	12.4	13.5	14.6	15.6	16.5	17.4	18.3	19.1	19.9	20.6	21.3	22.0	22.7	23.3	34.9	25.1	26.2				24
28	13.2	14.5	15.6	16.7	17.7	18.7	19.6	20.5	21.3	22.1	22.9	23.7	24.4	25.1	25.8	27.1	28.3	29.5	30.6		28
32	14.0	15.3	16.5	17.7	18.8	19.8	20.8	21.8	22.7	23.5	24.4	25.2	26.0	26.7	27.5	28.9	30.2	31.5	32.7	33.9	32
36	14.7	16.1	17.4	18.6	19.8	20.9	21.9	22.9	23.9	24.8	25.7	26.6	27.4	28.2	29.0	30.5	32.0	33.3	34.6	35.9	36
40	15.3	16.8	18.2	19.5	20.7	21.8	22.9	24.0	25.0	26.0	27.0	27.9	28.8	29.6	30.5	32.1	33.6	35.1	36.4	37.8	40
44	15.9	17.5	18.9	20.3	31.5	22.7	23.9	25.0	26.1	27.1	28.1	29.1	30.0	30.9	31.8	33.5	35.1	36.7	38.1	39.5	44
48	16.5	18.1	19.6	21.0	22.3	23.6	24.8	26.0	27.1	28.2	29.2	30.2	31.2	32.2	33.1	34.9	36.6	38.2	39.7	41.2	48
52	17.1	18.7	20.2	21.7	23.1	24.4	25.7	26.9	28.0	29.2	30.3	31.3	32.3	33.3	34.3	36.2	37.9	39.6	41.2	42.8	52
56	17.6	19.3	20.9	22.4	23.8	25.2	26.5	27.7	28.9	30.1	31.2	32.3	33.4	34.4	35.4	37.4	39.2	41.0	42.7	44.3	56
60	18.1	19.8	21.5	23.0	24.5	25.9	27.3	28.6	29.8	31.0	32.2	33.3	34.4	35.5	36.5	38.5	40.4	42.3	44.0	45.7	60
64		20.3	22.0	23.6	25.1	26.6	28.0	29.3	30.6	31.9	33.1	34.3	35.4	36.5	37.6	39.6	41.6	43.5	45.3	47.1	64

Table 1.7 Circular Equivalents of Rectangular Duct for Equal Friction and Capacity[a] *(Continued)*

Lgth. Adj.[b]	Length of One Side of Rectangular Duct (*a*), in. 32	34	36	38	40	42	44	46	48	50	52	56	60	64	68	72	76	80	84	88	Lgth. Adj.[b]
32	35.0																				32
36	37.1	38.2	39.4																		36
40	39.0	40.3	41.5	42.6	43.7																40
44	40.9	42.2	43.5	44.7	45.8	48.1															44
48	42.6	44.0	45.3	46.6	47.9	49.1	50.2	51.4	52.5												48
52	44.3	45.7	47.1	48.4	49.7	51.0	52.2	53.4	54.6	55.7	56.8										52
56	45.8	47.3	48.8	50.2	51.6	52.9	54.2	55.4	56.6	57.8	59.0	61.2									56
60	47.3	48.9	50.4	51.9	53.3	54.7	60.0	57.3	58.6	59.8	61.0	63.4	65.6								60
64	48.7	50.4	51.9	53.5	54.9	56.4	57.8	59.1	60.4	61.7	63.0	65.4	67.7	70.0							64
68	50.1	51.8	53.4	55.0	56.5	58.0	59.4	60.8	62.2	63.6	64.9	67.4	69.8	72.1	74.3						68
72	51.4	53.2	54.8	56.5	58.0	59.6	61.1	62.5	63.9	65.3	66.7	69.3	71.8	74.2	76.5	78.7					72
76	52.7	54.5	56.2	57.9	59.5	61.1	62.6	64.1	65.6	67.0	68.4	71.1	73.7	76.2	78.6	80.9	83.1				76
80	53.9	55.8	57.5	59.3	60.9	62.6	64.1	65.7	67.2	68.7	70.1	72.9	75.4	78.1	80.6	82.9	85.2	87.5			80
84	55.1	57.0	58.8	60.6	62.3	64.0	65.6	67.2	68.7	70.3	71.7	74.6	77.3	80.0	82.5	85.0	87.3	89.6	91.8		84
88	56.3	58.2	60.1	61.9	63.6	65.4	67.0	68.7	70.2	71.8	73.3	76.3	79.1	81.8	84.4	86.9	89.3	91.7	94.0	96.2	88
92	57.4	59.3	61.3	63.1	64.9	66.7	68.4	70.1	71.7	73.3	74.9	77.9	80.8	83.5	86.2	88.8	91.3	93.7	96.1	98.4	92
96	58.4	60.5	62.4	64.3	66.2	68.0	69.7	71.5	73.1	74.8	76.3	79.4	82.4	85.3	88.0	90.7	93.2	95.7	98.1	100.5	96

[a] Table based on $D_e = 1.30\,(ab)^{0.625}/(a + b)^{0.25}$
[b] Length of adjacent side of rectangular duct (*b*), in.

Table 1.8 Equivalent Flat Oval Duct Dimensions* [2017F, Ch 21, Tbl 3]

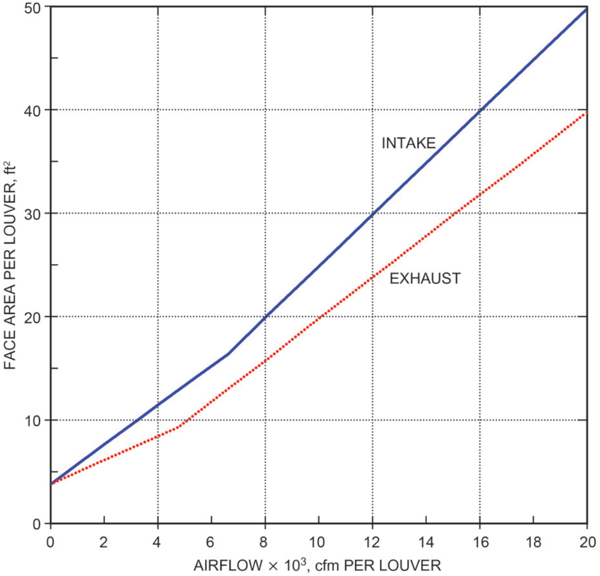
Circular Duct Diameter, in.	Minor Axis *a*, in.											
	3	4	5	6	7	8	9	10	11	12	14	16
	Major Axis *A*, in.											
5	8											
5.5	9	7										
6	11	9										
6.5	12	10	8									
7	15	12	10	8								
7.5	19	13	—	9								
8	22	15	11	—								
8.5		18	13	11	10							
9		20	14	12	—	10						
9.5		21	18	14	12	—						
10			19	15	13	11						
10.5			21	17	15	13	12					
11				19	16	14	—	12				
11.5				20	18	16	14	—				
12				23	20	17	15	13				
12.5				25	21	—	—	15	14			
13				28	23	19	17	16	—	14		
13.5				30	—	21	18	—	16	—		
14				33	—	22	20	18	17	15		
14.5				36	—	24	22	19	—	17		
15				39	—	27	23	21	19	18		
16				45	—	30	—	24	22	20	17	
17				52	—	35	—	27	24	21	19	
18				59	—	39	—	30	—	25	22	19

Circular Duct Diameter, in.	Minor Axis *a*, in.										
	8	9	10	11	12	14	16	18	20	22	24
	Major Axis *A*, in.										
19	46	—	34	—	28	23	21				
20	50	—	38	—	31	27	24	21			
21	58	—	43	—	34	28	25	23			
22	65	—	48	—	37	31	29	26			
23	71	—	52	—	42	34	30	27			
24	77	—	57	—	45	38	33	29	26		
25			63	—	50	41	36	32	29		
26			70	—	56	45	38	34	31		
27			76	—	59	49	41	37	34		
28					65	52	46	40	36		
29					72	58	49	43	39	35	
30					78	61	54	46	40	38	
31					81	67	57	49	44	39	37
32						71	60	53	47	42	40
33						77	66	56	51	46	41
34							69	59	55	47	44
35							76	65	58	50	46
36							79	68	61	53	49
37								71	64	57	52
38								78	67	60	55
40									77	69	62
42										75	68
44										82	74

* Table based on $D_e = 1.30\,(ab)^{0.625}/(a + b)^{0.25}$.

Table 1.9 Typical Design Velocities for HVAC Components

Duct Element	Face Velocity, fpm
Louvers	
Intake	
7000 cfm and greater	400
Less than 7000 cfm	See Figure 1.3
Exhaust	
5000 cfm and greater	500
Less than 5000 cfm	See Figure 1.3
Filters	
Panel filters	
Viscous impingement	200 to 800
Dry-type, extended-surface	
Flat (low efficiency)	Duct velocity
Pleated media (intermediate efficiency)	Up to 750
HEPA	250
Renewable media filters	
Moving-curtain viscous impingement	500
Moving-curtain dry media	200
Electronic air cleaners	
Ionizing type	150 to 350
Heating Coils	
Steam and hot water	500 to 1000
	200 min., 1500 max.
Electric	
Open wire	Refer to mfg. data
Finned tubular	Refer to mfg. data
Dehumidifying Coils	400 to 500
Air Washers	
Spray type	Refer to mfg. data
Cell type	Refer to mfg. data
High-velocity spray type	1200 to 1800



Parameters Used to Establish Figure	Intake Louver	Exhaust Louver
Minimum free area (48 in. square test section), %	45	45
Water penetration, oz/($ft^2 \cdot 0.25$ h)	Negligible (less than 0.01)	N/A
Maximum static pressure drop, in. of water	0.15	0.25

Figure 1.3 Criteria for Louver Sizing [2017F, Ch 21, Fig 19]

Table 1.10 Fan Laws[a,b]

For All Fan Laws: $\eta_{t1} = \eta_{t2}$ and (point of rating)$_1$ = (point of rating)$_2$

No.	Dependent Variables	Independent Variables	
1a	$Q_1 = Q_2$	$\times \left(\frac{D_1}{D_2}\right)^3 \times \frac{N_1}{N_2}$	$\times 1$
1b	Pressure$_1$ = Pressure$_2$[c]	$\times \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{N_1}{N_2}\right)^2$	$\times \frac{\rho_1}{\rho_2}$
1c	$W_1 = W_2$	$\times \left(\frac{D_1}{D_2}\right)^5 \times \left(\frac{N_1}{N_2}\right)^3$	$\times \frac{\rho_1}{\rho_2}$
2a	$Q_1 = Q_2$	$\times \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{\text{Press.}_1}{\text{Press.}_2}\right)^{1/2}$	$\times \left(\frac{\rho_2}{\rho_1}\right)^{1/2}$
2b	$N_1 = N_2$	$\times \left(\frac{D_2}{D_1}\right) \times \left(\frac{\text{Press.}_1}{\text{Press.}_2}\right)^{1/2}$	$\times \left(\frac{\rho_2}{\rho_1}\right)^{1/2}$
2c	$W_1 = W_2$	$\times \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{\text{Press.}_1}{\text{Press.}_2}\right)^{3/2}$	$\times \left(\frac{\rho_2}{\rho_1}\right)^{1/2}$
3a	$N_1 = N_2$	$\times \left(\frac{D_2}{D_1}\right)^3 \times \frac{Q_1}{Q_2}$	$\times 1$
3b	Pressure$_1$ = Pressure$_2$	$\times \left(\frac{D_2}{D_1}\right)^4 \times \left(\frac{Q_1}{Q_2}\right)^2$	$\times \frac{\rho_1}{\rho_2}$
3c	$W_1 = W_2$	$\times \left(\frac{D_2}{D_1}\right)^4 \times \left(\frac{Q_1}{Q_2}\right)^3$	$\times \frac{\rho_1}{\rho_2}$

a. The subscript 1 denotes that the variable is for the fan under consideration.

b. The subscript 2 denotes that the variable is for the tested fan.

c. Fan total pressure P_{tf}, fan velocity pressure P_{vf}, or fan static pressure P_{sf}.

Unless otherwise identified, fan performance data are based on dry air at standard conditions 14.696 psi and 70°F (0.075 lb_m/ft^3). In actual applications, the fan may be required to handle air or gas at some other density. The change in density may be because of temperature, composition of the gas, or altitude. As indicated by the Fan Laws, the fan performance is affected by gas density. With constant size and speed, the horsepower and pressure varies directly as the ratio of gas density to the standard air density.

The application of the Fan Laws for a change in fan speed N for a specific size fan is shown in Figure 1.4. The computed P_t curve is derived from the base curve. For example, point E(N_1 = 650) is computed from point D(N_2= 600) as follows:

At D,

$$Q_2 = 6 \text{ cfm and } P_{tf_2} = 1.13 \text{ in. of water} \tag{1.2}$$

Using Fan Law 1a at Point E

$$Q_1 = 6000(650/600) = 6500 \text{ cfm} \tag{1.3}$$

Using Fan Law 1b

$$P_{tf_1} = 1.13 \times (650/600)^2 = 1.33 \text{ psi} \tag{1.4}$$

The completed P_{tf_1}, N = 650 curve thus may be generated by computing additional points from data on the base curve, such as point G from point F.

$$\text{hp} = \frac{\text{cfm} \times \text{static pressure, in. of water}}{\text{fan efficiency (decimal)} \times 6356} \tag{1.5}$$

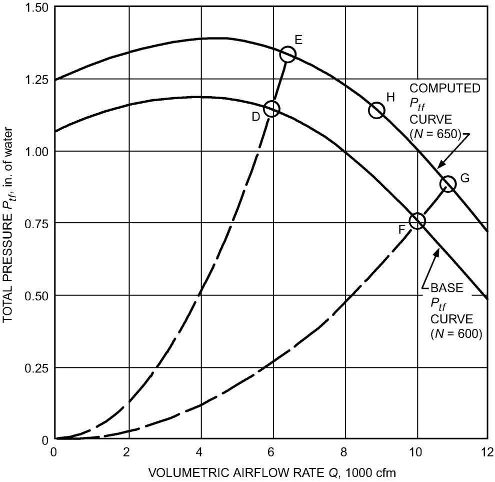


Figure 1.4 Example Calculation of Fan Laws [2016S, Ch 21, Fig 4]

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1]

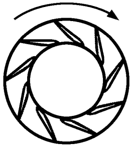
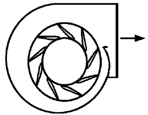
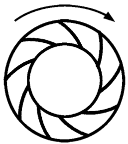
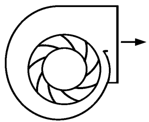
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans	Airfoil		• Blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. • Air leaves impeller at velocity less than tip speed. • For given duty, has highest speed of centrifugal fan designs.		• Scroll design for efficient conversion of velocity pressure to static pressure. • Maximum efficiency requires close clearance and alignment between wheel and inlet.	• Highest efficiency of all centrifugal fan designs and peak efficiencies occur at 50 to 60% of wide-open volume. • Fan has a non-overloading characteristic, which means power reaches maximum near peak efficiency and becomes lower, or self-limiting, toward free delivery.	• General heating, ventilating, and air-conditioning applications. • Usually only applied to large systems, which may be low-, medium-, or high-pressure applications. • Applied to large, clean-air industrial operations for significant energy savings.
	Backward-Inclined Backward-Curved		• Single-thickness blades curved or inclined away from direction of rotation. • Efficient for same reasons as airfoil fan.		• Uses same housing configuration as airfoil design.	• Similar to airfoil fan, except peak efficiency slightly lower. • Curved blades are slightly more efficient than straight blades.	• Same heating, ventilating, and air-conditioning applications as airfoil fan. • Used in some industrial applications where environment may corrode or erode airfoil blade.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

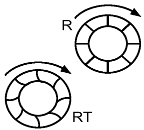
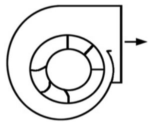
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans (continued)	Radial (R) Radial Tip (Rt)		• Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. • Curve may have a break to left of peak pressure.		• Scroll similar to and often identical to other centrifugal fan designs. • Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.	• Higher pressure characteristics than airfoil and backward-curved fans. • Pressure may drop suddenly at left of peak pressure, but this usually causes no problems. • Power rises continually to free delivery, which is an overloading characteristic. • Curved blades are slightly more efficient than straight blades.	• Primarily for materials handling in industrial plants. Also for some high-pressure industrial requirements. • Rugged wheel is simple to repair in the field. Wheel sometimes coated with special material. • Not common for HVAC applications.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

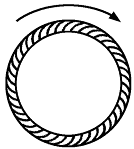
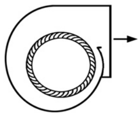
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans (continued)	Forward-Curved		• Flatter pressure curve and lower peak efficiency than the airfoil, backward-curved, and backward-inclined.		• Scroll similar to and often identical to other centrifugal fan designs. • Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.	• Pressure curve less steep than that of backward-curved fans. Curve dips to left of peak pressure. • Highest efficiency occurs at 40 to 50% of wide-open volume. • Operate fan to right of peak pressure. Use caution when selecting left of peak pressure, because instability is possible. • Power rises continually to free delivery which is an overloading characteristic.	• Primarily for low-pressure HVAC applications, such as residential furnaces, central station units, and packaged air conditioners.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

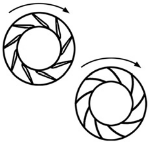
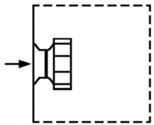
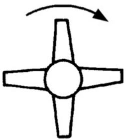
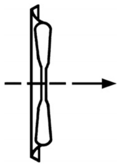
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans (continued)	Plenum/ Plug		• Plenum and plug fans typically use airfoil, backward inclined, or backward curved impellers in a single inlet configuration. Relative benefits of each impeller are the same as those described for scroll housed fans.		• Plenum and plug fans are unique in that they operate with no housing. The equivalent of a housing, or plenum chamber (dashed line), depends on the application. • The components of the drive system for the plug fan are located outside the airstream.	• Plenum and plug fans are similar to comparable housed airfoil/backward-curved fans but are generally less efficient because of inefficient conversion of kinetic energy in discharge airstream. • They are more susceptible to performance degradation caused by poor installation.	• Plenum and plug fans are used in a variety of HVAC applications such as air handlers, especially where direct-drive arrangements are desirable. • Other advantages of these fans are discharge configuration flexibility and potential for smaller-footprint units.
Axial Fans	Propeller		• Low efficiency. • Limited to low-pressure applications. • Usually low-cost impellers have two or more blades of single thickness attached to relatively small hub. • Primary energy transfer by velocity pressure.		• Simple circular ring, orifice plate, or venturi. • Optimum design is close to blade tips and forms smooth airfoil into wheel.	• High flow rate, but very low pressure capabilities. • Maximum efficiency reached near free delivery. • Discharge pattern circular and airstream swirls.	• For low-pressure, high-volume air-moving applications, such as air circulation in a space or ventilation through a wall without ductwork. • Used for makeup air applications.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

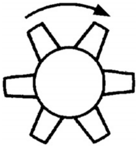
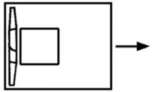
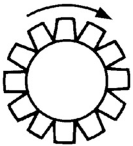
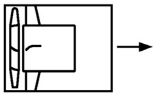
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Axial Fans (continued)	Tubeaxial		• Somewhat more efficient and capable of developing more useful static pressure than propeller fan. • Usually has 4 to 8 blades with airfoil or single- thickness cross section. • Hub is usually less than half the fan tip diameter.		• Cylindrical tube with close clearance to blade tips.	• High flow rate, medium pressure capabilities. • Pressure curve dips to left of peak pressure. Avoid operating fan in this region. • Discharge pattern circular and airstream rotates or swirls.	• Low- and medium-pressure ducted HVAC applications where air distribution downstream is not critical. • Used in some industrial applications, such as drying ovens, paint spray booths, and fume exhausts.
	Vaneaxial		• Good blade design gives medium- to high-pressure capability at good efficiency. • Most efficient have airfoil blades. • Blades may have fixed, adjustable, or controllable pitch. • Hub is usually greater than half fan tip diameter.		• Cylindrical tube with close clearance to blade tips. • Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.	• High-pressure characteristics with medium-volume flow capabilities. • Pressure curve dips to left of peak pressure. Avoid operating fan in this region. • Guide vanes correct circular motion imparted by impeller and improve pressure characteristics and efficiency of fan.	• General HVAC systems in low-, medium-, and high-pressure applications where straight-through flow and compact installation are required. • Has good downstream air distribution. • Used in industrial applications in place of tubeaxial fans. • More compact than centrifugal fans for same duty.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

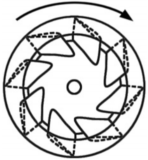
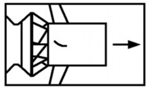
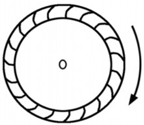
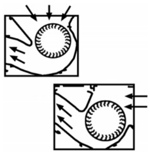
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Mixed-Flow	Mixed-Flow		• Combination of axial and centrifugal characteristics. Ideally suited in applications in which the air has to flow in or out axially. Higher pressure characteristic than axial fans.		• The majority of mixed-flow fans are in a tubular housing and include outlet turning vanes. • Can operate without housing or in a pipe and duct.	• Characteristic pressure curve between axial fans and centrifugal fans. Higher pressure than axial fans and higher volume flow than centrifugal fans.	• Similar HVAC applications to centrifugal fans or in applications where an axial fan cannot generate sufficient pressure rise.
Cross-Flow	Cross-Flow (Tangential)		• Impeller with forward-curved blades. During rotation the flow of air passes through part of the rotor blades into the rotor. This creates an area of turbulence which, working with the guide system, deflects the airflow through another section of the rotor into the discharge duct of the fan casing. Lowest efficiency of any type of fan.		• Special designed housing for 90°or straight through airflow.	• Similar to forward-curved fans. Power rises continually to free delivery, which is an overloading characteristic. • Unlike all other fans, performance curves include motor characteristics. • Lowest efficiency of any fan type.	• Low-pressure HVAC systems such as fan heaters, fireplace inserts, electronic cooling, and air curtains.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

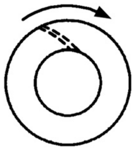
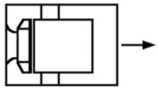
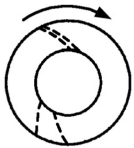
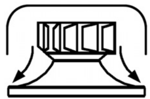
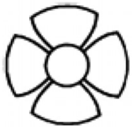
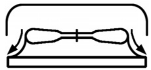
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Other Designs	Tubular Centrifugal		• Performance similar to backward-curved fan except capacity and pressure are lower. • Lower efficiency than backward-curved fan. • Performance curve may have a dip to the left of peak pressure.		• Cylindrical tube similar to vaneaxial fan, except clearance to wheel is not as close. • Air discharges radially from wheel and turns 90° to flow through guide vanes.	• Performance similar to backward-curved fan, except capacity and pressure are lower. • Lower efficiency than backward-curved fan because air turns 90°. • Performance curve of some designs is similar to axial flow fan and dips to left of peak pressure.	• Primarily for low-pressure, return air systems in HVAC applications. • Has straight-through flow.
	Power Roof Ventilators — Centrifugal		• Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. • Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. • Centrifugal units are slightly quieter than axial units.		• Normal housing not used, because air discharges from impeller in full circle. • Usually does not include configuration to recover velocity pressure component.	• Usually operated without ductwork; therefore, operates at very low pressure and high volume.	• Centrifugal units are somewhat quieter than axial flow units. • Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. • Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.

Table 1.11 Types of Fans [2016S, Ch 21, Tbl 1] ***(Continued)***

	Type		Impeller Design		Housing Design		Performance Characteristics	Applications
Other Designs (continued)	Power Roof Ventilators (continued)	Axial		• Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. • Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. • Hood protects fan from weather and acts as safety guard.		• Essentially a propeller fan mounted in a supporting structure. • Air discharges from annular space at bottom of weather hood.	• Usually operated without ductwork; therefore, operates at very low pressure and high volume.	• Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. • Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.

Fan System Effect

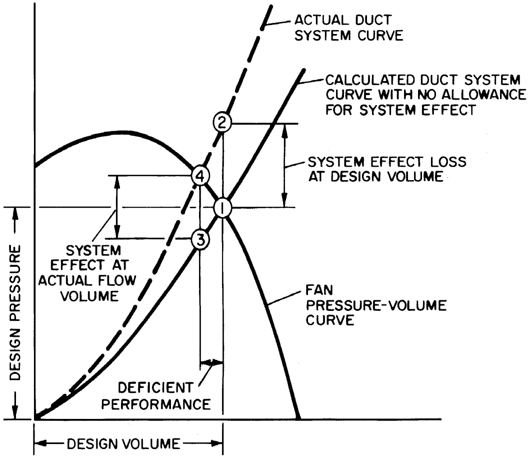


Figure 1.5 Deficient Fan/System Performance

Figure 1.5 illustrates deficient fan/system performance. System pressure losses have been determined accurately, and a fan has been selected for operation at point 1. However, no allowance has been made for the effect of system connections to the fan on fan performance. To compensate, a fan system effect must be added to the calculated system pressure losses to determine the actual system curve. The point of intersection between the fan performance curve and the actual system curve is point 4. The actual flow volume is, therefore, deficient by the difference from 1 to 4. To achieve design flow volume, a fan system effect pressure loss equal to the pressure difference between points 1 and 2 should be added to the calculated system pressure losses, and the fan should be selected to operate at point 2.

For rated performance, air must enter a fan uniformly over the inlet area in an axial direction without prerotation.

Fans within plenums and cabinets or next to walls should be located so that air may flow unobstructed into the inlets.

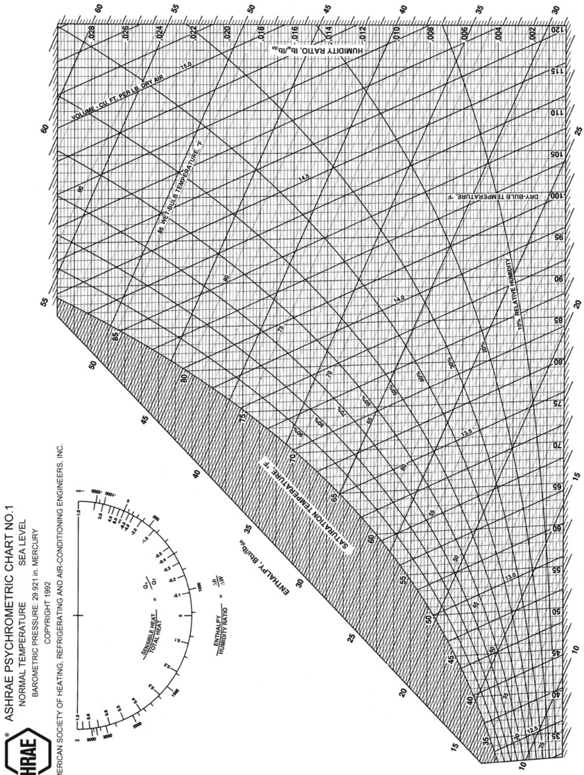


Figure 1.6 Psychrometric Chart for Normal Temperature, Sea Level [2017F, Ch 1, Fig 1]

Air-Conditioning Processes

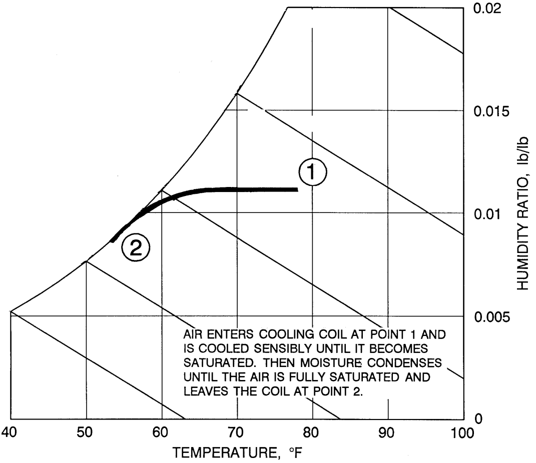


Figure 1.7 Direct Expansion or Chilled Water Cooling and Dehumidification

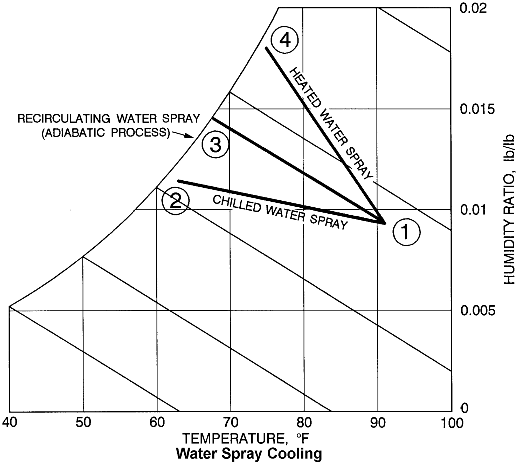


Figure 1.8 Direct Expansion or Chilled Water Cooling and Dehumidification

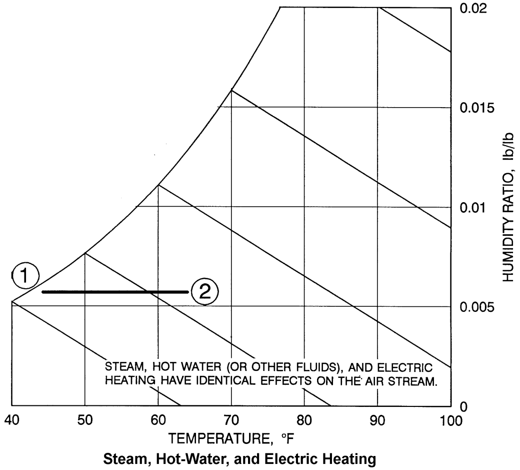


Figure 1.9 Direct Expansion or Chilled Water Cooling and Dehumidification

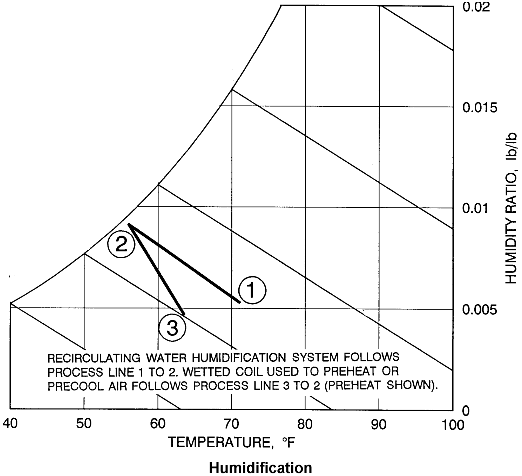
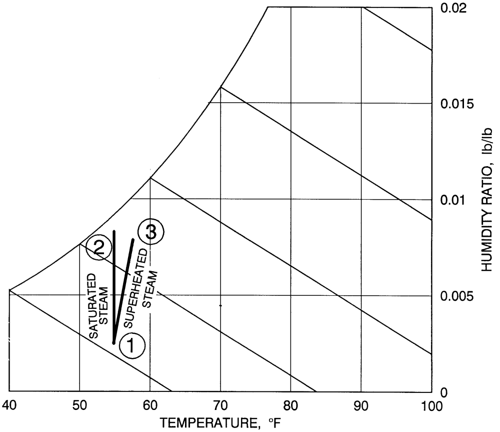
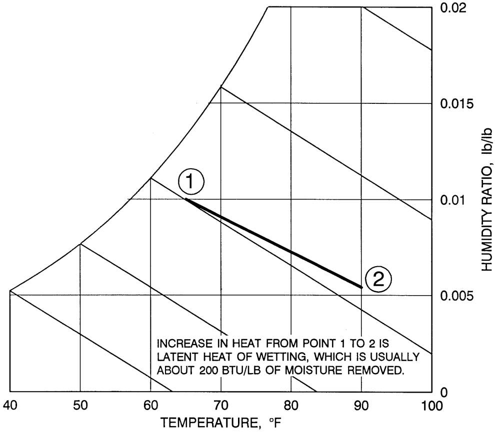


Figure 1.10 Direct Expansion or Chilled Water Cooling and Dehumidification



Steam Humidifier

Figure 1.11 Direct Expansion or Chilled Water Cooling and Dehumidification



Chemical Dehumidifier

Figure 1.12 Direct Expansion or Chilled Water Cooling and Dehumidification

Table 1.12 Specific Enthalpy of Moist Air at Standard Atmospheric Pressure, 14.696 psia
[2017F, Ch 1, Tbl 2, Abridged]

Temp., °F	Specific Enthalpy, Btu/lb$_{da}$	Temp., °F	Specific Enthalpy, Btu/lb$_{da}$
–80	–19.213	79	42.634
–70	–16.804	80	43.701
–60	–14.390	81	44.794
–50	–11.966	82	45.914
–40	–9.524	83	47.062
–30	–7.052	84	48.239
–20	–4.527	85	49.445
–15	–3.234	86	50.682
–10	–1.915	87	51.950
–5	–0.561	88	53.250
0	0.835	89	54.584
5	2.286	90	55.952
10	3.803	91	57.355
15	5.403	92	58.795
20	7.106	93	60.272
25	8.934	94	61.787
30	10.916	95	63.343
35	13.009	96	64.939
40	15.232	97	66.578
45	17.653	98	68.260
50	20.306	99	69.987
55	23.229	100	71.761
60	26.467	110	92.386
65	30.070	120	119.615
70	34.097	130	156.077
71	34.959	140	205.828
72	35.841	150	275.493
73	36.744	160	376.736
74	37.668	170	532.269
75	38.615	180	793.142
76	39.584	190	1303.297
77	40.576	200	2688.145
78	41.593		

Table 1.13 Standard Atmospheric Data for Altitudes to 30,000 ft
[2017F, Ch 1, Tbl 1]

Altitude, ft	Temperature, °F	Pressure, psia
–1000	62.6	15.236
–500	60.8	14.966
0	59.0	14.696
500	57.2	14.430
1,000	55.4	14.175
2,000	51.9	13.664
3,000	48.3	13.173
4,000	44.7	12.682
5,000	41.2	12.230
6,000	37.6	11.778
7,000	34.0	11.341
8,000	30.5	10.914
9,000	26.9	10.506
10,000	23.4	10.108
15,000	5.5	8.296
20,000	–12.3	6.758
30,000	–47.8	4.371

Source: Adapted from NASA (1976).

Table 1.14 Moisture and Air Relationships*

ASHRAE has adopted pounds of moisture per pound of dry air as standard nomenclature. Relations of other units are expressed below at various dew-point temperatures.

Equiv. Dew Pt. °F	Lb H_2O/ lb dry air	Parts per million	Grains/ lb dry air[a]	Percent Moisture%[b]
–100	0.000001	1	0.0007	—
–80	0.000005	5	0.0035	—
–60	0.000002	21	0.148	0.13
–40	0.000008	79	0.555	0.5
–20	0.00026	263	1.84	1.7
–10	0.00046	461	3.22	2.9
0	0.0008	787	5.51	5.0
10	0.0013	1315	9.20	8.3
20	0.0022	2152	15.1	13.6
30	0.0032	3154	24.2	21.8
40	0.0052	5213	36.5	33.0
50	0.0077	7658	53.6	48.4
60	0.0111	11080	77.6	70.2
70	0.0158	15820	110.7	100.0
80	0.0223	22330	156.3	—
90	0.0312	31180	218.3	—
100	0.0432	43190	302.3	—

a. 7000 grains = 1 lb
b. Compared to 70°F saturated.
* *NUMBERS*, 1985, Altadena, CA, by Bill Holladay and Cy Otterholm.

Space Air Diffusion

Room air diffusion methods can be classified as one of the following:

- **Fully mixed systems** produce little or no thermal stratification of air within the space. Overhead air distribution is an example of this type of system.
- **Fully (thermally) stratified systems** produce little or no mixing of air within the occupied space. Thermal displacement ventilation is an example of this type of system.
- **Partially mixed systems** provide some mixing within the occupied and/or process space while creating stratified conditions in the volume above. Most underfloor air distribution and task/ambient conditioning designs are examples of this type of system.

Air distribution systems, such as thermal displacement ventilation (TDV) and underfloor air distribution (UFAD), that deliver air in cooling mode at or near floor level and return air at or near ceiling level produce varying amounts of room air stratification. For floor-level supply, thermal plumes that develop over heat sources in the room play a major role in driving overall floor-to-ceiling air motion. The amount of stratification in the room is primarily determined by the balance between total room airflow and heat load. In practice, the actual temperature and concentration profile depends on the combined effects of various factors, but is largely driven by the characteristics of the room supply airflow and heat load configuration.

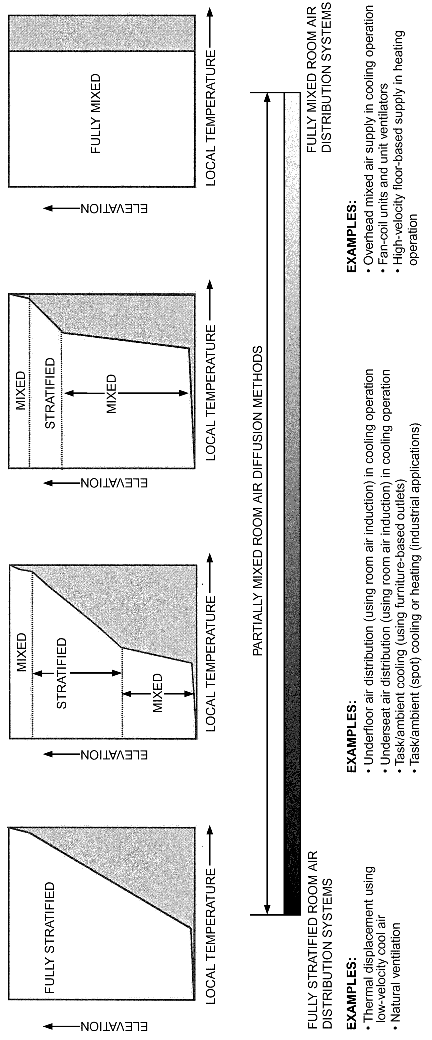


Figure 1.13 Classification of Air Diffusion Methods [2017F, Ch 20, Fig 1]

Principles of Jet Behavior

Air Jet Fundamentals

Air supplied to rooms through various types of outlets can be distributed by turbulent air jets (mixed and partially mixed systems) or in a low-velocity, unidirectional manner (stratified systems).

If an air jet is not obstructed or affected by walls, ceiling, or other surfaces, it is considered a **free jet**. When outlet area is small compared to the dimensions of the space normal to the jet, the jet may be considered free as long as

$$X \le 1.5\sqrt{A_R} \quad (1.6)$$

where

X	=	distance from face of outlet, ft
A_R	=	cross-sectional area of confined space normal to jet, ft^2

Characteristics of the air jet in a room might be influenced by reverse flows created by the same jet entraining ambient air. If the supply air temperature is equal to the ambient room air temperature, the air jet is called an **isothermal jet**. A jet with an initial temperature different from the ambient air temperature is called a **nonisothermal jet**. The air temperature differential between supplied and ambient room air generates thermal forces (buoyancy) in jets, affecting the jet's (1) trajectory, (2) location at which it attaches to and separates from the ceiling/floor, and (3) throw. The significance of these effects depends on the ratio between the thermal buoyancy of the air and jet momentum.

Jet Expansion Zones. The full length of an air jet, in terms of the maximum or centerline velocity and temperature differential at the cross section, can be divided into four zones:

- **Zone 1** is a short core zone extending from the outlet face, in which the maximum velocity and temperature of the airstream remains practically unchanged.
- **Zone 2** is a transition zone, with its length determined by the type of outlet, aspect ratio of the outlet, initial airflow turbulence, etc.
- **Zone 3** is a zone of jet degradation, where centerline air velocity and temperature decrease rapidly. Turbulent flow is fully established and may be 25 to 100 equivalent air outlet diameters (i.e., widths of slot air diffusers) long.
- **Zone 4** is of major engineering importance because, in most cases, the jet enters the occupied area in this zone. Distance to this zone and its length depend on the velocities and turbulence characteristics of ambient air. In a few diameters or widths, air velocity becomes less than 50 fpm.

Centerline Velocities in Zones 1 and 2. In zone 1, the ratio V_x/V_o is constant and ranges between 1.0 and 1.2, equal to the ratio of the center velocity of the jet at the start of expansion to the average velocity. The ratio V_x/V_o varies from approximately 1.0 for rounded entrance nozzles to about 1.2 for straight pipe discharges; it has much higher values for diverging discharge outlets.

Experimental evidence indicates that, in zone 2,

$$\frac{V_x}{V_o} = \sqrt{\frac{K_c H_o}{X}} \quad (1.7)$$

where

V_x	=	centerline velocity at distance X from outlet, fpm
V_o	=	$V_c/C_d R_{fa}$ = average initial velocity at discharge from open-ended duct or across contracted stream at vena contracta of orifice or multiple-opening outlet, fpm
V_c	=	nominal velocity of discharge based on core area, fpm
C_d	=	discharge coefficient (usually between 0.65 and 0.90)
R_{fa}	=	ratio of free area to gross (core) area
H_o	=	width of jet at outlet or at vena contracta, ft
K_c	=	centerline velocity constant, depending on outlet type and discharge pattern (see Table 1.15)
X	≥	$(1/K_c H_o)^{1/2}$ = distance from outlet to measurement of centerline velocity V_x, ft

Table 1.15 Generic Values for Centerline Velocity Constant K_c for Commercial Supply Outlets for Fully and Partially Mixed Systems, Except UFAD [2017F, Ch 20, Tbl 1]

Outlet Type	Discharge Pattern	A_o	K_c
High sidewall grilles	0° deflection[a]	Free	5.7
	Wide deflection	Free	4.2
High sidewall linear	Core less than 4 in. high[b]	Free	4.4
	Core more than 4 in. high	Free	5.0
Low sidewall	Up and on wall, no spread	Free	4.5
	Wide spread[b]	Free	3.0
Baseboard	Up and on wall, no spread	Core	4.0
	Wide spread	Core	2.0
Floor grille	No spread[b]	Free	4.7
	Wide spread	Free	1.6
Ceiling	360° horizontal[c]	Neck	1.1
	Four-way; little spread	Neck	3.8
Ceiling linear slot	Horizontal/vertical along surface[b]	Free	5.5
	Horizontal/vertical free jet	Free	3.9
	Free jet (air curtain units)	Free	6.0

[a]Free area is about 80% of core area.
[b]Free area is about 50% of core area.
[c]Cone free area is greater than duct area.

Centerline Velocity in Zone 3. In zone 3, centerline velocities of radial and axial isothermal jets can be determined accurately from Equation 1.8:

$$V_x = \frac{K_c V_o \sqrt{A_o}}{X} = \frac{K_c Q_o}{X\sqrt{A_o}} \tag{1.8}$$

where

K_c = centerline velocity constant
A_o = free area, core area, or neck area as shown in Table 1.14 (obtained from outlet manufacturer), ft^2
A_c = measured gross (core) area of outlet, ft^2
Q_o = discharge from outlet, cfm

The effective area, according to ASHRAE Standard 70, can be used in place of A_o in Equation 1.8 with the appropriate value of K_c.

Throw. Equation 1.8 can be transposed to determine the throw X of an outlet if the discharge volume and the centerline velocity are known:

$$X = \frac{K_c Q_o}{V_x \sqrt{A_o}} \tag{1.9}$$

Comparison of Free Jet to Attached Jet

Most manufacturers' throw data obtained in accordance with ASHRAE Standard 70 assume the discharge is attached to a surface. An attached jet induces air along the exposed side of the jet, whereas a free jet can induce air on all its surfaces. Because a free jet's induction rate is larger compared to that of an attached jet, a free jet's throw distance will be shorter. To calculate the throw distance X for a noncircular free jet from catalog data for an attached jet, the following estimate can be used.

$$X_{free} = X_{attached} \times 0.707 \tag{1.10}$$

Circular free jets generally have longer throws compared to noncircular jets.

Jets from ceiling diffusers initially tend to attach to the ceiling surface, because of the force exerted by the Coanda effect. However, cold air jets will detach from the ceiling if the air-stream's buoyancy forces are greater than the inertia of the moving air stream.

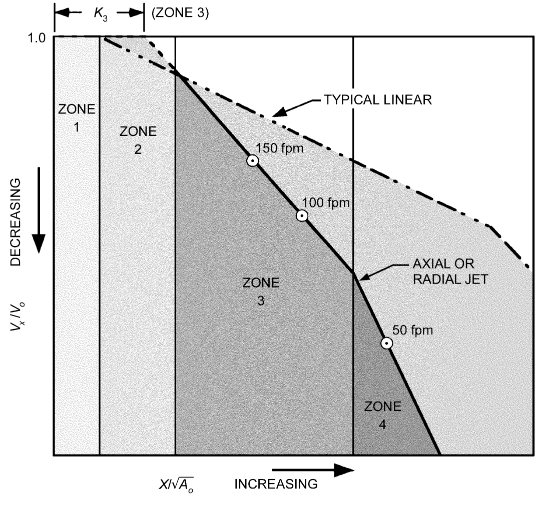
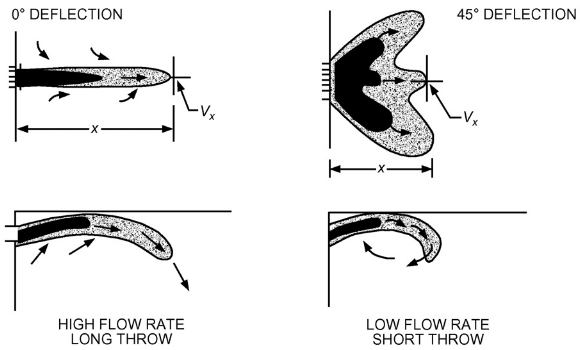
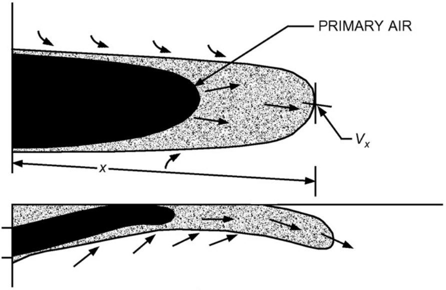


Figure 1.14 Chart for Determining Centerline Velocities of Axial and Radial Jets
[2017F, Ch 20, Fig 11]



A. HIGH SIDEWALL GRILLES



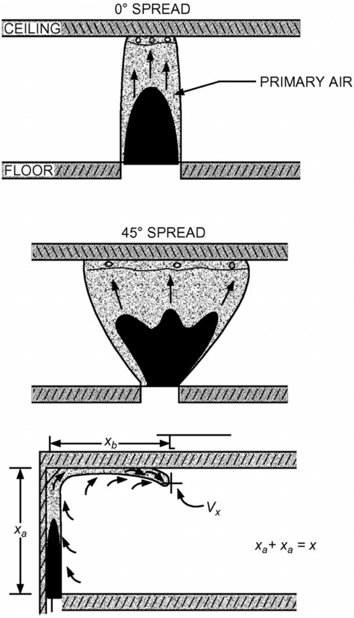
Note: Throw values based on 4 ft long, isothermal active section. Correction multipliers for other lengths are as follows:

Active length, ft	2	4	8	10
Throw correction	0.72	1	1.5	1.7

B. HIGH SIDEWALL LINEAR

Note: Airflow patterns shown with darker shading indicate primary air patterns for terminal velocities above 150 fpm.

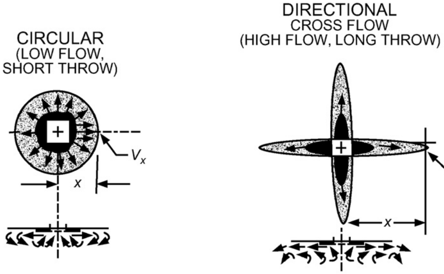
Figure 1.15 Airflow Patterns of Different Diffusers



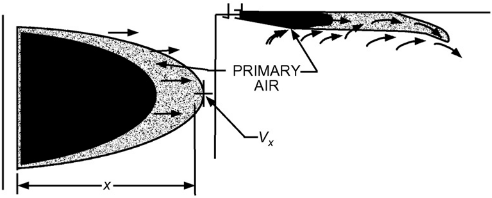
C. LOW SIDEWALL BASEBOARD OR FLOOR WITH VERTICAL DISCHARGE

Note: Airflow patterns shown with darker shading indicate primary air patterns for terminal velocities above 150 fpm.

Figure 1.15 Airflow Patterns of Different Diffusers
(Continued)



D. CEILING DIFFUSERS



Note: Throw values based on 4 ft long, isothermal active section. Correction multipliers for other lengths are as follows:

Active length, ft	2	4	8	10
Throw correction	0.72	1	1.5	1.7

E. CEILING LINEAR

Note: Airflow patterns shown with darker shading indicate primary air patterns for terminal velocities above 150 fpm.

Figure 1.15 Airflow Patterns of Different Diffusers
(Continued)

System Design

Fully Mixed Air Distribution

In mixed air systems, high-velocity supply jets from air outlets maintain comfort by mixing room air with supply air. This air mixing, heat transfer, and resultant velocity reduction should occur outside the occupied zone. Occupant comfort is maintained not directly by motion of air from outlets, but from secondary air motion from mixing in the unoccupied zone. Comfort is maximized when uniform temperature distribution and room air velocities of less than 50 fpm are maintained in the occupied zone.

Maintaining velocities less than 50 fpm in the occupied zone is often overlooked by designers, but is critical to maintaining comfort. The outlet's selection, location, supply air volume, discharge velocity, and air temperature differential determine the resulting air motion in the occupied zone.

Principles of Operation

Mixed systems generally provide comfort by entraining room air into discharge jets located outside occupied zones, mixing supply and room air. Ideally, these systems generate low-velocity air motion (less than 50 fpm) throughout the occupied zone to provide uniform temperature gradients and velocities. Proper selection of an air outlet is critical for proper air distribution; improper selection can result in room air stagnation, unacceptable temperature gradients, and unacceptable velocities in the occupied zone that may lead to occupant discomfort.

The location of a discharge jet relative to surrounding surfaces is important. Discharge jets attach to parallel surfaces, given sufficient velocity and proximity. When a jet is attached, the throw increases by about 40% over a jet discharged in an open area. This difference is important when selecting an air outlet. For detailed discussion of the surface effect on discharge jets, see Chapter 20 of the 2017 *ASHRAE Handbook—Fundamentals*.

Mixed air systems typically use either ceiling or sidewall outlets discharging air horizontally, or floor- or sill-mounted outlets discharging air vertically. They are the most common method of air distribution in North America.

Horizontal Discharge Cooling with Ceiling-Mounted Outlets

Ceiling-mounted outlets typically use the surface effect to transport supply air in the unoccupied zone. The supply air projects across the ceiling and, with sufficient velocity, can continue down wall surfaces and across floors. In this application, supply air should remain outside the occupied zone until it is adequately mixed and tempered with room air.

Overhead outlets may also be installed on exposed ducts, in which case the surface effect does not apply. Typically, if the outlet is mounted 1 ft or more below a ceiling surface, discharge air will not attach to the surface. The unattached supply air has a shorter throw and can project downward, resulting in high air velocities in the occupied zone. Some outlets are designed for use in exposed duct applications. Typical outlet performance data presented by manufacturers are for outlets with surface effect; consult manufacturers for information on exposed duct applications.

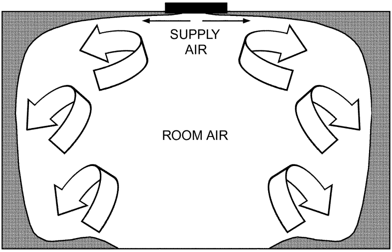


Figure 1.16 Air Supplied at Ceiling Induces Room Air into Supply Jet [2015A, Ch 57, Fig 3]

Vertical-Discharge Cooling or Heating with Ceiling-Mounted Outlets

Vertically projected outlets are typically selected for high-ceiling applications that require forcing supply air down to the occupied zone. It is important to keep cooling supply air velocity below 50 fpm in the occupied zone. For heating, supply air should reach the floor.

There are outlets specifically designed for vertical projection and it is important to review the manufacturer's performance data notes to understand how to apply catalog data. Throws for heating and cooling differ and also vary depending on the difference between supply and room air temperatures.

Cooling with Sidewall Outlets

Sidewall outlets are usually selected when access to the ceiling plenum is restricted. Sidewall outlets within 1 ft of a ceiling and set for horizontal or a slightly upward projection the sidewall outlet provide a discharge pattern that attaches to the ceiling and travels in the unoccupied zone. This pattern entrains air from the occupied zone to provide mixing.

In some applications, the outlet must be located 2 to 4 ft below the ceiling. When set for horizontal projection, the discharge at some distance from the outlet may drop into the occupied zone. Most devices used for sidewall application can be adjusted to project the air pattern upwards toward the ceiling. This allows the discharge air to attach to the ceiling, increasing throw distance and minimizing drop. This application provides occupant comfort by inducing air from the occupied zone into the supply air.

Some outlets may be more than 4 ft below the ceiling (e.g., in high-ceiling applications, the outlet may be located closer to the occupied zone to minimize the volume of the conditioned space). Most devices used for sidewall applications can be adjusted to project the air pattern upward or downward, which allows the device's throw distance to be adjusted to maximize performance.

When selecting sidewall outlets, it is important to understand the manufacturer's data. Most manufacturers offer data for outlets tested with surface effect, so they only apply if the device is set to direct supply air toward the ceiling. When the device is 4 ft or more below a ceiling, or supply air is directed horizontally or downward, the actual throw distance of the device is typically shorter. Many sidewall outlets can be adjusted to change the spread of supply air, which can significantly change throw distance. Manufacturers usually publish throw distances based on specific spread angles.

Cooling with Floor-Mounted Air Outlets

Although not typically selected for nonresidential buildings, floor-mounted outlets can be used for mixed system cooling applications. In this configuration, room air from the occupied zone is induced into the supply air, providing mixing. When cooling, the device should be selected to discharge vertically along windows, walls, or other vertical surfaces. Typical nonresidential applications include lobbies, long corridors, and houses of worship.

It is important to select a device that is specially designed for floor applications. It must be able to withstand both the required dynamic and static structural loads (e.g., people walking on them, loaded carts rolling across them). Also, many manufacturers offer devices designed to reduce the possibility of objects falling into the device. It is strongly recommended that obstructions are not located above these in-floor air terminals, to avoid restricting their air jets.

Long floor-mounted grilles generally have both functioning and nonfunctioning segments. When selecting air outlets for floor mounting, it is important to note that the throw distance and sound generated depend on the length of the active section. Most manufacturers' catalog data include correction factors for length's effects on both throw and sound. These corrections can be significant and should be evaluated. Understanding manufacturers' performance data and corresponding notes is imperative.

Cooling with Sill-Mounted Air Outlets

Sill-mounted air outlets are commonly used in applications that include unit ventilators and fan coil units. The outlet should be selected to discharge vertically along windows, walls, or other vertical surfaces, and project supply air above the occupied zone.

As with floor-mounted grilles, when selecting and locating sill grilles, consider selecting devices designed to reduce the nuisance of objects falling inside them. It is also recommended that sills be designed to prevent them from being used as shelves.

Heating and Cooling with Perimeter Ceiling-Mounted Outlets

When air outlets are used at the perimeter with vertical projection for heating and/or cooling, they should be located near the perimeter surface, and selected so that the published 150 fpm isothermal throw extends at least halfway down the surface or 5 ft above the floor, whichever is lower. In this manner, during heating, warm air mixes with the cool downdraft on the perimeter surface, to reduce or even eliminate drafts in the occupied space.

If a ceiling-mounted air outlet is located away from the perimeter wall, in cooling mode, the high-velocity cool air reduces or overcomes the thermal updrafts on the perimeter surface. To accomplish this, the outlet should be selected for horizontal discharge toward the wall. Outlet selection should be such that isothermal throw to the terminal velocity of 150 fpm should include the distance from the outlet to the perimeter surface. For heating, the supply air temperature should not exceed 15°F above the room air temperature.

Space Temperature Gradients and Airflow Rates

A fully mixed system creates homogeneous thermal conditions throughout the space. As such, thermal gradients should not be expected to exist in the occupied zone. Improper selection, sizing, or placement may prevent full mixing and can result in stagnant areas, or having high-velocity air entering the occupied zone.

Supply airflow requirements to satisfy space sensible heat gains or losses are inversely proportional to the temperature difference between supply and return air. Equation 1.11 can be used to calculate space airflow requirements (at standard conditions):

$$Q = \frac{q_s}{1.08(t_r - t_s)} \tag{1.11}$$

where

Q	=	required supply airflow rate to meet sensible load, cfm
q_s	=	net sensible heat gain in the space, Btu/h
t_r	=	return or exhaust air temperature, °F
t_s	=	supply air temperature, °F

For fully mixed systems with conventional ceiling heights, the return (or exhaust) and room air temperatures are the same; for example, a room with a set-point temperature of 75°F has, on average, a 75°F return or exhaust air temperature.

The object of air diffusion in warm-air heating, ventilating, and air-conditioning systems is to create the proper combination of temperature, humidity, and air motion in the occupied zone of the conditioned room—from the floor to 6 ft above floor level.

Discomfort can arise due to any of the following: excessive air motion (draft), excessive room air temperature variations (horizontal, vertical, or both), failure to deliver or distribute air according to the load requirements at different locations, overly rapid fluctuation of room temperature.

Air Diffusion Performance Index (ADPI)

ADPI is the percentage of locations where measurements are taken that meet these specifications for effective draft temperature and air velocity. If the ADPI is maximum (approaching 100%), the most desirable conditions are achieved. ADPI should be used only for cooling mode in sedentary occupancies. Where air does not strike a wall but collides with air from a neighboring diffuser, L is one-half the distance between the diffusers plus the distance the mixed air drops to the occupied zone.

Table 1.16 Characteristic Room Length for Several Diffusers

[2015A, Ch 57, Tbl 5]

Diffuser Type	Characteristic Length *L*
High sidewall grille	Distance to wall perpendicular to jet
Circular ceiling pattern diffuser	Distance to closest wall or intersecting air jet
Sill grille	Length of room in direction of jet flow
Ceiling slot diffuser	Distance to wall or midplane between outlets
Light troffer diffusers	Distance to midplane between outlets plus distance from ceiling to top of occupied zone
Perferated, louvered ceiling diffusers	Distance to wall or midplane between outlets

Table 1.17 Air Diffusion Performance Index (ADPI) Selection Guide

[2015A, Ch 57, Tbl 6]

Terminal Device	Room Load, Btu/h·ft^2	X_{50}/L for Maximum ADPI	Maximum ADPI	For ADPI Greater than	Range of X_{50}/L
High sidewall grilles	80	1.8	68	—	—
	60	1.8	72	70	1.5 to 2.2
	40	1.6	78	70	1.2 to 2.3
	20	1.5	85	80	1.0 to 1.9
	<10	1.4	90	80	0.7 to 2.1
Circular ceiling diffusers	80	0.8	76	70	0.7 to 1.3
	60	0.8	83	80	0.7 to 1.2
	40	0.8	88	80	0.5 to 1.5
	20	0.8	93	80	0.4 to 1.7
	<10	0.8	99	80	0.4 to 1.7
Sill grille, straight vanes	80	1.7	61	60	1.5 to 1.7
	60	1.7	72	70	1.4 to 1.7
	40	1.3	86	80	1.2 to 1.8
	20	0.9	95	90	0.8 to 1.3
Sill grille, spread vanes	80	0.7	94	90	0.6 to 1.5
	60	0.7	94	80	0.6 to 1.7
	40	0.7	94	—	—
	20	0.7	94	—	—
Ceiling slot diffusers (for T_{100}/L)	80	0.3	85	80	0.3 to 0.7
	60	0.3	88	80	0.3 to 0.8
	40	0.3	91	80	0.3 to 1.1
	20	0.3	92	80	0.3 to 1.5
Light troffer diffusers	60	2.5	86	80	<3.8
	40	1.0	92	90	<3.0
	20	1.0	95	90	<4.5
Perforated, louvered diffusers	11 to 50	2.0	96	90	1.4 to 2.7
	11 to 50	2.0	96	80	1.0 to 3.4

Fully Stratified Air Distribution

Systems that discharge cool air at low sidewall or floor locations with very little entrainment of (and thus mixing with) room air create (vertical) thermal stratification throughout the space. These **displacement ventilation** systems have been popular in northern Europe for some time. Floor-based outlets in underfloor applications may also be used to provide fully stratified air distribution.

Principles of Operation

Thermal displacement ventilation (TDV) systems (see Figure 1.17) use very low discharge velocities, typically 50 to 70 fpm, to deliver cool supply air to the space. The discharge temperature of the supply air is generally above 60°F, although lower temperatures may be used in industrial applications, exercise or sports facilities, and transient areas. The cool air is negatively buoyant compared to ambient air and drops to the floor after discharge. It then spreads across the lower level of the space.

As convective heat sources (see Figure 1.17) in the space transfer heat to the cooler air around them, natural convection currents form and rise along the heat transfer boundary. Without significant room air movement, these currents rise to form a convective heat plume around and above the heat source. As the plume rises, it expands by entraining surrounding air. Its growth and ascent are proportional to the heat source's size and intensity and temperature of ambient air above it. Ambient air from below and around the heat source fills the void created by the rising plume. If the heat source is near the floor (e.g., an occupant), the plume entrains cool, conditioned air from the floor level, which is drawn to the respiration level, and serves as the source of inhaled air. Exhaled air rises with the escaping heat plume, because it is warmer and more humid than the ambient air. Convective heat from sources located above the occupied zone has little effect on occupied-zone air temperature.

At a certain height, where plume temperature equals ambient temperature, the plume disintegrates and spills horizontally. Two distinct zones are thus formed in the room: a lower occupied zone with little or no recirculation flow (close to displacement flow), and an upper zone with recirculation flow. The boundary between these two zones is called **shift zone**. The shift zone height is calculated as the height above the floor where the total amount of air carried in convective plumes above heat sources equals the supply airflow distributed through displacement diffusers. Actual and simplified representations of the temperature gradient in the space are shown in Figure 1.18.

Outlet Characteristics

Displacement outlets are designed for average face velocities between 50 and 70 fpm, and are typically in a low sidewall or floor location. Return or exhaust air intakes should always be located above the occupied zone for human thermal comfort applications.

Displacement outlets are available in a number of configurations and sizes. Some models are designed to fit in corners or along sidewalls, or stand freely as columns. It is important to consider the degree of flow equalization the outlet achieves, because use of the entire outlet surface for air discharge is paramount to minimizing clear zones and maintaining acceptable temperatures at the lower levels of the space.

Stationary occupants should not be subjected to discharge velocities exceeding about 40 fpm because air at the ankle level within this velocity envelope tends to be quite cool. As such, most outlet manufacturers define a **clear zone** in which location of stationary, low-activity occupants is strongly discouraged, but transient occupancy, such as in corridors or aisles, is possible. Occupants with high activity levels may also find the clear zone acceptable.

Unlike mixed systems, outlets in thermal displacement systems discharge air at very low velocities, resulting in very little mixing. As such, design of these systems primarily involves determining a supply airflow rate to manage the thermal gradients in the space in accordance with ASHRAE comfort guidelines. ASHRAE Standard 55 recommends that the vertical temperature difference between the ankle and head levels of space occupants be limited to no more than 5.4°F to maintain a high degree (>95%) of occupant satisfaction.

Application Considerations

Displacement ventilation is a cooling-only method of room air distribution. For heating, a separate system is generally recommended. Displacement ventilation can be used successfully in

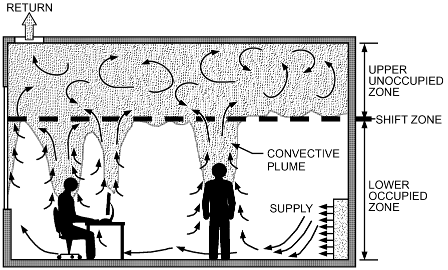


Figure 1.17 Displacement Ventilation System Characteristics [2015A, Ch 57, Fig 4]

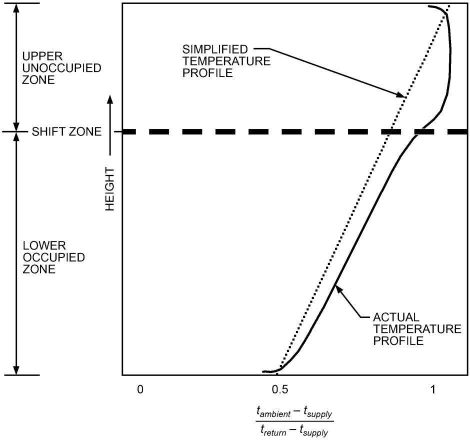


Figure 1.18 Temperature Profile of Displacement Ventilation [2015A, Ch 57, Fig 5]

combination with radiators and convectors installed at the exterior walls to offset space heat losses. Radiant heating panels and heated floors also can also be used with displacement ventilation. To maintain displacement ventilation, outlets should supply ventilation air about 4°F lower than the desired room temperature.

Thermal displacement ventilation systems can be either constant or variable air volume. A thermostat in a representative location in the space or return plenum should determine the delivered air volume or temperature. If the time-averaged requirements of ASHRAE Standard 62.1-2004 are met, intermittent on/off airflow control can be used.

Avoid using thermal displacement and mixed air systems in the same space, because mixing destroys the natural stratification that drives the thermal displacement ventilation system. Thermal displacement systems can be complemented by hydronic systems such as chilled floors. Use caution when combining chilled ceilings, beams, or panels with fully stratified systems, because cold surfaces in the upper zone of the space may recirculate contaminants stratified in the upper zone back into the occupied zone.

Partially Mixed Air Distribution

A partially mixed system's characteristics fall between a fully mixed system and a fully stratified system. It includes both a high-velocity mixed air zone and a low-velocity stratified zone where room air motion is caused by thermal forces. For example, floor-based outlets, when operating in a cooling mode with relatively high discharge velocities (>150 fpm), create mixing, thus affecting the amount of stratification in the lower portions of the room. In the upper portions of the room, away from the influence of floor outlets, room air often remains thermally stratified in much the same way as displacement ventilation systems.

Principles of Operation

Supply air is discharged, usually vertically, at relatively high velocities and entrains room air in a similar fashion to outlets used in mixed air systems. This entrainment, as shown in Figure 1.19 reduces the temperature and velocity differentials between supply and ambient room air. This discharge results in a vertical plume that rises until its velocity is reduced to about 50 fpm. At this point, its kinetic energy is insufficient to entrain much more room air, so mixing stops. Because air in the plume is still cooler than the surrounding air, the supply air spreads horizontally across the space, where it is entrained by rising thermal plumes generated by nearby heat sources.

Research and experience have shown that the amount of room air stratification varies depending on design, commissioning, and operation. Control of stratification includes the following considerations:

- By reducing airflow and mixing in the occupied zone, fan energy can be reduced and stratification can be increased, approaching a reasonable target at 3°F to 4°F temperature difference from head to ankle height, which satisfies ASHRAE Standard 55-2010.
- By increasing airflow and mixing in the occupied zone, excessive stratification can be avoided, thereby improving thermal comfort.

Figure 1.19 shows one example of the resulting room air distribution in which the room air is mixed in the **lower mixed zone**, which is bounded by the floor and the elevation (**throw height**) at which the 50 fpm terminal velocity occurs. At this elevation, stratification begins to occur and a linear temperature gradient, similar to that found in thermal displacement systems, forms and extends through the **stratified zone**. As with thermal displacement ventilation, convective heat plumes from space heat sources draw conditioned air from the lower (mixed) level through the stratified zone and to the overhead return location. A third zone, referred to as the **upper mixed zone**, may exist where the volume of rising heat plumes terminate. Although velocities in this area are quite low, the air tends to be mixed.

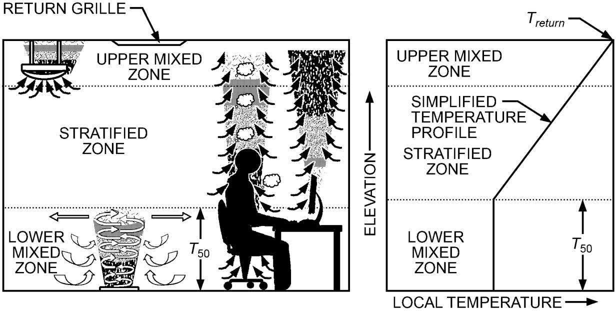


Figure 1.19 UFAD System in Partially Stratified Application [2015A, Ch 57, Fig 7]

Outlet Characteristics

One outlet type is a swirl diffuser with a high-induction core, which induces large amounts of room air to quickly reduce supply to ambient air velocity and temperature differentials. Supply air is injected into the room as a swirling vertical plume close to the outlet. Properly selected, these outlets produce a limited vertical projection of the supply air plume, restricting mixing to the lower portions of the space. Most of these outlets allow occupants to adjust the outlet airflow rate easily. Other versions incorporate automatically controlled dampers that are repositioned by a signal from the space thermostat and/or central control system.

Another category includes more conventional floor grilles designed for directional discharge of supplied airflow. These grilles may be either linear or modular in design, and may allow occupants to adjust the discharge air pattern by repositioning the core of the outlet. Most floor grilles include an integral actuated damper, or other means, that automatically throttles the volume of air in response to the zone conditioning requirements.

Room air induction allows UFAD diffusers to comfortably deliver supply air a few degrees cooler than possible with outlets used for thermal displacement ventilation outlets. The observance of clear, or adjacent, zones above and around the diffusers, where stationary occupants should not reside, is recommended. Outlet manufacturers typically identify such restrictive areas in their product literature.

As for thermal displacement systems, design involves determining a supply airflow rate that limits thermal gradients in the occupied zone in accordance with ASHRAE Standard 55 guidelines. ASHRAE Standard 55 recommends that the vertical temperature difference between the ankle and head level of space occupants be limited to no more than 5.4°F if a high degree (>95%) of occupant comfort is to be maintained.

Application Considerations

Some considerations include the following:

- Supply temperatures in the access floor cavity should be kept at 60°F or above, to minimize the risk of condensation and subsequent mold growth.
- Most UFAD outlets can be adjusted automatically by a space thermostat or other control system, or manually by the occupant. In the latter case, outlets should be located within the workstation they serve.
- Use of manually adjusted outlets should be restricted to open office areas where cooling loads do not tend to vary considerably or frequently. Perimeter areas and conference rooms require automatic control of supply air temperatures and/or flow rates because their thermal loads are highly transient.
- Heat transfer to and from the floor slab affects discharge air temperature and should be considered when calculating space airflow requirements. Floor plenums should be well sealed to minimize air leakage, and exterior walls should be well insulated and have good vapor retarders. Night and holiday temperature setbacks should likely be avoided, or at least reduced, to minimize plenum condensation and thermal mass effect problems. With air-side economizers, using enthalpy control rather than dry-bulb control can help reduce hours of admitting high moisture-content air, thus also reducing the potential for condensation in the floor plenums.
- Avoid using stratified and mixed air systems in the same space, because mixing destroys the natural stratification that drives the stratified system.
- Return static pressure drop should be relatively equal throughout the spaces being served by a common UFAD plenum. This reduces the chance of unequal pressurization in the UFAD plenum.

Return Air Inlets

The success of a mixed air distribution system depends primarily on supply diffuser location. Return grille location is far less critical than with outlets. In fact, the return air intake affects room air motion only immediately around the grille. Measurements of velocity near a return air grille show a rapid decrease in magnitude as the measuring device is moved away from the grille face. Table 1.18 shows recommended maximum return air grille velocities as a function of grille location. Every enclosed space should have return/transfer inlets of adequate size per this table.

For stratified and partially mixed air distribution systems, there are advantageous locations for return air inlets. For example, an intake can be located to return the warmest air in cooling season.

If the outlet is selected to provide adequate throw and directed away from returns or exhausts, supply short-circuiting is normally not a problem. The success of this practice is confirmed by the availability and use of combination supply and return diffusers.

Table 1.18 Recommended Return Inlet Face Velocities
[2015A, Ch 57, Tbl 4]

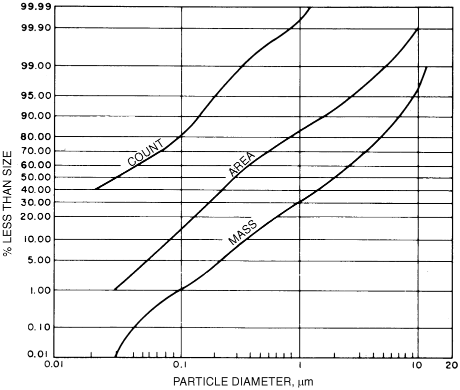
Inlet Location	Velocity Across Gross Area, fpm
Above occupied zone	>800
In occupied zone, not near seats	600 to 800
In occupied zone, near seats	400 to 600
Door or wall louvers	200 to 300
Through undercut area of doors	200 to 300

2. AIR CONTAMINANTS AND CONTROL

Table 2.1 National Ambient Air Quality Standards for the United States
[2017F, Ch 11, Tbl 12]

Contaminant	Primary or Secondary Standard	Averaging Time	Level	Details
Carbon monoxide	Primary	1 h	35 ppm	Not to be exceeded more than once per year
		8 h	9 ppm	
Nitrogen dioxide	Primary	1 h	100 ppb	98th percentile, averaged over 3 years
	Primary/secondary	1 yr	53 ppb	Annual mean
Ozone	Primary/secondary	8 h	75 ppb	Annual fourth-highest daily maximum 8 h concentration, averaged over 3 years
Sulfur dioxide	Primary	1 h	75 ppb	99th percentile of 1 h daily maximum concentrations, averaged over 3 years
	Secondary	3 h	500 ppb	Not to be exceeded more than once per year
Particulate, $PM_{2.5}$[a]	Primary/secondary	24 h	35 $\mu g/m^3$	98th percentile, averaged over 3 years
		1 yr	15 $\mu g/m^3$	Annual mean, averaged over 3 years
Particulate, PM_{10}[b]	Primary/secondary	24 h	150 $\mu g/m^3$	Not to be exceeded more than once per year on average over 3 years
Lead (Pb) in particles	Primary/secondary	3 mo	0.15 $\mu g/m^3$	Not to be exceeded

[a] $PM_{2.5}$ = particulates below 2.5 μm diameter.
[b] PM_{10} = particulates below 10 μm diameter.
ppb = parts per 10^9
Source: *National Ambient Air Quality Standards* (NAAQS), U.S. Environmental Protection Agency, Washington, DC, 2015.



Count curve: Based on measurements by electron microscope.
Area curve: Calculated.
Mass curve: Solid section based on measurements by sedimentation.

Figure 2.1 Particle Size Distribution of Atmospheric Dust

Electronic Air Cleaners

Electronic air cleaners use electrostatic precipitation to remove and collect particulate contaminants such as dust, smoke, and pollen. Wires with a positive direct current potential of between 6 and 25 kV DC are suspended equidistant between grounded plates, creating an ionizing field for charging particles.

The collecting plate section consists of parallel plates with a positive voltage of 4 to 10 kV (dc) applied to alternate plates. Plates that are not charged are at ground potential. As particles pass into this section, they are forced to the plates by the electric field on the charges they carry, and thus are removed from the airstream and collected by the plates.

Electronic air cleaners typically operate from a 120- or 240-V AC single-phase electrical service. Power consumption ranges from 20 to 40 watts per 1000 cfm of capacity.

This type of air filter can remove and collect airborne contaminants with average efficiencies of up to 98% at low airflow velocities (150 to 350 fpm) when tested per ASHRAE Standard 52.1. Efficiency decreases (1) as the collecting plates become loaded with particulates, (2) with higher velocities, or (3) with nonuniform velocity.

As with most air filtration devices, the duct approaches to and from the air cleaner housing should be arranged so that the airflow is distributed uniformly over the face area. Panel prefilters should also be used to help distribute the airflow and to trap large particles that might short out or cause excessive arcing within the high-voltage section.

Bioaerosols

Bioaerosols, particulates of biological origin, are of concern in indoor air due to their association with allergies and asthma and their ability to cause disease.

Airborne viral and bacterial aerosols are generally transmitted by droplet nuclei, averaging about 3 μm in diameter. Fungal spores range between 2 μm and 5 μm. Fifty to seventy percent dust spot efficiency filters can remove most microbial agents 1 μm to 2 μm in diameter. Sixty percent dust spot efficiency filters can remove 85% or more of 2.5 μm particles; while 85% filters can remove about 96%.

Filter Installation

Efficiency is sharply reduced if air leaks through poorly designed or installed frames. Install filters with face area at right angles to air flow whenever possible. Install high-efficiency filters as close as possible to the room to minimize pickup of particles between filter and outlet. Provide at least 20 in. access in front of or behind filters, or both.

ASHRAE Air Filtration Standards

ASHRAE Standard 52.1 (withdrawn in 2009) contained a test procedure for measuring the weight of a synthetic dust captured by a filter (arrestance). This gives a standard for comparing ability of fibers to remove coarse particles. ASHRAE Standard 52.2 contains the test procedure for comparing filter removal efficiency by particle size. For more efficient filters, arrestance is essentially 100% efficient, and their efficiency in removing smaller particles is tested. The dust spot efficiency of Standard 52.1 is replaced by the Standard 52.2 tests and classification.

Table 2.2 Filter Minimum Efficiency Reporting Value (MERV) Parameters

Standard 52.2 Minimum Efficiency Reporting Value (MERV)	Composite Average Particle Size Efficiency, % in Size Range, μm			Average Arrestance,%, by Standard 52.1 Method	Minimum Final Resistance	
	Range 1 0.30–1.0	Range 2 1.0–3.0	Range 3 3.0–10.0		Pa	in. of water
1	n/a	n/a	$E_3 < 20$	$A_{avg} < 65$	75	0.3
2	n/a	n/a	$E_3 < 20$	$65 \leq A_{avg} < 70$	75	0.3
3	n/a	n/a	$E_3 < 20$	$70 \leq A_{avg} < 75$	75	0.3
4	n/a	n/a	$E_3 < 20$	$75 \leq A_{avg}$	75	0.3
5	n/a	n/a	$20 \leq E_3 < 35$	n/a	150	0.6
6	n/a	n/a	$35 \leq E_3 < 50$	n/a	150	0.6
7	n/a	n/a	$50 \leq E_3 < 70$	n/a	150	0.6
8	n/a	n/a	$70 \leq E_3$	n/a	150	0.6
9	n/a	$E_2 < 50$	$85 \leq E_3$	n/a	250	1.0
10	n/a	$50 \leq E_2 < 65$	$85 \leq E_3$	n/a	250	1.0
11	n/a	$65 \leq E_2 < 80$	$85 \leq E_3$	n/a	250	1.0
12	n/a	$80 \leq E_2$	$90 \leq E_3$	n/a	250	1.0
13	$E_1 < 75$	$90 \leq E_2$	$90 \leq E_3$	n/a	350	1.4
14	$75 \leq E_1 < 85$	$90 \leq E_2$	$90 \leq E_3$	n/a	350	1.4
15	$85 \leq E_1 < 95$	$90 \leq E_2$	$90 \leq E_3$	n/a	350	1.4
16	$95 \leq E_1$	$95 \leq E_2$	$95 \leq E_3$	n/a	350	1.4

Table 2.3 Filter Application Guidelines [2016S, Ch 29, Tbl 3]

	Standard 52.2 MERV	Intended Standard 52.1 Value	Arrestance Value	Example Range of Contaminants Controlled	Example Applications	Sample Air Cleaner Type(s)
HEPA Filters	N/A	N/A	N/A	0.12 to 0.5 μm particles: virus (unattached), carbon dust, sea salt, radon progeny, combustion smoke	Cleanroom, pharmaceutical manufacturing and exhaust, radioactive material handling and exhaust, orthopedic and organ transplant surgery, carcinogenic materials, welding fumes	SULPA >99.999% 0.1 to 0.2 μm IEST type F (ceiling panel) ULPA >99.999% 0.3 μm IEST type D (ceiling panel) HEPA > 99.99% 0.3 μm IEST type C (ceiling or up to 12 in. deep) HEPA > 99.97% 0.3 μm IEST type A (box style 6 to 12 in. deep)
E-1 Range	MERV 16 MERV 15 MERV 14	Intended to replace 70 to 98% dust-spot efficiency filters	N/A	0.3 to 1.0 μm size range: bacteria, smoke (ETS), paint pigments, face powder, some virus, droplet nuclei, insecticide dusts, soldering fumes	Day surgery, general surgery, hospital general ventilation, turbo equipment, compressors, welding/soldering air cleaners, prefilters to HEPAs, LEED for existing (EB) and new (NC) commercial buildings, smoking lounges	Box-style wet-laid or lofted fiberglass, box-style synthetic media, minipleated synthetic or fiberglass paper, depths from 4 to 12 in., Pocket filters of fiberglass or synthetic media 12 to 36 in.
E-2 Range	MERV 12 MERV 11 MERV 10 MERV 9	Intended to replace 50 to 80% dust-spot efficiency filters	N/A	1.0 to 3.0 μm size range: milled flour, lead dust, combustion soot, *Legionella*, coal dust, some bacteria, process grinding dust	Food processing facilities, air separation plants, commercial buildings, better residential, industrial air cleaning, prefiltration to higher-efficiency filters, schools, gymnasiums	Box-style wet-laid or lofted fiberglass, box-style synthetic media, minipleated synthetic or fiberglass paper, depths from 2 to 12 in. Pocket filters either rigid or flexible in synthetic or fiberglass, depths from 12 to 36 in.
E-3 Range	MERV 8 MERV 7 MERV 6 MERV 5	Intended to replace 20 to 60% dust-spot efficiency filters	N/A	3.0 to 10 μm size range: pollens, earth-origin dust, mold spores, cement dust, powdered milk, snuff, hair spray mist	General HVAC filtration, industrial equipment filtration, commercial property, schools, prefilter to high-efficiency filters, paint booth intakes, electrical/phone equipment protection	Wide range of pleated media, ring panels, cubes, pockets in synthetic or fiberglass, disposable panels, depths from 1 to 24 in.
	MERV 4 MERV 3 MERV 2 MERV 1	<20% <20% <20% <20%	>70% >70% >65% <65%	Arrestance method	Protection from blowing large particle dirt and debris, industrial environment ventilation air	Inertial separators

Note: MERV also includes test airflow rate, but it is not shown here because it is of no significance for the purposes of this table.
N/A = not applicable.

Table 2.4 Sources and Indoor and Outdoor Concentrations of Selected Indoor Contaminants [2017F, Ch 11, Tbl 13]

Contaminant	Sources of Indoor Contaminants	Typical Indoor Concentration	Typical Outdoor Concentration	Locations
Carbon monoxide	Combustion equipment, engines, faulty heating systems	0.5 to 5 ppm[a] (without gas stoves) 5 to 15 ppm[a] (with gas stoves)	2 ppm[a]	Indoor ice rinks, homes, cars, vehicle repair shops, parking garages
$PM_{2.5}$	Stoves, fireplaces, cigarettes, condensation of volatiles, aerosol sprays, cooking	7 to 10 μg/m^3[a]	<10 μg/m^3[a]	Homes, offices, cars, public facilities, bars, restaurants
PM_{10}	Combustion, heating system, cooking	40 to 60 μg/m^3[a]	60 μg/m^3[a]	Homes, offices, transportation, restaurants
Organic vapors	Combustion, solvents, resin products, pesticides, aerosol sprays, cleaning products, building materials, paints	Different for each VOC[c] (2 to 5 times outdoor levels)	See 2017F, Ch 11, Tbl 11	Homes, restaurants, public facilities, offices, hospitals
Nitrogen dioxide	Combustion, gas stoves, water heaters, gas-fired dryers, cigarettes, engines	<8 ppb[a] (without combustion appliances)		
		>15 ppb with combustion appliances)	15 ppb[a]	Homes, indoor ice rinks
Nitric oxide	Combustion, gas stoves, water heaters, gas-fired dryers, cigarettes, engines	—	Various	Homes, any building with combustion source
Sulfur dioxide	Heating system	20 μg/m^3[b]	<20 μg/m^3[b]	Mechanical/furnace rooms
			3 ppb[a]	
Formaldehyde	Insulation, product binders, pressed wood products, carpets	0.1 to 0.3 ppm[a]	NA	Homes, schools, offices
Radon and progeny	Building materials, groundwater, soil	1.3 pCi/L[a]	4 pCi/L[a]	Homes, schools
Carbon dioxide	Combustion appliances, humans, pets	600 to 1000 ppm[c]	300 to 500 ppm[c]	
Biological contaminants	Humans, pets, rodents, insects, plants, fungi, humidifiers, air conditioners	NA	NA (lower than indoor levels)	Homes, hospitals, schools, offices, public facilities
Ozone	Electric arcing, electronic air cleaners, copiers, printers	42 ppb[d]	70 ppb[a]	Airplanes, offices, homes

Sources:
[a]EPA (2011)
[b]NRC (1981)
[c]Seppänen et al. (1999) and ASHRAE Standard 62.1, Appendix C
[d]Weschler (2000)

NA = not applicable
ppb = parts per 10^9

Table 2.5 Media Selection by Contaminant [2015A, Ch546, Tbl 7]

Gaseous Contaminant	PIA	AC	AIC	BIC	Gaseous Contaminant	PIA	AC	AIC	BIC	Gaseous Contaminant	PIA	AC	AIC	BIC
Acetaldehyde	1	2			Dichlorofloromethane		1			Methyl formate	2	1		
Acetic acid (!)	1	2		2,1	R-114 (see note)		1			Methyl isobutyl ketone	2	1		
Acetic anhydride (!)	1,2	1		2	Diethylamine	2	1			Methyl sulfide	1			
Acetone (!)	1	2			Dimethylamine		1	2		Methyl vinyl ketone	2	1		
Acetylene	1				Dioctyl phthalate		1			Naphtha		1		
Acrolein	1	2			Dioxane	1	2			Naphthalene		1		
Acrylic acid (!)	1	1		2	Ethanol	1	2			Nicotine	1	2		
Allyl sulfide	1	2			Ethyl acetate	2	1			Nitric acid				1
Ammonia (NH_3)			1		Ethyl chloride (!)	1,2	2,1			Nitric oxide (NO)	1			2
Aniline	2	1			Ethylene (C_2H_4)	1				Nitrobenzene		1		
Arsine	1				Ethylene oxide	1	2			Nitrogen dioxide	1			2
Benzene		1			Ethyl ether	2	1			Nitromethane	1			
Borane (!)	1	2,2			Ethyl mercaptan (!)	1,1	2		2	Nitrous oxide				1
Bromine		1			Formaldehyde	1				Octane (!)	2	1,1		
1,3 Butadiene	1	2			Gasoline	1				Ozone (O_3) (!)	2	1,1		
Butane		1			General halocarbons		1			Perchloroethylene	2	1		
2-Butanone	1	2			General hydrocarbons	2	1			Peroxy acetyl nitrate (PAN)		1		
2-Butoxyethanol	2	1			General VOC	2	1			Phenol	2	1		
Butyl acetate (!)	1,2	2,1			Heptane		1			Phosgene	2	1		
Butyl alcohol	2	1			Hydrogen bromide		2		1	Phosphine	1			
Butyl mercaptan	2	1			Hydrogen chloride		2		1	Putrescine	1	2		
Butylene	2	1			Hydrogen cyanide	1				Pyridine (!)	1	1		
Butyne	2	1			Hydrogen fluoride	1			1	Skatole	2	1		

Table 2.5 Media Selection by Contaminant [2015A, Ch546, Tbl 7] *(Continued)*

Gaseous Contaminant	PIA	AC	AIC	BIC	Gaseous Contaminant	PIA	AC	AIC	BIC	Gaseous Contaminant	PIA	AC	AIC	BIC
Butyraldehyde	2	1			Hydrogen iodide	2				Silane	1			
Butyric acid		1		2	Hydrogen selenide					Stoddard solvent	1			
Cadaverine	2	1			Hydrogen sulfide	1			1	Stibine	1			
Camphor		1			Iodine		1			Styrene (!)	2	1,1		
Carbon dioxide (CO_2)	Carbon w/catalyst				Iodoform	2	1			Sulfur dioxide	1			1
Carbon disulfide	2	1			Isopropanol	2	1			Sulfur trioxide	1			1
Carbon monoxide (CO)	Carbon w/catalyst				Kerosene		1			Sulfuric acid		2		1
Carbon tetrachloride		1			Lactic acid		1			Toluene		1		
Chlorine (Cl_2)				1	Menthol	2	1			Triethylamine		2	1	
Chloroform		1			Mercury vapor		Impreg. AC			Trichlorethylene		1		
Creosote (!)	1,2	2,1			Methanol	2	1			1,1,1, trichloroethane (!)	1	2,1		
Cyclohexane		1			Methyl acrylate	2	1			R-11 (see below)		1		
Cyclohexanol	2	1			Methyl bromide (!)	2,1	1			Turpentine	2	1		
Cyclohexanone	2	1			Methyl butyl ketone (!)	1,2	2,1			Urea (!)	2	1,1		
Cyclohexene		1			Methyl cellosolve acetate	2	1			Uric acid (!)	1	1		2,2
Decane		1			Methylchloroform		1			Vinyl chloride		1		
Diborane	1				Methylcyclohexane		1			Xylene		1		
Dichlorobenzene		1			Methylene chloride		1							

1 = primary media selection for contaminant; 2 = secondary media selection.
PIA = permanganate-impregnated alumina; AC = activated carbon; AIC = acid-impregnated carbon; BIC = base-impregnated carbon.
R-114 is dichlorotetrafluoroethane; R-11 is trichlorofluoromethane.

Comments: Some contaminant molecules have isomers that, because they have different physical properties (boiling point, vapor pressures), require different treatment methods. For some contaminants, preferred treatment is ion exchange or another (nonlisted) impregnated carbon. For some contaminants, manufacturer recommendations differ. “!” is used to identify these cases.

Table 2.6 Example Generation of Gaseous Contaminants by Building Materials [2015A, Ch 46, Tbl 2]

Contaminant	Emission Factor Averages (ranges), µg/(h·m²)					
	Acoustic Ceiling Panels	Carpets	Fiberboards	Gypsum Boards	Paints on Gypsum Board	Particle Boards
4-Phenylcyclo-hexene (PCH)		8.4 (n.d.- 85)				
Acetaldehyde		2.8 (n.d.- 37)	9.0 (n.d.-32)			28 (n.d.-55)
Acetic acid			8.4 (n.d.-26)			
Acetone	12 (n.d.-33)		35 (n.d.-67)	37 (n.d.-110)	35 (n.d.-120)	
Ethylene glycol			140 (n.d.-290)		19 (n.d.-190)	160 (140-200)
Formaldehyde	5.8 (n.d.-25)	3.6 (n.d.- 41)	220 (n.d.-570)	6.8 (n.d.-19)		49 (n.d.-97)
Naphthalene		11 (n.d.-59)	3.0 (n.d.-8.2)			
n-Heptane			21 (n.d.-53)			
Nonanal	4.9 (1.7-11)	11 (n.d.-68)		10 (n.d.-28)	3.7 (n.d.-24)	
Toluene			19 (n.d.-46)			
TVOC*	32 (3.2-150)	1900 (270-9100)	400 (52-850)	15 (n.d.-61)	2500 (170-6200)	420 (240-510)

Table 2.6 Example Generation of Gaseous Contaminants by Building Materials [2015A, Ch 46, Tbl 2] ***(Continued)***

Contaminant	Emission Factor Averages (ranges) in µg/(h·m²)					
	Plastic Laminates and Assemblies	Non-Rubber-Based Resilient Flooring	Rubber-Based Resilient Flooring	Tackable Wall Panels	Thermal Insulations	Wall Bases (Rubber-Based)
1,2,4-Trimethylbenzene			210 (n.d.-590)			
2-Butoxy-ethanol		2.7 (n.d.- 24)	1.6 (n.d.-24)			
Acetaldehyde		11 (n.d.- 49)				
Acetone	75 (4.8-150)	120 (n.d.- 830)			12 (1.8-21)	220 (30-400)
Butyric acid		0.51 (n.d. - 5.1)				
Dodecane			1.3 (n.d.-20)			
Ethylene glycol		38 (n.d.- 210)				
Formaldehyde	13 (n.d.-29)	6.8 (n.d.- 79)			5.9 (0.35-14)	32 (3.6-61)
Naphthalene		3.4 (n.d.- 14)	5.6 (n.d.-28)	6.6 (6.6)		
n-Butanol						100 (n.d.-200)
Nonanal		5.7 (n.d.- 19)	1.4 (n.d.-11)		1.8 (0.57-4)	
Octane						150 (n.d.-300)
Phenol	9.4 (4.4-19)	35 (n.d.- 310)				340 (n.d.-680)
Toluene		5.1 (n.d.- 12)				
Undecane						140 (13-270)
TVOC*	160 (6.3-310)	680 (100-2100)	15000 (1500-100000)	270 (100-430)	7.5 (0.57-26)	7100 (1200-13000)

Source: Material Emissions Study, California Integrated Waste Management Board, Publication 433-03-015, 2003.
n.d. = nondetectable
* TVOC concentrations calculated from total ion current (TIC) from GC/MS analysis by adding areas of integrated peaks with retention times greater than 5 min, subtracting from sum of area of internal standard chlorobenzene-d5, and using response factor of chlorobenzene-d5 as calibration.

Kitchen Ventilation

From ASHRAE Standard 154-2016, *Ventilation for Commercial Cooking Operations*

(See complete standard for detailed guidance.)

This section provides guidance on hoods used in commercial kitchens. For information on laboratory applications, see Chapter 16 of the 2015 *ASHRAE Handbook—HVAC Applications*; also see *ASHRAE Laboratory Design Guide*. For cleanroom applications, see *ASHRAE Design Guide for Cleanrooms*; also see Chapter 18 of the 2015 *ASHRAE Handbook—HVAC Applications*.

3. DEFINITIONS

air curtain supply: see *replacement air, makeup air (dedicated replacement air)*, *air curtain*.

appliance: a cooking device or apparatus used in a kitchen that consumes energy provided by gas, electricity, solid fuel, steam, or another fuel source.

appliance duty level: an appliance rating category based on the exhaust airflow required to capture, contain, and remove the cooking effluent and products of combustion under typical operating conditions with a nonengineered wall-mounted canopy hood (based on ASHRAE RP-1362). This is different from the historical approach, in which duty levels were based on the temperature of the cooking surface. The following appliance duty classifications are used in this standard:

a. ***light:*** a cooking process requiring an exhaust airflow rate of less than 200 cfm/ft for capture, containment, and removal of the cooking effluent and products of combustion.
b. ***medium:*** a cooking process requiring an exhaust airflow rate of 200 to 300 cfm/ft for capture, containment, and removal of the cooking effluent and products of combustion.
c. ***heavy:*** a cooking process requiring an exhaust airflow rate of 300 to 400 cfm/ft for capture, containment, and removal of the cooking effluent and products of combustion.
d. ***extra-heavy:*** a cooking process requiring an exhaust airflow rate greater than 400 cfm/ft for capture, containment, and removal of the cooking effluent and products of combustion.

approved: acceptable to the authority having jurisdiction.

back-wall supply: see *replacement air*, *makeup air (dedicated replacement air)*, *back-wall*.

baffle filter: see *grease removal device*.

capture area: the area within an exhaust hood that contains cooking effluent until it is exhausted.

capture and containment (C&C): an exhaust hood's ability to capture and contain the cooking effluent and heat generated during cooking operations.

cartridge filter: see *grease removal device*.

centrifugal fan: see *exhaust fan*.

certified: see *listed*.

compensating hood: see *replacement air*, *makeup air (dedicated replacement air)*, *internal*.

commercial cooking appliance: an appliance specifically designed to be used in a food-service-establishment kitchen, such as, but not limited to, a restaurant or cafeteria kitchen. Appliances designed for residential use shall be treated as commercial appliances when installed in commercial food-service establishments.

condensate hood: see *hood, Type II hood*.

cooking effluent: the emissions generated by cooking appliances during their operation; for example, convective heat, moisture, vapor, products of combustion, smoke, and particulate matter.

demand-control ventilation: a ventilation system that utilizes an automatically controlled variable-speed device, such as a multispeed fan or variable-speed drive, to modulate the exhaust airflow rates in response to the variation in cooking load.

duct: a conduit for conveying cooking effluent from the hood to the outdoors or for conveying replacement air into a room or space.

ductless hood: see *recirculating hood*.

end skirt: see *side panel*.

exfiltration: leakage or flow of indoor air out of the building or space through openings in the building or space envelope, whether intentional or unintentional. The driving force for exfiltration is a positive pressure in the building or space relative to the exterior of the building envelope.

exhaust fan: a fan used to exhaust cooking effluent collected by a hood. Also referred to as a *power roof ventilator*. The majority of these fans have a centrifugal fan wheel. Fans used in Type I hood applications must include provisions for handling grease and access for cleaning.

a. ***in-line exhaust fan*** **or** ***tubular centrifugal fan:*** a fan designed for mounting indoors or outdoors in a section of duct between the hood and the point of discharge. Air enters the fan axially and discharges linear to the entrance.
b. ***roof exhaust fan*** **or** ***power roof ventilator:*** a fan designed for curb mounting on a roof and that discharges downward toward the roof, vertically up away from the roof, or horizontally away from the building. Fans that discharge downward may be used only for Type II hood applications.
c. ***up-blast exhaust fan:*** a fan designed for curb-mounting on a roof or for wall mounting. Air enters the fan axially but discharges radially from the centrifugal impeller and turns 90 degrees to exit the fan vertically where roof-mounted and horizontally where wall-mounted.
d. ***side-wall exhaust fan:*** a fan design similar to an up-blast exhaust fan but designed to mount outdoors on the side wall of a building. The mounting arrangement and internal construction may be specific to side discharge orientation. The fan discharges horizontally away from the building.
e. ***utility-set exhaust fan:*** a fan typically designed with a single-inlet, a scroll housing, and a backward-inclined or an airfoil centrifugal impeller. It can provide a higher static efficiency capability than a typical power roof ventilator. Air enters the impeller axially and leaves it in a substantially radial direction. These can be mounted indoors or outdoors in-line having additional duct between the fan outlet and the point of discharge.

exhaust fire (actuated) damper: a damper arranged to automatically close to restrict the passage of fire airflow into the exhaust duct.

fire resistance rating: the time rating of a material or assembly indicating its ability to withstand exposure to a fire.

fire suppression system: an automatic fire suppression system that is specifically designed to protect Type I hood systems and, where required, the cooking appliances served by the hood system(s).

front-face supply: see *replacement air*, *makeup air (dedicated replacement air)*, *front-face*.

grease duct: a duct system for the conveyance of cooking effluent. The system is designed and installed to reduce the accumulation of combustible condensation, thus reducing the possibility of fire within the duct system.

grease laden: containing grease particles and/or grease vapor.

grease removal device: a device designed and installed in a Type I hood to remove grease vapor and/or particles from the airstream. As used in this standard, the term refers to devices that are certified to UL Standard 1046, *Grease Filters for Exhaust Ducts*, or to UL Standard 710, *Exhaust Hoods for Commercial Cooking Equipment*, as part of the hood. Devices include but are not limited to the following:

a. ***baffle filter:*** a filter typically having a series of vertical baffles designed to capture grease and drain to a grease trough. Filters are removable for cleaning and maintenance of the hood.

b. ***cartridge filter:*** a filter having a horizontal slot opening with a series of internal deflectors designed to capture grease and drain to a grease trough. Filters are removable for cleaning and maintenance of the hood.

c. ***fixed or stationary extractor:*** a device typically having horizontal slot openings with a series of internal deflectors designed to capture grease and drain to a grease trough. Extractors are not removable from the hood and typically have access doors for cleaning and maintenance of the hood.

d. ***multistage extractor or filter:*** these devices consist of a series of two or more grease removal devices located in the hood.

e. ***removable extractor:*** any style of grease removal device that is removable from the hood.

f. ***water wash:*** a version of the fixed extractor that has a system of built-in nozzles for cleaning the grease removal device.

greasetight: designed to prevent the leakage of grease under normal operating conditions.

hood: a device designed to capture and contain cooking effluent, including grease, smoke, steam, heat, and vapor, until it is exhausted through a duct or recirculating system. Hoods are categorized as Type I or Type II.

Type I hood: a hood used for collecting and removing convective heat, grease particulate, condensible vapor, and smoke. This category includes listed grease filters, baffles, or extractors for removing the grease and a fire-suppression system. Type I hoods are installed over cooking appliances, such as ranges, fryers, griddles, broilers, and ovens, that produce smoke or grease-laden vapors. For Type I hoods, the following types of hoods are commonly available.

a. ***wall-mounted canopy hood:*** a wall canopy exhaust hood is mounted against a wall above a single appliance or a line of appliances, or it may be freestanding with a vertical back panel extending from the rear of the appliance(s) to the hood. It typically overhangs the front and sides of the appliance(s) on all open sides of the hood. The wall acts as a back panel, forcing replacement air to be drawn across the front and/or side(s) of the cooking appliance, thus increasing the effectiveness of the hood to capture and contain effluent generated by the cooking operations. Mounting height varies.

b. ***single-island canopy hood:*** a single-island canopy hood is placed over a single appliance or line of appliances. It is open on all sides and overhangs the front, rear, and sides of the appliance(s). A single-island canopy is more susceptible to cross drafts and requires greater exhaust airflow than an equivalent-sized wall-mounted canopy to capture and contain effluent generated by the cooking operations. Mounting height varies.

c. ***double-island canopy hood:*** a double-island canopy hood is placed over back-to-back appliances or lines of appliances. It is open on all sides and overhangs the front and the sides of the appliance(s). It may have a wall panel between the backs of the appliances. Mounting height varies.

d. ***backshelf hood:*** also referred to as a *noncanopy hood*, *low-proximity hood*, or *sidewall hood* (where wall mounted). Its front lower lip is low over the appliance(s) and is typically set back from the front of the appliance(s), which means there may be no front overhang of appliance(s). It is always closed to the rear of the appliances by a panel where freestanding or by a panel or wall when wall mounted, and its height above the cooking surface varies. This style of hood can be constructed with partial end panels to increase its effectiveness in capturing the effluent generated by the cooking operations.

e. ***eyebrow hood:*** an eyebrow hood is mounted directly to the face or top of an appliance above the opening(s) or door(s) from which effluent is emitted, overhanging the front of the opening(s) to capture the effluent. Mounting height is fixed.

f. ***pass-over hood:*** a pass-over hood is a backshelf hood constructed and installed low enough to allow food to be passed over the top. Mounting height varies.

g. ***ventilated ceiling hood:*** typically installed so that the bottom edge of the hood is flush with the ceiling height.

h. ***recirculating hood:*** a hood with an integral or non-integral electric cooking appliance to capture and contain cooking effluent, consisting of a fan, air filtering system, and a fire extinguishing system.

Type II hood: a hood that collects and removes steam, heat, and products of combustion where grease or smoke is not present. It may or may not have grease filters or baffles and is not designed to have a fire-suppression system. A Type II hood can be used where the cooking operation from each appliance underneath the hood does not produce grease in excess of 3.1×10^{-7} lb/ft^3 when measured at 500 cfm exhaust airflow.

hood type: see *hood*, *Type I hood* and *hood, Type II hood*.

infiltration: see *replacement air*, *infiltration*.

interlock, direct: the direct connection between equipment, such as between a common circuit, relays, etc.

interlock, indirect: the indirect connection between equipment through an external controller; for example, a timeclock, building automation system, heat sensor, etc.

internal discharge makeup air: see *replacement air*, *makeup air (dedicated replacement air)*, *internal*.

labeled: equipment or materials to which a label, symbol, or other identifying mark of an organization, acceptable to the authority having jurisdiction, has been attached. This organization is concerned with product evaluation and maintains periodic inspection of the production of labeled equipment or materials. By labeling the equipment or materials, the manufacturer indicates compliance with appropriate standards or performance in a specified manner.

liquid-tight: constructed and performing so as not to permit the leakage of any liquid at any temperature.

listed: equipment or materials included in a list published by an organization acceptable to the authority having jurisdiction. This organization is concerned with product evaluation and performs periodic inspections of production of listed equipment or materials. The list states either that the equipment or material meets appropriate standards or that it has been tested and found suitable for use in a specified manner.

makeup air: see *replacement air*, *makeup air (dedicated replacement air)*.

mounting height: typically the height above the finished floor at which the bottom front edge of a canopy or noncanopy hood is mounted. Listed hoods are typically rated at the minimum and maximum heights above the cooking surface at which they may be mounted.

multiple-hood exhaust system: a system in which more than one hood is connected to a common exhaust duct and fan system.

multistage extractor: see *grease removal device*.

net exhaust flow rate: the exhaust flow rate for a hood, minus any internal discharge makeup airflow rate.

overhang: the horizontal distance that the lower front edge of the hood extends beyond the top horizontal cooking surface of the appliance.

outdoor air: the air outside of a building or air taken from the outdoors and not previously circulated through an HVAC system.

packaged: provided by a manufacturer or vendor in a substantially complete and operable condition.

power roof ventilator: see *exhaust fan.*

recirculating system: systems for control of smoke and grease-laden vapors from commercial cooking appliances that do not exhaust to the outdoors.

replacement air: outdoor air that is used to replace air removed from a building through an exhaust system. Replacement air may be derived from one or more of the following: makeup air, supply air, transfer air, and infiltration. However, the ultimate source of all replacement air is outdoor air.

***makeup air* (*dedicated replacement air*):** air deliberately brought into the building from the outdoors and supplied to the vicinity of an exhaust hood to replace the air and cooking effluent being exhausted. Makeup air is generally filtered and fan-forced, and it may be heated or cooled depending on the requirements of the application. Makeup air may be delivered through outlets integral to the exhaust hood (compensating hoods) or through outlets in the same room. Following are systems commonly used to supply makeup air.

a. ***air curtain:*** air that is introduced vertically downward through a slot, louvers, or holes along the front edge of the hood or around the perimeter of the hood. This design has also been referred to as *down-discharge*.

b. ***back-wall:*** air that is introduced behind and/or below the cooking equipment. A makeup air plenum is installed between the back of the hood and the wall. The full-length plenum typically extends down the wall to approximately 6 in. (150 mm) below the cooking surface or 2 to 3 ft (600 to 900 mm) above the floor. The air supplied by this system mostly enters the kitchen space rather than remain contained in the cooking zone.

c. ***front-face:*** air that is introduced either horizontally or at a slight downward angle from horizontal from the front of the hood plenum.

d. ***internal:*** typically in this design, untempered makeup air is introduced directly into the hood cavity. This design has also been referred to as *short-circuit*.

e. ***perimeter:*** makeup air is discharged vertically downward from a plenum above and outside of the front and sides of the hood.

supply air: air entering a space from an air-conditioning, heating, or ventilating system for the purpose of comfort conditioning. Supply air is generally filtered, fan-forced, and heated, cooled, humidified, or dehumidified as necessary to maintain specified temperature and humidity conditions. Only the quantity of outdoor air within the supply airflow is used as replacement air. Following are systems commonly used for delivering supply air.

a. ***louvered ceiling diffusers:*** ceiling-installed, aspirating, two-, three-, or four-way diffusers. Air should not be directed toward the hood.

b. ***perforated diffusers:*** a ceiling-installed diffuser with a perforated face. Air should not be directed toward the hoods.

c. ***linear slot diffusers:*** ceiling-installed diffusers, typically placed around the perimeter of rooms. These have a higher discharge velocity than a louvered ceiling.

d. ***displacement diffusers:*** floor-, wall-, and ceiling-mounted diffusers with perforated face areas providing laminar low-velocity flow from the face.

transfer air: air transferred from one room to another through openings in the room envelope, whether it is transferred intentionally or not. The driving force for transfer air is generally a small pressure differential between the rooms, although one or more fans may be used. Only that portion of air transferred from another room that originated as outdoor air may be considered transfer air.

infiltration: leakage or flow of outdoor air into the building or space through openings in the building or space envelope, whether intentional or unintentional. The driving force for infiltration is a negative pressure in a space or building relative to the exterior of the building envelope.

setback: the horizontal distance that the top horizontal cooking surface of an appliance extends beyond the front edge of a backshelf or pass-over hood.

short-circuit makeup air: see *replacement air*, *makeup air (dedicated replacement air)*, *internal*.

side panel: a panel that is attached to the lower edge of the end wall of a hood effectively extending the side of the hood down closer to the cooking appliance.

smoke bomb: a device that combusts to produce a large volume of smoke, greater than 400 cfm.

***smoke candle* or *smoke puffer*:** a device that is ignited and combusts to produce smoke or uses a chemical interaction (such as titanium tetrahydrochloride [$TiCL_4$] with humid air) to produce smoke or that emits a silica powder to produce smoke.

solid-fuel cooking appliance: an appliance that combusts a solid fuel such as wood, charcoal, or coal to provide all or part of the heat for the cooking process.

solid-fuel flavoring cooking appliance: an appliance that uses an energy source other than solid fuel to provide all of the heat for the cooking process and also combusts solid fuel solely for the purpose of imparting flavor to the food being cooked.

supply air: see *replacement air*, *supply air.*

transfer air: see *replacement air*, *supply air*.

tubular centrifugal fan: see *exhaust fan*, *tubular centrifugal fan*.

4. EXHAUST HOODS

4.1 Hood Requirements

4.1.1 Type I hoods shall be listed in accordance with UL Standard 710, UL Standard 710B, or UL Standard 710C and shall be installed in accordance with their listing requirements. Type II hoods shall meet the requirements of Sections 4.2 through 4.8. Type I hoods shall meet the requirements of Section 4.2 and Sections 4.5 through 4.8. Where a Type II hood is required, a Type II or listed Type I hood shall be provided.

4.1.1.1 Recirculating systems and recirculating hoods shall be listed in accordance with UL Standard 710B.

4.1.2 A performance test of an installed Type I hood shall be carried out as specified in Section 4.7.

4.2 Where Required

4.2.1 Table 2.7 specifies the Type I hood requirements by appliance description. Table 2.8 specifies the appliance duty classification as it relates to the Type II hood requirements.

Exception: Equipment that is listed in Table 2.8 and the additional heat and moisture loads generated by unhooded electric appliances are included in the sensible and latent cooling load calculations to determine the required capacity of the HVAC system.

4.2.2 Type II hoods shall be installed in accordance with the overhangs shown in Table 2.9 and the net exhaust airflow rates shown in Table 2.10, based on the maximum appliance duty level shown in Table 2.8 for the appliances underneath the hood. Type II hoods may also be installed where cooking or dishwashing appliances produce heat, steam, or products of combustion and do not produce grease in excess of 3.1×10^{-7} lb/ft^3 when measured at an exhaust airflow of 500 cfm.

Informative Note: The 3.1×10^{-7} lb/ft^3 grease concentration when measured at 500 cfm of exhaust air is equivalent to 9.3×10^{-3} lb/h of grease generated by the cooking process.

4.2.3 A Type I hood shall be provided where a cooking operation within a commercial or institutional food service facility produces smoke or grease-laden vapors. Appliances that produce greater than 3.1×10^{-7} lb/ft^3 of grease (when measured at 500 cfm exhaust airflow) shall require a Type I hood. Type I hoods shall be installed in accordance with the overhangs shown in Table 2.9.

Exceptions:

1. Cooking appliances not used for commercial purposes and installed within dwelling units.
2. Appliances listed in Table 2.8 that produce less than 3.1×10^{-7} lb/ft^3 of grease (when measured at 500 cfm exhaust airflow).

Informative Note: The 3.1×10^{-7} lb/ft^3 grease concentration when measured at 500 cfm of exhaust air is equivalent to 9.3×10^{-3} lb/h of grease generated by the cooking process.

4.2.4 Solid-Fuel Cooking Appliances. Exhaust hood systems, including hoods, ducts, and exhaust fans, serving one or more solid-fuel cooking appliances shall be independent of all other exhaust systems.

Exception: Cooking processes that only use solid fuel for flavoring are exempt from this requirement.

4.3 Type II Hood Sizing

4.3.1 Type II hood overhangs and setbacks shall comply with Table 2.9 on all open sides, measured in the horizontal plane from the inside edge of the hood to the edge of the top horizontal surface of the appliance. The vertical distance between the front lower lip of the hood and appliance cooking surface shall not exceed 4 ft.

Exception: A side overhang is not required where full side panels or panels angled from the front lip of the hood to the front of the appliance at cooking-surface height are installed (see Figure 2.2).

4.3.2 The spaces between appliances, the backs of appliances, and the spaces from the appliances to walls or end panels shall be included in overall hood dimensions. In the case of island hoods, appliance flues shall be included in the cooking surface dimensions.

4.3.3 Hoods shall be mounted above the cooking surface as follows:

4.3.3.1 Canopy Type Hood. The vertical distance between the front lower edge of the hood and the cooking surface shall not exceed 48 in. The vertical distance between the front lower edge of the hood and the finished floor shall not be less than 78 in. The inside hood height shall be at least 24 in.

4.3.3.2 Eyebrow Type Hood. The front lower edge of the hood shall be at least 78 in. above the finished floor.

4.3.3.3 Backshelf/Pass-Over Type Hood. The vertical distance between the front lower edge of the hood and the cooking surface shall be a maximum of 24 in. above the cooking surface.

4.4 Type II Hood Airflow Rates

4.4.1 The net exhaust flow rate (see definition in Section 3) for Type II hoods shall comply with Table 2.10. The duty level for the hood shall be the duty level of the appliance that has the highest (heaviest) duty level of all the appliances that are installed underneath the hood according to Table 2.8.

Exception: Type II hoods that are shown by the performance test in Section 4.7 to provide equivalent capture and containment at lower airflow rates.

4.5 Internal Discharge Makeup

4.5.1 Where a Type I or Type II hood has internal discharge makeup air, the makeup airflow shall not exceed 10% of the exhaust airflow. The exhaust airflow required to meet this standard shall be the net exhaust from the hood, calculated as follows:

$$E_{NET} = E_{HOOD} - MA_{ID} \tag{2.1}$$

where

E_{NET} = net hood exhaust, cfm
E_{HOOD} = total hood exhaust, cfm
MA_{ID} = makeup air, internal discharge, cfm

4.6 Type I Hood Grease Extraction

4.6.1 Type I hoods shall be provided with a grease removal device in accordance with their listing.

4.6.2 For grease removal devices that report grease removal efficiency, the efficiency data shall be reported as determined by ASTM F2519.

4.7 Hood Performance Test

4.7.1 Type II Hood Performance Test. A performance test shall be conducted upon the completion of—and before final approval of—installation of a ventilation system serving commercial cooking appliances. The test shall verify the rate of exhaust airflow required by Section 4.2. The permit holder shall furnish the necessary test equipment and devices required to perform the tests.

4.7.2 Type I Hood Capture and Containment Test. The permit holder shall verify the capture and containment performance of Type I hoods. A field test shall be conducted with all appliances under the hood at operating temperatures, all the hoods operating at design airflows, and with all sources of replacement air operating at design airflows for the restaurant. Capture and containment shall be verified visually by observing smoke or steam produced by actual cooking operation or by simulating cooking using devices such as smoke candles or smoke puffers. Smoke bombs shall not be used.

Informative Note: Smoke bombs typically create new effluent from a point source and do not necessarily show whether the cooking effluent is being captured. Actual cooking at the normal production rate is the most reliable method of generating smoke.

Table 2.7 Type I Hood Requirements by Appliance Type [Std 154-2016, Tbl 1]

Appliance Description	Size	Type I Hoods[a]			
		Light Duty	Medium Duty	Heavy Duty	Extra-Heavy Duty
Braising pan/tilting skillet, electric	All	•			
Oven, baking, electric and gas	All	•			
Oven, rotisserie, electric and gas	All	•			
Oven, combination, electric and gas	All	•			
Oven, convection, full-size, electric and gas	All	•			
Oven, convection, half-size, electric and gas (protein cooking)	All	•			
Oven, conveyor, electric	All	•			
Oven, deck, electric and gas	All	•			
Oven, duck, electric and gas	All	•			
Oven, revolving rack, electric and gas	All	•			
Oven, rapid cook, electric	All	•			
Oven, roasting, electric and gas	All	•			
Oven, rotisserie, electric and gas	All	•			
Oven, stone hearth, gas	All	•			
Range, cook-top, induction	All	•			
Range, discrete element, electric (with or without oven)	All	•			
Salamander, electric and gas	All	•			
Braising pan/tilting skillet, gas	All		•		
Broiler, chain conveyor, electric	All		•		
Broiler, electric, underfired	All		•		
Fryer, doughnut, electric and gas	All		•		
Fryer, kettle, electric and gas	All		•		
Fryer, open deep-fat, electric and gas	All		•		
Fryer, pressure, electric and gas	All		•		
Griddle, double-sided, electric and gas	All		•		
Griddle, flat, electric and gas	All		•		
Oven, conveyor, gas	All		•		
Range, open burner, gas (with or without oven)	All		•		
Range, hot top, electric and gas	All		•		
Smoker, electric and gas	All		•		
Broiler, chain conveyor, gas	All			•	
Broiler, electric and gas, over-fired (upright)	All			•	
Broiler, gas, underfired	All			•	
Grill, plancha, electric and gas	All			•	
Oven, tandoor, gas	All			•	
Range, wok, gas and electric	All			•	
Oven, stone hearth, wood-fired or wood for flavoring	All				•
Solid fuel cooking appliances combusting a solid fuel (such as wood, charcoal, or coal) to provide all or part of the heat for the cooking process[b]	All				•

[a]Where recirculating systems or recirculating hoods are used, the additional heat and moisture loads generated by such appliances shall be accounted for in the sensible and latent loads for the HVAC system.
[b]Solid-fuel flavoring cooking appliances shall comply with this table as if they do not combust solid fuel.

Table 2.8 Type II Hood Requirements by Appliance Description
[Std 154-2016, Tbl 2]

Appliance Description	Size	Hood Not Required[a]	Type II Hoods[b]	
			Light Duty	Medium Duty
Cabinet, holding, electric	All	•		
Cabinet, proofing, electric	All	•		
Cheese-melter, electric	All	•		
Coffee maker, electric	All	•		
Cooktop, induction, electric	All	•		
Dishwasher, door-type rack, hot water sanitizing, heat recovery and vapor reduction, electric	All	•		
Dishwasher, door-type rack, chemical sanitizing, heat recovery and vapor reduction, electric	All	•		
Dishwasher, door-type dump and fill, hot water sanitizing, electric	All	•		
Dishwasher, door-type dump and fill, chemical sanitizing, electric	All	•		
Dishwasher, pot and pan, hot water sanitizing, heat recovery and vapor reduction, electric	All	•		
Dishwasher, powered sink, electric	All	•		
Dishwasher, under-counter, chemical sanitizing, electric	All	•		
Dishwasher, under-counter, electric	All	•		
Dishwasher, undercounter, hot water sanitizing, heat recovery and vapor reduction, electric	All	•		
Drawer warmer, 2 drawer, electric	All	•		
Egg cooker, electric	All	•		
Espresso machine, electric	All	•		
Grill, panini, electric	All	•		
Hot dog cooker, electric	All	•		
Hot plate, countertop, electric	All	•		
Ovens, microwave, electric	All	•		
Popcorn machine, electric	All	•		
Rethermalizer, electric	All	•		
Rice cooker, electric	All	•		
Steam table, electric	All	•		
Steamers, bun, electric	All	•		
Steamer, compartment atmospheric, countertop, electric	All	•		
Steamer, compartment pressurized, countertop, electric	All	•		
Table, hot food, electric	All	•		
Toaster, electric	All	•		
Waffle iron, electric	All	•		
Kettle, steam jacketed, tabletop, electric, gas and direct steam	<20 gallons		•	
Oven, convection, half-size, electric and gas (non-protein cooking)	All		•	
Pasta cooker, electric	All		•	
Rethermalizer, gas	All		•	

Table 2.8 Type II Hood Requirements by Appliance Description
[Std 154-2016, Tbl 2] ***(Continued)***

Appliance Description	Size	Hood Not Required[a]	Type II Hoods[b]	
			Light Duty	Medium Duty
Rice cooker, gas	All		•	
Steamer, atmospheric, gas	All		•	
Steamer, pressurized, gas	All		•	
Steamer, atmospheric, floor-mounted, electric	All		•	
Steamer, pressurized, floor-mounted, electric	All		•	
Kettle, steam-jacketed floor mounted, electric, gas, and direct steam	<20 gal		•	
Dishwasher, conveyor rack, chemical sanitizing	All			•
Dishwasher, conveyor rack, hot water sanitizing	All			•
Dishwasher, door-type rack, chemical sanitizing	All			•
Dishwasher, door-type rack, hot water sanitizing	All			•
Dishwasher, pot and pan, hot water sanitizing	All			•
Pasta cooker, gas	All			•
Steam-jacketed kettle, floor mounted, electric and gas	≥20 gal			•

[a]Where hoods are not required, the additional heat and moisture loads generated by such appliances shall be accounted for in the sensible and latent loads for the HVAC system.
[b]Where recirculating systems or recirculating hoods are used, the additional heat and moisture loads generated by such appliances shall be accounted for in the sensible and latent loads for the HVAC system.

Table 2.9 Minimum Overhang Requirements for Type II Hoods
[Std 154-2016, Tbl 3]

Type of Hood	End Overhang, in.	Front Overhang, in.	Rear Overhang, in.
Wall-mounted canopy	6	12	N/A
Single-island canopy	12	12	12
Double-island canopy	12	12	N/A
Eyebrow	N/A	12	N/A
Backshelf/proximity/pass-over	6	10 (setback)	N/A

N/A = Not Applicable

Table 2.10 Type II Hood Minimum Net Exhaust Airflow Rates [Std 154-2016, Tbl 4]

Type of Hood	Minimum Net Exhaust Flow Rate per Linear Hood Length, cfm/ft	
	Light Duty Equipment	Medium Duty Equipment
Wall-mounted canopy	200	300
Single island	400	500
Double island	250	250
Eyebrow	250	250
Backshelf (pass-over)	200	300

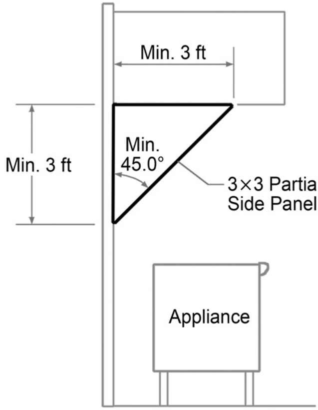


Figure 2.2 3 × 3 ft partial side panel [Std 154-2016, Fig 1]

4.8 Hood Clearance to Combustibles

4.8.1 Type I hoods shall be installed with a minimum 18 in. clearance to combustibles from any hood surface.

Exception: Type I hoods that are listed to reduced clearances in accordance with Standard UL710 or Standard UL 710B shall be installed at a minimum clearance to combustibles in accordance with their listings.

5. EXHAUST SYSTEMS

5.1 Duct Systems

5.1.1 Ducts serving Type I hoods shall be constructed of carbon steel of a minimum 16 gage thickness or stainless steel of a minimum 18 gage thickness. All seams, joints, and penetrations shall have a liquid-tight continuous external or internal weld. Internal welds shall be flush with the duct walls and accessible for inspection.

Exception: Factory-built ducts listed in accordance with UL 1978.

5.1.2 Ducts shall be constructed and installed so that grease cannot collect in any portion thereof, and ducts shall slope not less than one-fourth unit vertical in 12 units horizontal (2% slope) toward the hood or toward an approved grease reservoir. Where horizontal ducts exceed 75 ft (22.8 m) in length, the slope shall not be less than one unit vertical in 12 units horizontal (8.3% slope).

Exception: Listed factory-built ducts constructed of a round cross section shall be permitted to be installed at a reduced slope as allowed by their listing and the manufacturer's installation instructions.

5.1.3 Ducts shall not pass through firewalls unless enclosed in accordance with the applicable codes and standards.

5.1.4 Ducts shall lead to the exterior of the building.

5.1.5 A separate grease duct system shall be provided for each Type I hood. A separate grease duct system is not required where all of the following conditions are met:

a. All interconnected hoods are located within the same story.
b. All interconnected hoods are located within the same room or in adjoining rooms.
c. Interconnecting ducts do not penetrate fire barriers.
d. The grease duct system does not serve solid fuel-fired appliance(s).

5.1.6 Ducts shall be installed without forming dips or traps that might collect grease, except where unavoidable. In such situations, the duct section having a dip or trap shall be provided with drain access for regular cleanout.

Informative Note: For other duct construction and installation details, such as welded duct connections, access openings for inspection and maintenance, clearance to combustible material, interior installation (including fire-rated enclosures and the clearance between the duct and interior surface of the enclosures), exterior installation, and exhaust system termination on the roof or at a wall, refer to NFPA 96, the *International Mechanical Code*, or local codes.

5.1.7 Ducts with field-applied insulation listed in accordance with ASTM E 2336, *Standard Test Methods for Fire Resistive Grease Duct Enclosure Systems*, and factory-built ducts with integral insulation listed in accordance with UL 2221, *Tests of Fire Resistive Grease Duct Enclosure Assemblies*, are acceptable, where included in NFPA 96, the *International Mechanical Code*, or the *Uniform Mechanical Code* for use as an alternative to a duct and fire-resistance-rated shaft enclosure around the duct.

5.2 Duct Leakage Testing

5.2.1 Prior to the use or concealment of any portion of a grease duct system, a leakage test shall be performed to determine that all welded joints and seams are liquid tight. The leakage test shall consist of a light test, water pressure test, or an approved equivalent test. The permit holder shall be responsible for providing the necessary equipment and for performing the test.

5.2.1.1 Light Test. The light test shall be performed by passing a lamp having a power rating of not less than 100 W through the entire section of ductwork to be tested. The lamp shall be open so as to emit light equally in all directions perpendicular to the duct walls. No light from the duct interior shall be visible through any exterior surface.

5.2.1.2 Water Test. The water test shall be performed by use of a pressure washer operating at a minimum of 1500 psi, simulating cleaning operations. The water shall be applied directly to all areas to be tested. No water applied to the duct interior shall be visible on any exterior surface in any volume during the test.

5.3 Airflow Performance

5.3.1 The velocity in the duct shall be at least 500 fpm.

Informative Note: This standard does not limit the airflow velocity by specifying a maximum velocity, but due to typical spatial and cost constraints, general design duct velocities between 1500 and 1800 fpm are often used when designing for maximum airflows. Duct velocities greater than 2500 fpm can cause unwanted duct pressure and noise levels.

5.3.2 Lower exhaust airflow than that required for full-load cooking conditions is permitted during no-load cooking conditions, where engineered controls or listed multispeed or variable-speed controls automatically operate the exhaust system to maintain capture and removal of cooking effluents.

5.4 Fans

5.4.1 Fans shall be of sufficient capacity to provide the required airflow against the system's resistance. Expected air temperatures, altitude, windage, and system effects shall be taken into account when determining fan capacity. Fan air performance shall be tested and certified according to AMCA Standard 210.

Informative Note: Belt-drive fans and adjustable-drive sheaves provide a means of adjusting the fan speed for final system balancing. A variable-speed controller allows a broader range of speed adjustability.

5.4.2 Exhaust fans (up-blast, in-line, or utility-set fans) serving Type I hoods shall be capable of handling hot, grease-laden air and flare-up conditions. Fans shall be designed to contain and properly drain grease removed from the airstream. The fan housing or scroll that contains the grease shall be fully welded so that it is liquid tight. The fan impeller shall be a self-cleaning design.

Exception: Fans that are listed to UL 705, *Standard for Power Ventilators*, and UL 762, *Outline of Investigation for Power Roof Ventilators for Restaurant Exhaust Applications*.

5.4.3 Up-blast fans shall be hinged with tip-over restraints and have a flexible weatherproof electrical cable to permit inspection and cleaning. Utility-set exhaust fans shall be provided with access panels for inspection and cleaning.

5.4.4 Access shall be provided for cleaning the fan wheel. The access opening shall be a minimum of 3 × 5 in. or have a circular diameter of at least 4 in. on the curvature of the outer fan housing. Fan drive assemblies shall be separated from the airstream. Covers shall be provided with motor weather protection for outdoor installation and belt guards for indoor applications.

5.4.5 The ductwork extending to up-blast fans shall extend a minimum of 18 in. above the roof surface.

5.5 Other Equipment

5.5.1 Thermal recovery units, air pollution control devices, or other devices can be used in the exhaust systems when specifically approved for such use except where prohibited. Refer to Section 514.2 of the *International Mechanical Code*, for prohibited applications.

5.5.2 Clearance, installation, and fire-extinguishing system requirements shall comply with applicable codes and standards.

5.5.3 Pollution control units not equipped with electrostatic precipitators shall be listed in accordance with the applicable requirements of UL710 and UL1978. Pollution control units equipped with electrostatic precipitators shall be listed in accordance with UL867 and the applicable requirements of UL710 and UL1978.

5.6 Exhaust Discharge

5.6.1 Exhaust systems shall be designed to prevent re-entrainment into building intakes. Prevailing winds and velocities shall be considered when locating intake and exhaust openings. The minimum horizontal distance between discharge and intake shall be 10 ft. Where this horizontal distance is not achievable, the exhaust shall discharge a minimum of 2 ft above any outdoor air. Exhaust discharge shall not impinge on overhangs, parapets, other equipment, or higher parts of buildings.

Informative Note: Refer to Chapter 16 of the 2013 *ASHRAE Handbook—Fundamentals* for airflow patterns around buildings.

5.6.2 Exhaust airstreams for Type I hoods shall be located a minimum of 40 in.above the finished roof surface and be directed away from roof and building surfaces.

5.6.3 Additional protection for roofing material at the exhaust discharge of a Type I hood shall be provided to prevent material degradation or failure.

5.7 Operation and Maintenance

5.7.1 Appliance Interlock

5.7.1.1 The exhaust fan serving a Type I hood shall have automatic controls that will activate the fan when any appliance that requires such Type I hood is turned on, or a means of interlock shall be provided that will prevent operation of such appliances when the exhaust fan is not turned on.

5.7.1.2 Where one or more temperature or energy sensors are used to activate a Type I hood exhaust fan, the fan shall activate not more than 15-minutes after the first appliance served by that hood has been turned on. A method of interlock between an exhaust hood system and appliances equipped with standing pilot burners shall not cause the pilot burners to be extinguished. A method of interlock between an exhaust hood system and cooking appliances shall not involve or depend on any component of a fire extinguishing system.

5.7.2 The entire exhaust system shall be inspected at regular intervals for grease buildup by a properly trained, qualified, and certified company or person(s) acceptable to the authority having jurisdiction.

5.7.2.1 The schedule of inspection for grease buildup in the exhaust system and cleaning of the exhaust system shall comply with NFPA 96.

5.7.2.2 Upon inspection, if the exhaust system is found to be contaminated with grease deposits, the contaminated portions of the exhaust system shall be thoroughly cleaned by a properly trained, qualified, and certified company or person(s) acceptable to the authority having jurisdiction.

5.7.3 Inspection and maintenance of thermal recovery units, air pollution control devices, or other devices shall be conducted by properly trained and qualified persons at a frequency specified in the manufacturer's instructions or equipment listing.

6. REPLACEMENT AIR

6.1 Air Introduction

6.1.1 The terminal velocity of air introduced from devices in the kitchen shall not exceed 50 fpm at the lowest edges of the hood.

Informative Note:

1. Using perforated ceiling or perimeter diffusers generally results in a lower terminal velocity at the lower edge of the hood than directional ceiling diffusers.
2. Best practice is to bring conditioned air into the kitchen away from the hood and distribute it throughout the kitchen to improve worker productivity and comfort as well as to lower hood exhaust rates.

6.1.2 Transfer air from dining or other areas that passes through openings such as windows or walkways shall be sized for air velocities not to exceed 75 fpm based on the free area of the opening. Openings provided for transfer air shall remain open during system operation.

Informative Note: Such openings should be arranged to avoid creating drafts on personnel. Consideration should be given to minimizing air velocity when openings are used as pass-through openings for prepared food.

6.2 Air Balance

6.2.1 Design plans for a facility with a commercial kitchen ventilation system shall include a table or diagram indicating the design outdoor air balance (see Informative Annex A, Section A1). The design outdoor air balance shall indicate all exhaust and replacement air for the facility, plus the net exfiltration if applicable. The total replacement air airflow rate shall equal the total exhaust airflow rate plus the net exfiltration. It is permissible to supply replacement air to the kitchen space by using transfer air from areas other than the kitchen.

Informative Note: Although individual replacement air sources are not required to be 100% outdoor air, sufficient outdoor air must be introduced into the system to compensate for each exhaust and exfiltration component. For example, for 100 cfm transfer air from room A to room B to qualify as replacement air, at least 100 cfm outdoor air must be provided to room A (e.g., as outdoor air to an environmental air system serving room A, infiltration to room A, or transfer air from another room).

6.2.2 Operation of systems where airflows can vary (including but not limited to HVAC systems incorporating variable air volume, systems with outdoor air economizer control, or exhaust systems with variable airflow) shall be controlled to comply with the requirements of this standard over the full range of anticipated airflows. Additional air balance diagrams or tables shall be provided as necessary to indicate compliance over the full range of anticipated airflow.

6.2.3 Where the design air balance relies on transfer air from a source beyond the facility's control (e.g., air drawn into an individual tenant's facility from the common areas of a shopping mall), this source shall be identified.

6.3 Pressure Differentials

6.3.1 The commercial kitchen ventilation system shall be designed to establish pressure differentials to control odor migration and to control dust, dirt, and insects in accordance with the criteria in the following subsections.

6.3.1.1 The kitchen of a food-service facility shall be maintained under a negative pressure with respect to dining areas and adjacent nonfood areas. The maximum negative pressure shall not exceed 0.02 in. of water.

6.3.1.2 A freestanding food-service facility (i.e., a food-service facility that entirely occupies a single building) shall be maintained under a positive pressure with respect to outdoors.

Exception: Where migration of food odors to adjacent interior rooms within the same tenancy would not be objectionable. Display cooking under a hood located in the dining area is not considered a kitchen.

6.3.2 Where a food-service facility shares a wall with an adjacent non-food-service facility, such as a retail center or a shopping mall, the food-service facility shall be maintained under a negative pressure with respect to outdoors and the adjacent spaces.

Exceptions:

1. Where the separation between the food service facility and the adjacent interior room is sealed substantially airtight to prevent odor migration.
2. In shopping malls and other occupancies where a food-service facility is open to another tenancy or to the mall common area, the food-service facility shall be permitted to be under a negative pressure with respect to the non-food-service occupancy.

6.3.3 The pressure in any room in which a draft-hood vented appliance, such as a gas water heater, is located shall be maintained not less than 0.02 in. of water below outdoor ambient pressure.

7. SYSTEM CONTROLS

7.1 Operating Controls

7.1.1 Replacement air systems shall be interlocked to ensure operation upon activation of the exhaust system.

7.1.2 Demand-Control Ventilation

7.1.2.1 The exhaust flow rate is permitted to be reduced during partial load cooking and when there is no cooking through the means of demand-control ventilation.

7.1.2.2 Exhaust rates shall maintain capture and containment of appliance flue gases and cooking effluent during full-load, partial, or idle operating conditions.

7.1.2.3 During periods of reduced exhaust airflow, replacement air shall be automatically controlled to maintain the building pressure differentials in accordance with Section 6.3.

Informative Note: Replacement air units may have minimum airflow requirements for safe or effective operation of heating and/or cooling/dehumidification functions. Demand-control ventilation systems' minimum airflow settings must not be set lower than the replacement air systems minimum operating airflow.

7.1.2.4 Demand-control ventilation systems shall be part of a listed hood, shall be listed for the purpose, or shall be engineered.

Fire Suppression

Exhaust systems serving grease-producing equipment must include a fire-extinguishing system unless listed grease removal devices are installed. Wet chemical systems with nozzles over cooking equipment, in the hood and at the duct collar downstream of hood are commonly used, per NFPA 17A. Water from wet-pipe sprinkler systems can be used, per NFPA 13.

3. WATER

Table 3.1 Common Pump Terms, Symbols, and Formulas

Term	Symbol	Units	Formula
Velocity	v	ft/s	
Volume	V	ft^3	
Flow rate	Q_v	gpm	
Pressure	p	psi	
Density	ρ	lb/ft^3	
Acceleration of gravity	g	32.17 ft/s^2	
Speed	n	rpm	
Specific gravity	SG	—	$= \frac{\text{Mass of liquid}}{\text{Mass of water at 39°F}}$
Head	H	ft	$2.31\, p/\text{SG}$
Net positive suction head (NPSH)	H	ft	
Efficiency (percent)			
Pump	η_p		
Electric motor	η_m		
Variable speed drive	η_v		
Equipment (constant-speed pumps)	η_e		$\eta_e = \eta_p \eta_m / 100$
Equipment (variable-speed pumps)	η_e		$\eta_e = 10^{-4} \eta_p \eta_m \eta_v$
Utilization	$\eta_{\boldsymbol{u}}$		
Q_D = design flow Q_A = actual flow H_D = design head H_A = actual head			$\eta_u = 100 \frac{Q_D H_D}{Q_A H_A}$
System Efficiency Index (decimal)			$\text{SEI} = 10^{-4} \eta_\varepsilon \eta_{\boldsymbol{u}}$
Output power (pump)	P_o	hp	$Q_v \text{HSG}/3960$
Shaft power	P_s	hp	$100 P_o / \eta_p$
Input power	P_i	kW	$74.6 P_s / \eta_m$

Table 3.2 Affinity Laws for Pumps

Impeller Diameter	Speed	Specific Gravity (SG)	To Correct for	Multiply by
Constant	Variable	Constant	Flow	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)$
			Head	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^2$
			Power	$\left(\frac{\text{New Speed}}{\text{Old Speed}}\right)^3$
Variable	Constant	Constant	Flow	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)$
			Head	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^2$
			Power	$\left(\frac{\text{New Diameter}}{\text{Old Diameter}}\right)^3$
Constant	Constant	Variable	Power	$\left(\frac{\text{New SG}}{\text{Old SG}}\right)$

$$\text{pump hp} = \frac{\text{gpm} \times \text{ft head} \times \text{sp. gr.}}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

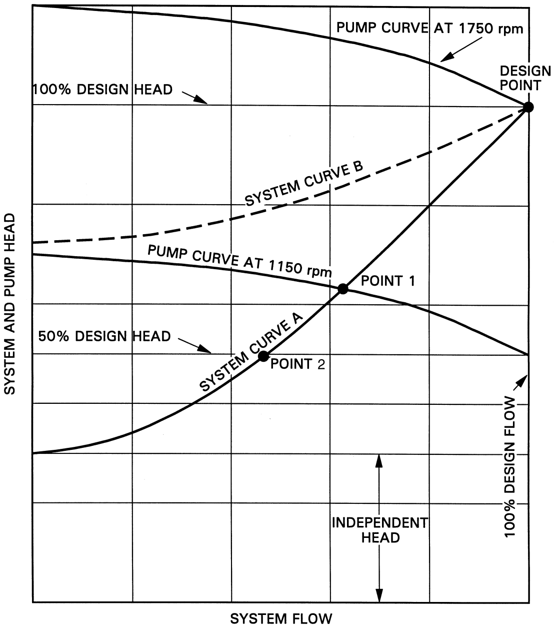


Figure 3.1 Pump Curves and System Curves

If the hydronic system has a system head curve as shown in curve A, the pump at 1150 rpm will operate at point 1, not at point 2, as would be predicted by the affinity laws alone. If the hydronic system has a system head curve like curve B of Figure 3.1, the pump at 1150 rpm will run at shutoff head and deliver no water. This demonstrates that the affinity laws should be used to develop new pump head/capacity curves, but not to predict performance with a particular hydronic system unless its system head curve is known.

Net Positive Suction Characteristics

Particular attention must be given to the pressure and temperature of the water as it enters the pump, especially in condenser towers, steam condensate returns, and steam boiler feeds.

The pressure in excess of that required to prevent vapor pockets from forming is the net positive suction head required (NPSHR). NPSHR is a characteristic of a given pump and varies with pump speed and flow. It is determined by the manufacturer and is included on the pump performance curve.

If the absolute pressure at the suction nozzle approaches the vapor pressure of the liquid, vapor pockets form in the impeller passages. The collapse of the vapor pockets (cavitation) is noisy and can be destructive to the pump impeller.

NPSHR is particularly important when a pump is operating with hot liquids or is applied to a circuit having a suction lift. The vapor pressure increases with water temperature and reduces the net positive suction head available (NPSHA). Each pump has its NPSHR, and the installation has its NPSHA, which is the total useful energy above the vapor pressure at the pump inlet.

$$\text{NPSHA} = h_p + h_z - h_{vpa} - h_f \tag{3.1}$$

where

h_p	=	absolute pressure on surface of liquid that enters pump, ft of head
h_z	=	static elevation of liquid above center line of pump (h_z is negative if liquid level is below pump center line), ft
h_{vpa}	=	absolute vapor pressure at pumping temperature, ft
h_f	=	friction and head losses in suction piping, ft

To determine the NPSHA in an existing installation, Equation 3.2 may be used (see Figure 3.2):

$$\text{NPSHA} = h_a + h_s + \frac{V^2}{2g} - h_{vpa} \tag{3.2}$$

where

h_a	=	atmospheric head for elevation of installation, ft
h_s	=	head at inlet flange corrected to center line of pump (h_s is negative if below atmospheric pressure), ft
$V^2/2g$	=	velocity head at point of measurement of h_s, ft

For trouble-free design, the NPSHA must always be greater than the pump's NPSHR. In closed hot- and chilled-water systems where sufficient system fill pressure is exerted on the pump suction, NPSHR is normally not a factor.

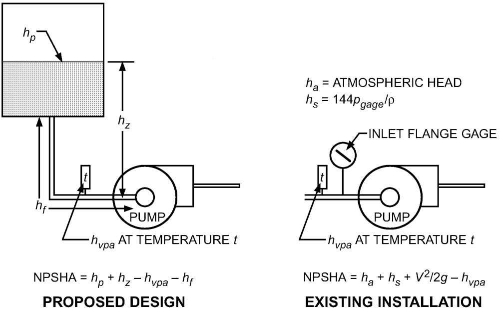


Figure 3.2 Net Positive Suction Head Available [2016S, Ch 44, Fig 33]

Water

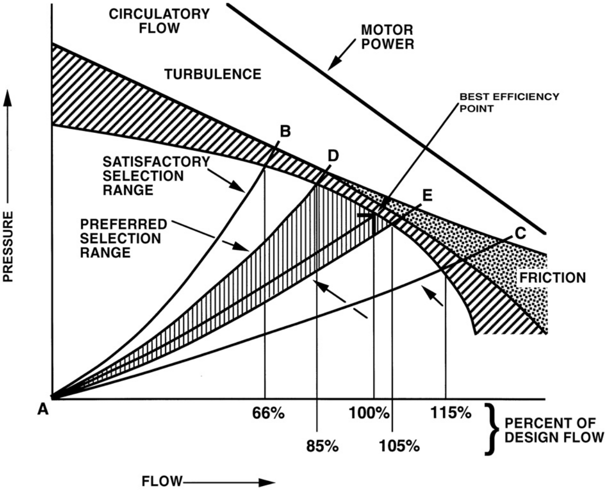


Figure 3.3 Pump Selection Regions [2016S, Ch 44, Fig 35]

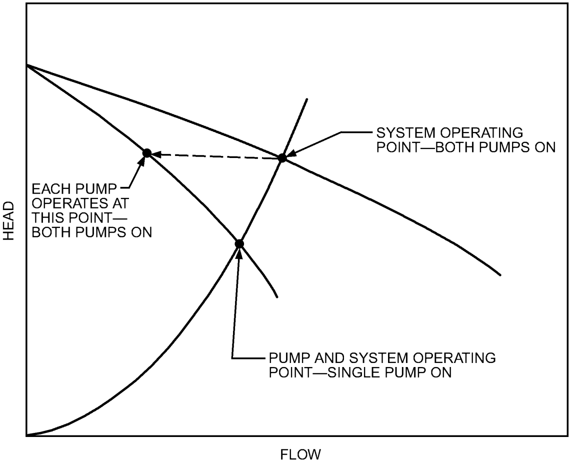


Figure 3.4 Operating Conditions for Parallel Operation [2062S, Ch 44, Fig 37]

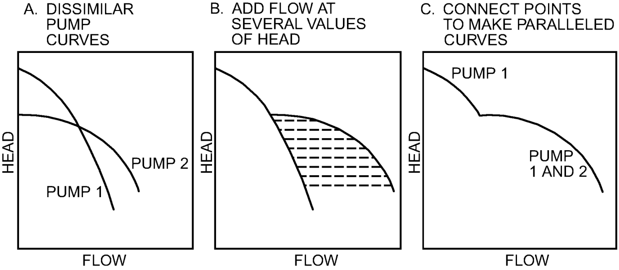


Figure 3.5 Construction of Curve for Dissimilar Parallel Pumps [2016S, Ch 44, Fig 38]

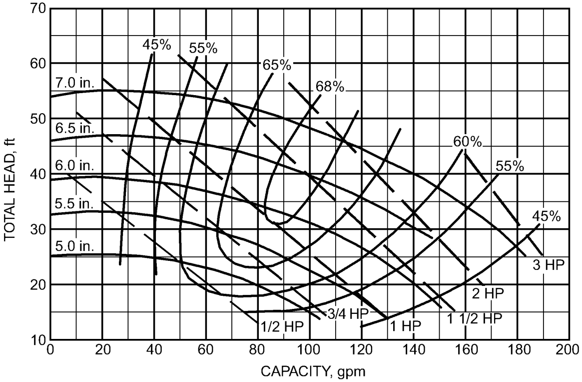


Figure 3.6 Typical Pump Curves (Curves Vary by Manufacturer) [2016S, Ch 44, Fig 13]

Table 3.3 General Information on Water

Specific gravity of water is usually given as 1.0 at 60°F. However, for some purposes it is given as 1.0 at 39.2°F, the point of maximum density. Based on water at 39.2°F as 1.0, water at 60°F has a specific gravity of 0.999. Therefore, which base is selected makes no practical difference.

Viscosity of water varies as follows:

	32°F	50°F	60°F	70°F	80°F	100°F	120°F	140°F	160°F	180°F	212°F
Absolute viscosity, centipoises	1.70	1.31	1.12	.98	.86	.68	.56	.47	.40	.35	.28
Kinematic viscosity, centistokes	1.79	1.31	1.12	.98	.86	.69	.57	.45	.41	.36	.29

Table 3.4 Weight and Volume Equivalents

Convert from	Convert to								
	U.S. Gallon	Imperial Gallon	Cubic Inch	Cubic Foot	* Pound	* Cwt (U.S.)	* Ton (U.S.)	Litre	Cubic Metre
U.S. Gallon	1.	.8327	231.	.13368	8.345	.08345	.00418	3.785	.00378
Imperial Gallon	1.201	1.	22.741	.1605	10.02	.1002	.00502	4.546	.00455
Cubic Inch	.004329	.003607	1.	.000579	.036124	–	–	.0164	–
Cubic Foot	7.4805	6.229	1728.	1.	62.425	.6243	.03121	28.317	.0283
Pound[a]	.1198	.0998	27.68	.01602	1.	.01	.0005	.454	-
Cwt (U.S.)*	11.98	9.98	2765.	1.602	100.	1.	.05	45.36	.045
Ton (U.S.)*	239.6	199.6	–	32.04	2000.	20.0	1.	906.9	.907
Litre	.2642	.22	61.023	.0353	2.205	.022	.0011	1.	.001
Cubic Metre	264.2	220.	–	35.314	2204.5	22.045	1.102	1000.	1.

a. Volume—weight relationship taken for water at greatest density (39.2°F).

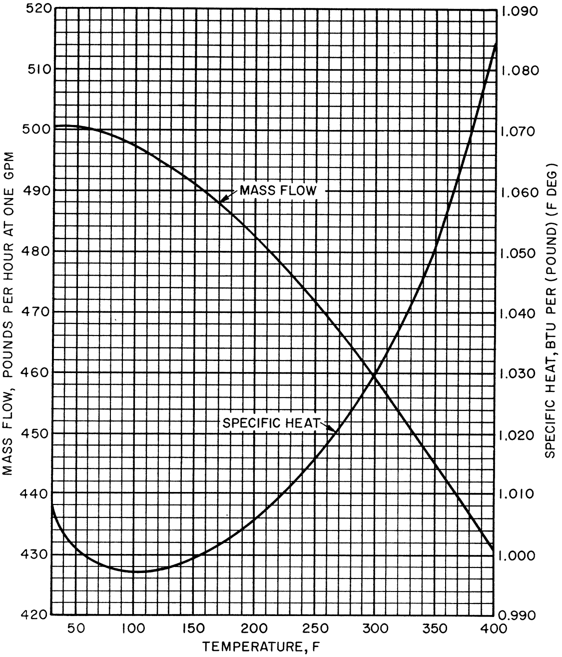


Figure 3.7 Mass Flow and Specific Heat of Water

Table 3.5 Freezing Points for Solutions of Ethylene Glycol and Propylene Glycol

Glycol, % by mass	Ethylene Glycol °F	Propylene Glycol °F
10	26.2	26.1
15	22.2	22.9
20	17.9	19.2
25	12.7	14.7
30	6.7	9.2
40	–8.1	–6.0
50	–28.9	–28.3
60	–54.8	–59.9

Ethylene glycol solutions are less viscous than propylene glycol solutions at the same concentration. Less toxic propylene glycol is preferred for applications involving possible human contact.

Table 3.6 Volume of Vertical Cylindrical Tanks in Gallons per Foot of Depth

Diameter in ft	in.	U.S. Gallons	Diameter in ft	in.	U.S. Gallons	Diameter in ft	in.	U.S. Gallons
1	0	5.875	3	6	71.97	6	0	211.5
1	2	7.997	3	8	78.99	6	6	248.2
1	4	10.44	3	10	86.33	7	0	287.9
1	6	13.22	4	0	94.00	7	6	330.5
1	8	16.32	4	2	102.0	8	0	376.0
1	10	19.75	4	4	110.3	8	6	424.5
2	0	23.50	4	6	119.0	9	0	475.9
2	2	27.58	4	8	127.9	9	6	530.2
2	4	31.99	4	10	137.3	10	0	587.5
2	6	36.72	5	0	146.9	10	6	647.7
2	8	41.78	5	2	156.8	11	0	710.9
2	10	47.16	5	4	167.1	11	6	777.0
3	0	52.88	5	6	177.7	12	0	846.0
3	2	58.92	5	8	188.7	12	6	918.0
3	4	65.28	5	10	199.9			

Water

Table 3.7 Quantities for Various Depths of Vertical Cylindrical Tanks in Horizontal Position

% Depth Filled	% of Capacity	% Depth Filled	% of Capacity	% Depth Filled	% of Capacity	% Depth Filled	% of Capacity
1	.20	26	20.73	51	51.27	76	81.50
3	.90	28	23.00	53	53.81	78	83.68
5	1.87	30	25.31	55	56.34	80	85.77
7	3.07	32	27.66	57	58.86	82	87.76
9	4.45	34	30.03	59	61.36	84	89.68
11	5.98	36	32.44	61	63.86	86	91.50
13	7.64	38	34.90	63	66.34	88	93.20
15	9.40	40	37.36	65	68.81	90	94.80
17	11.27	42	39.89	67	71.16	92	96.26
19	13.23	44	42.40	69	73.52	94	97.55
21	15.26	46	44.92	71	75.93	96	98.66
23	17.40	48	47.45	73	78.14	98	99.50
25	19.61	50	50.00	75	80.39	100	100.0

Table 3.8 Volume of Water in Standard Pipe and Tube

Nominal Pipe Size, in.	Schedule No.	Standard Steel Pipe Inside Diameter, in.	Standard Steel Pipe Volume, gal/ft	Type *L* Copper Tube Inside Diameter, in.	Type *L* Copper Tube Volume, gal/ft
3/8	—	—	—	0.430	0.0075
1/2	40	0.622	0.0157	0.545	0.0121
5/8	—	—	—	0.666	0.0181
3/4	40	0.824	0.0277	0.785	0.0251
1	40	1.049	0.0449	1.025	0.0429
1 1/4	40	1.380	0.0779	1.265	0.0653
1 1/2	40	1.610	0.106	1.505	0.0924
2	40	2.067	0.174	1.985	0.161
2 1/2	40	2.469	0.249	2.465	0.248
3	40	3.068	0.384	2.945	0.354
3 1/2	40	3.548	0.514	3.425	0.479
4	40	4.026	0.661	3.905	0.622
5	40	5.047	1.04	4.875	0.970
6	40	6.065	1.50	5.845	1.39
8	30	8.071	2.66	7.725	2.43
10	30	10.136	4.19	9.625	3.78
12	30	12.090	5.96	11.565	5.46

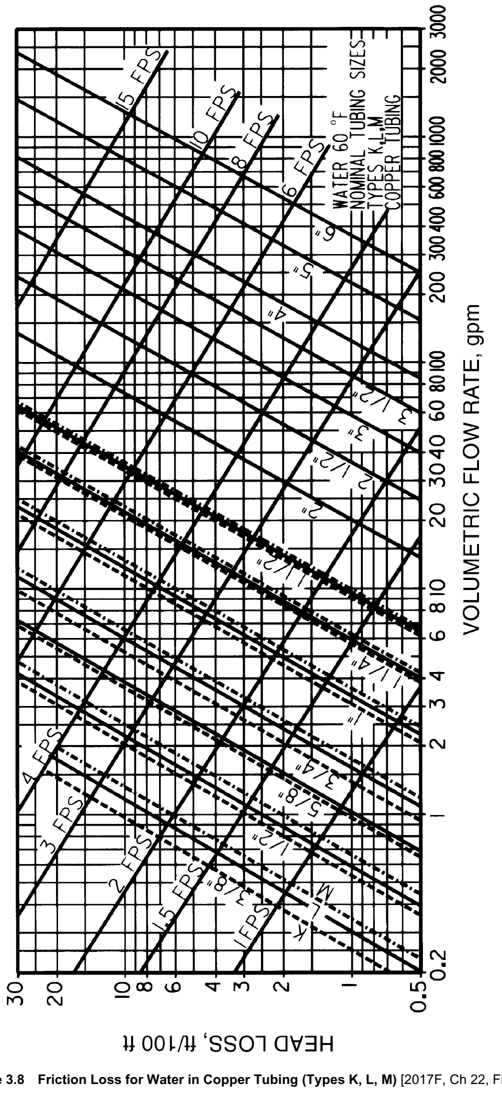


Figure 3.8 Friction Loss for Water in Copper Tubing (Types K, L, M) [2017F, Ch 22, Fig 15]

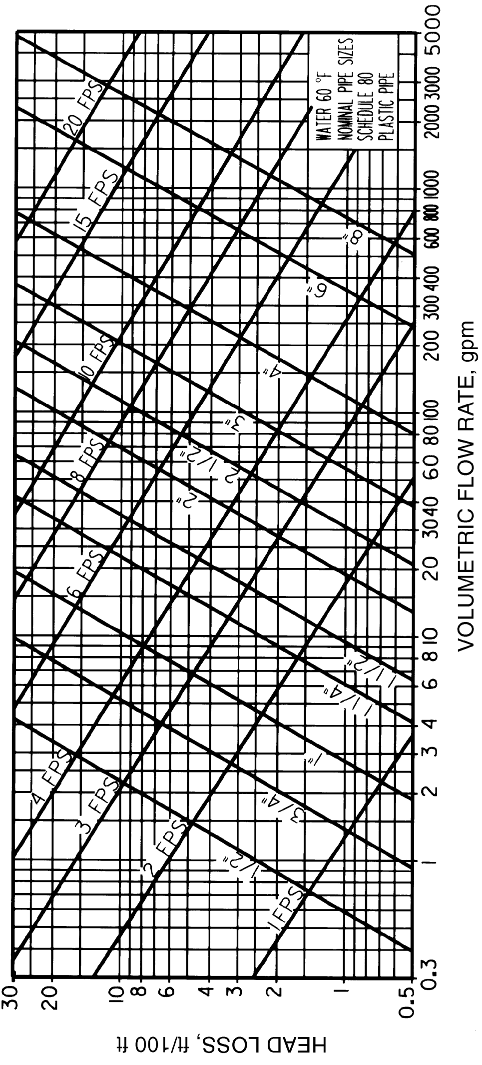


Figure 3.9 Friction Loss for Water in Plastic Pipe (Schedule 80) [2017F, Ch 22, Fig 16]

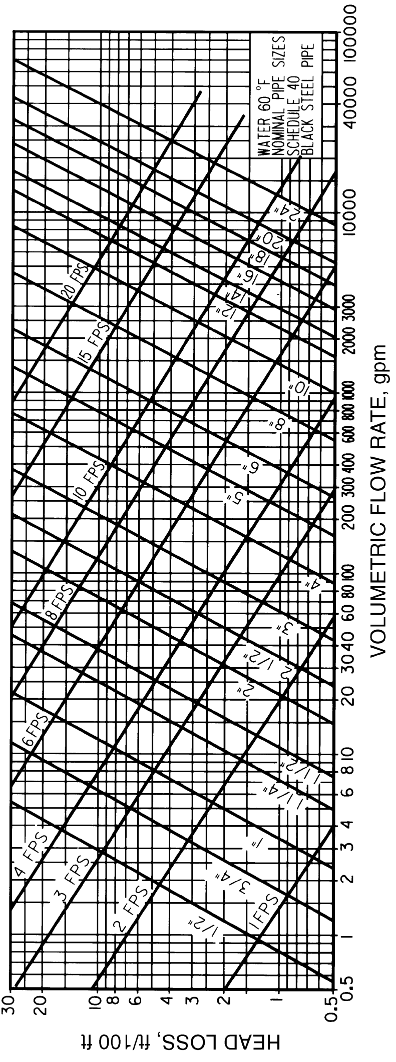


Figure 3.10 Friction Loss for Water in Commercial Steel Pipe (Schedule 40) [2017F, Ch 22, Fig 14]

Valve and Fitting Losses

Valves and fittings cause pressure losses greater than those caused by the pipe alone. One formulation expresses losses as

$$\Delta p = K\left(\frac{\rho}{g_c}\right)\left(\frac{V^2}{2}\right) \text{ or } \Delta h = K\frac{V^2}{2g} \tag{3.3}$$

where K = geometry- and size-dependent loss coefficient (see Tables 3.9 to 3.14).

ASHRAE research project RP-1193 found the data in Tables 3.9 to 3.14 giving K factors for Schedule 80 PVC 2, 4, 6, and 8 in. ells, reducers, expansions, and tees. In general, PVC fitting geometry varied much more from one manufacturer to another than steel fittings did.

Calculating Pressure Losses

The most common engineering design flow loss calculation selects a pipe size for the desired total flow rate and available or allowable pressure drop.

Because either formulation of fitting losses requires a known diameter, pipe size must be selected before calculating the detailed influence of fittings. A frequently used rule of thumb assumes that the design length of pipe is 50 to 100% longer than actual to account for fitting losses. After a pipe diameter has been selected on this basis, the influence of each fitting can be evaluated.

Table 3.9 *K* Factors: Threaded Pipe Fittings [2017F, Ch 22, Tbl 3]

Nominal Pipe Dia., in.	90° Standard Elbow	90° Long-Radius Elbow	45° Elbow	Return Bend	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve	Bell Mouth Inlet	Square Inlet	Projected Inlet
3/8	2.5	—	0.38	2.5	0.90	2.7	20	0.40	—	8.0	0.05	0.5	1.0
1/2	2.1	—	0.37	2.1	0.90	2.4	14	0.33	—	5.5	0.05	0.5	1.0
3/4	1.7	0.92	0.35	1.7	0.90	2.1	10	0.28	6.1	3.7	0.05	0.5	1.0
1	1.5	0.78	0.34	1.5	0.90	1.8	9	0.24	4.6	3.0	0.05	0.5	1.0
1 1/4	1.3	0.65	0.33	1.3	0.90	1.7	8.5	0.22	3.6	2.7	0.05	0.5	1.0
1 1/2	1.2	0.54	0.32	1.2	0.90	1.6	8	0.19	2.9	2.5	0.05	0.5	1.0
2	1.0	0.42	0.31	1.0	0.90	1.4	7	0.17	2.1	2.3	0.05	0.5	1.0
2 1/2	0.85	0.35	0.30	0.85	0.90	1.3	6.5	0.16	1.6	2.2	0.05	0.5	1.0
3	0.80	0.31	0.29	0.80	0.90	1.2	6	0.14	1.3	2.1	0.05	0.5	1.0
4	0.70	0.24	0.28	0.70	0.90	1.1	5.7	0.12	1.0	2.0	0.05	0.5	1.0

Source: *Engineering Data Book* (Hydraulic Institute 1990).

Table 3.10 *K* Factors: Flanged Welded Pipe Fittings [2017F, Ch 22, Tbl 4]

Nominal Pipe Dia., in.	90° Standard Elbow	90° Long-Radius Elbow	45° Long-Radius Elbow	Return Bend Standard	Return Bend Long-Radius	Tee-Line	Tee-Branch	Globe Valve	Gate Valve	Angle Valve	Swing Check Valve
1	0.43	0.41	0.22	0.43	0.43	0.26	1.0	13	—	4.8	2.0
1 1/4	0.41	0.37	0.22	0.41	0.38	0.25	0.95	12	—	3.7	2.0
1 1/2	0.40	0.35	0.21	0.40	0.35	0.23	0.90	10	—	3.0	2.0
2	0.38	0.30	0.20	0.38	0.30	0.20	0.84	9	0.34	2.5	2.0
2 1/2	0.35	0.28	0.19	0.35	0.27	0.18	0.79	8	0.27	2.3	2.0
3	0.34	0.25	0.18	0.34	0.25	0.17	0.76	7	0.22	2.2	2.0
4	0.31	0.22	0.18	0.31	0.22	0.15	0.70	6.5	0.16	2.1	2.0
6	0.29	0.18	0.17	0.29	0.18	0.12	0.62	6	0.10	2.1	2.0
8	0.27	0.16	0.17	0.27	0.15	0.10	0.58	5.7	0.08	2.1	2.0
10	0.25	0.14	0.16	0.25	0.14	0.09	0.53	5.7	0.06	2.1	2.0
12	0.24	0.13	0.16	0.24	0.13	0.08	0.50	5.7	0.05	2.1	2.0

Source: *Engineering Data Book* (Hydraulic Institute 1990).

Table 3.11 Summary of *K* Factors for Reducers and Expansions [2017F, Ch 22, Tbl 6]

		ASHRAE Research[a,b]		
		4 fps	8 fps	12 fps
Reducer	(2 by 1.5 in.) thread	0.53	0.28	0.20
	(4 by 3 in.) weld	0.23	0.14	0.10
	(6 by 4 in.) weld	0.62	0.54	0.53
	(8 by 6 in.) weld	0.31	0.28	0.26
	(10 by 8 in.) weld	0.16	0.14	0.14
	(12 by 10 in.) weld	0.14	0.14	0.14
	(16 by 12 in.) weld	0.17	0.16	0.17
	(20 by 16 in.) weld	0.16	0.13	0.13
	(24 by 20 in.) weld	0.053	0.053	0.055
Expansion	(1.5 by 2 in.) thread	0.16	0.13	0.02
	(3 by 4 in.) weld	0.11	0.11	0.11
	(4 by 6 in.) weld	0.28	0.28	0.29
	(6 by 8 in.) weld	0.15	0.12	0.11
	(8 by 10 in.) weld	0.11	0.09	0.08
	(10 by 12 in.) weld	0.11	0.11	0.11
	(12 by 16 in.) weld	0.073	0.076	0.073
	(16 by 20 in.) weld	0.024	0.021	0.022
	(20 by 24 in.) weld	0.020	0.023	0.020

Source: Rahmeyer (2003a). [a]Rahmeyer (1999a, 2002a). [b]Ding et al. (2005)

Table 3.12 Summary of *K* Factors for Pipe Tees [2017F, Ch 22, Tbl 7]

		ASHRAE Research[a,b]		
		4 fps	8 fps	12 fps
2 in. thread tee,	100% branch	0.93	—	—
	100% line (flow-through)	0.19	—	—
	100% mix	1.19	—	
4 in.weld tee,	100% branch	—	0.57	—
	100% line (flow-through)	—	0.06	—
	100% mix	—	0.49	—
6 in.weld tee,	100% branch	—	0.56	—
	100% line (flow-through)	—	0.12	—
	100% mix	—	0.88	—
8 in.weld tee,	100% branch	—	0.53	—
	100% line (flow-through)	—	0.08	—
	100% mix	—	0.70	—
10 in.weld tee,	100% branch	—	0.52	—
	100% line (flow-through)	—	0.06	—
	100% mix	—	0.77	—
12 in.weld tee,	100% branch	0.70	0.63	0.62
	100% line (flow-through)	0.062	0.091	0.096
	100% mix	0.88	0.72	0.72
16 in.weld tee,	100% branch	0.54	0.55	0.54
	100% line (flow-through)	0.032	0.028	0.028
	100% mix	0.74	0.74	0.76

[a]Rahmeyer (1999b, 2002b). [b]Ding et al. (2005).

Table 3.13 Test Summary for Loss Coefficients *K* and Equivalent Loss Lengths [2017F, Ch 22, Tbl 8]

Schedule 80 PVC Fitting		*K*	*L*, ft
Injected molded elbow,	2 in.	0.91 to 1.00	8.4 to 9.2
	4 in.	0.86 to 0.91	18.3 to 19.3
	6 in.	0.76 to 0.91	26.2 to 31.3
	8 in.	0.68 to 0.87	32.9 to 42.1
8 in. fabricated elbow, Type I, components		0.40 to 0.42	19.4 to 20.3
Type II, mitered		0.073 to 0.76	35.3 to 36.8
6 by 4 in. injected molded reducer		0.12 to 0.59	4.1 to 20.3
Bushing type		0.49 to 0.59	16.9 to 20.3
8 by 6 in. injected molded reducer		0.13 to 0.63	6.3 to 30.5
Bushing type		0.48 to 0.68	23.2 to 32.9
Gradual reducer type		0.21	10.2
4 by 6 in. injected molded expansion		0.069 to 1.19	1.5 to 25.3
Bushing type		0.069 to 1.14	1.5 to 24.2
6 by 8 in. injected molded expansion		0.95 to 0.96	32.7 to 33.0
Bushing type		0.94 to 0.95	32.4 to 32.7
Gradual reducer type		0.99	34.1

Table 3.14 Test Summary for Loss Coefficients *K* of PVC Tees [2017F, Ch 22, Tbl 9]

Branching		
Schedule 80 PVC Fitting	$K_{1\text{-}2}$	$K_{1\text{-}3}$
2 in. injection molded branching tee, 100% line flow	0.13 to 0.26	—
50/50 flow	0 to 0.12	0.74 to 1.02
100% branch flow	—	0.98 to 1.39
4 in. injection molded branching tee, 100% line flow	0.07 to 0.22	—
50/50 flow	0.03 to 0.13	0.74 to 0.82
100% branch flow	—	0.97 to 1.12
6 in. injection molded branching tee, 100% line flow	0.01 to 0.14	—
50/50 flow	0.06 to 0.11	0.70 to 0.84
100% branch flow	—	0.95 to 1.15
6 in. fabricated branching tee, 100% line flow	0.21 to 0.22	—
50/50 flow	0.04 to 0.09	1.29 to 1.40
100% branch flow	—	1.74 to 1.88
8 in. injection molded branching tee, 100% line flow	0.04 to 0.09	—
50/50 flow	0.04 to 0.07	0.64 to 0.75
100% branch flow	—	0.85 to 0.96
8 in. fabricated branching tee, 100% line flow	0.09 to 0.16	—
50/50 flow	0.08 to 0.13	1.07 to 1.16
100% branch flow	—	1.40 to 1.62
Mixing		
PVC Fitting	$K_{1\text{-}2}$	$K_{3\text{-}2}$
2 in. injection molded mixing tee, 100% line flow	0.12 to 0.25	—
50/50 flow	1.22 to 1.19	0.89 to 1.88
100% mix flow	—	0.89 to 1.54
4 in. injection molded mixing tee, 100% line flow	0.07 to 0.18	—
50/50 flow	1.19 to 1.88	0.98 to 1.88
100% mix flow	—	0.88 to 1.02
6 in. injection molded mixing tee, 100% line flow	0.06 to 0.14	—
50/50 flow	1.26 to 1.80	1.02 to 1.60
100% mix flow	—	0.90 to 1.07
6 in. fabricated mixing tee, 100% line flow	0.19 to 0.21	—
50/50 flow	2.94 to 3.32	2.57 to 3.17
100% mix flow	—	1.72 to 1.98
8 in. injection molded mixing tee, 100% line flow	0.04 to 0.09	—
50/50 flow	1.10 to 1.60	0.96 to 1.32
100% mix flow	—	0.81 to 0.93
8 in. fabricated mixing tee, 100% line flow	0.13 to 0.70	—
50/50 flow	2.36 to 10.62	2.02 to 2.67
100% mix flow	—	1.34 to 1.53

Coefficients based on average velocity of 8 fps. Range of values varies with fitting manufacturers. Line or straight flow is Q_2/Q_1 = 100%. Branch flow is Q_2/Q_1 = 0%.

4. STEAM

Table 4.1 Properties of Saturated Steam

Pressure p	Temperature t, °F	Specific Volume V_g, cu ft/lb	Enthalpy, Btu/lb Saturated Water h_f	Enthalpy, Btu/lb Evaporation h_{fg}	Enthalpy, Btu/lb Saturated Steam h_g
0.25 in. Hg	40.34	2423.7	8.28	1071.1	1079.4
0.50	58.80	1256.4	26.86	1060.6	1087.5
1.00	79.03	652.3	47.05	1049.2	1096.3
2.00	101.14	339.2	69.10	1036.6	1105.7
2 psia	126.08	173.7	93.99	1022.2	1116.2
3	141.48	118.7	109.37	1013.2	1122.6
4	152.97	90.63	120.86	1006.4	1127.3
5	162.24	73.52	130.13	1001.0	1131.1
6	170.06	61.98	137.94	996.2	1164.2
7	176.85	53.64	144.76	992.1	1136.9
8	182.86	47.34	150.79	988.5	1139.3
9	188.28	42.40	156.22	985.2	1141.4
10	193.21	38.42	161.17	982.1	1143.3
12	201.96	32.40	169.96	976.6	1146.6
14	209.56	28.04	177.61	971.9	1149.5
14.696	212.00	26.80	180.07	970.3	1150.4
20	227.96	20.09	196.16	960.1	1156.3
30	250.33	13.75	218.82	945.3	1164.1
40	267.25	10.50	236.03	933.7	1169.7
50	281.01	8.515	250.09	924.0	1174.1
60	292.71	7.175	262.09	915.5	1177.6
70	302.92	6.206	272.61	907.9	1180.6
80	312.03	5.472	282.02	901.1	1183.1
90	320.27	4.896	290.56	894.7	1185.3
100	327.81	4.432	298.40	888.8	1187.2
120	341.25	3.728	312.44	877.9	1190.4
140	353.02	3.220	324.82	868.2	1193.0
160	363.53	2.834	335.93	859.2	1195.1
180	373.06	2.532	346.03	850.8	1196.9
200	381.79	2.228	355.36	843.0	1198.4

Sources:

1. Keenan, J., and F. Keyes. 1936. *Thermodynamic Properties of Steam*. John Wiley and Sons, New York.
2. Holladay, W., and C. Otterholm. 1985. *Numbers*. Altadena, CA.

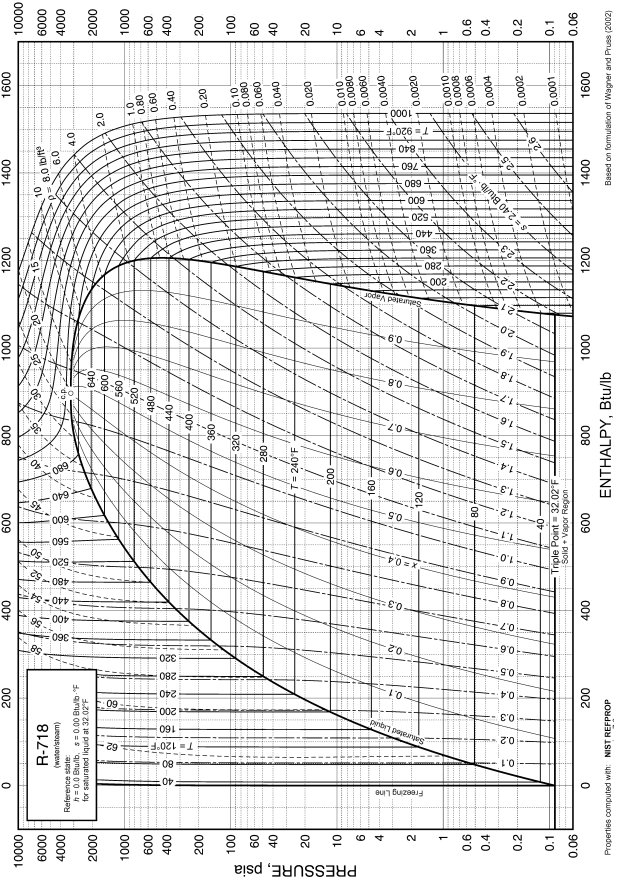


Figure 4.1 Pressure-Enthalpy Diagram for Refrigerant 718 (Water/Steam) [2017F, Ch 30, Fig 20]

Table 4.2 Flow Rate of Low-Pressure Steam in Schedule 40 Pipe [2017F, Ch 22, Tbl 32]

Nominal Pipe Size, in.	Pressure Drop per 100 ft of Length: 1/16 psi (1 oz/in²)		1/8 psi (2 oz/in²)		1/4 psi (4 oz/in²)		1/2 psi (8 oz/in²)		3/4 psi (12 oz/in²)		1 psi		2 psi	
	Sat. Press., psig		Sat. Press., psig		Sat. Press., psig		Sat. Press., psig		Sat. Press., psig		Sat. Press., psig		Sat. Press., psig	
	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12	3.5	12
3/4	9	11	14	16	20	24	29	35	36	43	42	50	60	73
1	17	21	26	31	37	46	54	66	68	82	81	95	114	137
1-1/4	36	45	53	66	78	96	111	138	140	170	162	200	232	280
1-1/2	56	70	84	100	120	147	174	210	218	260	246	304	360	430
2	108	134	162	194	234	285	336	410	420	510	480	590	710	850
2-1/2	174	215	258	310	378	460	540	660	680	820	780	950	1,150	1,370
3	318	380	465	550	660	810	960	1,160	1,190	1,430	1,380	1,670	1,950	2,400
3-1/2	462	550	670	800	990	1,218	1,410	1,700	1,740	2,100	2,000	2,420	2,950	3,450
4	640	800	950	1,160	1,410	1,690	1,980	2,400	2,450	3,000	2,880	3,460	4,200	4,900
5	1,200	1,430	1,680	2,100	2,440	3,000	3,570	4,250	4,380	5,250	5,100	6,100	7,500	8,600
6	1,920	2,300	2,820	3,350	3,960	4,850	5,700	6,800	7,000	8,600	8,400	10,000	11,900	14,200
8	3,900	4,800	5,570	7,000	8,100	10,000	11,400	14,300	14,500	17,700	16,500	20,500	24,000	29,500
10	7,200	8,800	10,200	12,600	15,000	18,200	21,000	26,000	26,200	32,000	30,000	37,000	42,700	52,000
12	11,400	13,700	16,500	19,500	23,400	28,400	33,000	40,000	41,000	49,500	48,000	57,500	67,800	81,000

Notes:
1. Flow rate is in lb/h at initial saturation pressures of 3.5 and 12 psig. Flow is based on Moody friction factor, where the flow of condensate does not inhibit the flow of steam.
2. The flow rates at 3.5 psig cover saturated pressure from 1 to 6 psig, and the rates at 12 psig cover saturated pressure from 8 to 16 psig with an error not exceeding 8%.

Table 4.3 Medium-Pressure Steam Pipe Capacities (30 psig)—Pounds Per Hour

Pipe Size (in.)	Pressure Drop per 100 ft					
	1/8 psi	1/4 psi	1/2 psi	3/4 psi	1 psi	2 psi
Supply Mains and Risers 25–35 psig—Max Error 8%						
3/4	15	22	31	38	45	63
1	31	46	63	77	89	125
1 1/4	69	100	141	172	199	281
1 1/2	107	154	219	267	309	437
2	217	313	444	543	627	886
2 1/2	358	516	730	924	1,033	1,460
3	651	940	1,330	1,628	1,880	2,660
3 1/2	979	1,414	2,000	2,447	2,825	4,000
4	1,386	2,000	2,830	3,464	4,000	5,660
5	2,560	3,642	5,225	6,402	7,390	10,460
6	4,210	6,030	8,590	10,240	12,140	17,180
8	8,750	12,640	17,860	21,865	25,250	35,100
10	16,250	23,450	33,200	40,625	46,900	66,350
12	25,640	36,930	52,320	64,050	74,000	104,500
Return Mains and Risers 0–4 psig—Max Return Pressure						
3/4	115	170	245	308	365	
1	230	340	490	615	730	
1 1/4	485	710	1,025	1,285	1,530	
1 1/2	790	1,155	1,670	2,100	2,500	
2	1,575	2,355	3,400	4,300	5,050	
2 1/2	2,650	3,900	5,600	7,100	8,400	
3	4,850	7,100	10,250	12,850	15,300	
3 1/2	7,200	10,550	15,250	19,150	22,750	
4	10,200	15,000	21,600	27,000	32,250	
5	19,000	27,750	40,250	55,500	60,000	
6	31,000	45,500	65,500	83,000	98,000	

Table 4.4 High-Pressure Steam Pipe Capacities (150 psig)—Pounds Per Hour

Pipe Size (in.)	Pressure Drop per 100 ft							
	1/8 psi	1/4 psi	1/2 psi	3/4 psi	1 psi	2 psi	5 psi	6 psi
Supply Mains and Risers 130–180 psig—Max Error 8%								
3/4	29	41	58	82	116	184	300	420
1	58	82	117	165	233	369	550	790
1 1/4	130	185	262	370	523	827	1,230	1,720
1 1/2	203	287	407	575	813	1,230	1,730	2,600
2	412	583	825	1,167	1,650	2,000	3,410	4,820
2 1/2	683	959	1,359	1,920	2,430	3,300	5,200	7,600
3	1,237	1,750	2,476	3,500	4,210	6,000	9,400	13,500
3 1/2	1,855	2,626	3,715	5,250	6,020	8,500	13,100	20,000
4	2,625	3,718	5,260	7,430	8,400	12,300	19,200	28,000
5	4,858	6,875	9,725	13,750	15,000	21,200	33,100	47,500
6	7,960	11,275	15,950	22,550	25,200	36,500	56,500	80,000
8	16,590	23,475	33,200	46,950	50,000	70,200	120,000	170,000
10	30,820	43,430	61,700	77,250	90,000	130,000	210,000	300,000
12	48,600	68,750	97,250	123,000	155,000	200,000	320,000	470,000
Return Mains and Risers 1–20 psig—Max Return Pressure								
3/4	156	232	360	465	560	890		
1	313	462	690	910	1,120	1,780		
1 1/4	650	960	1,500	1,950	2,330	3,700		
1 1/2	1,070	1,580	2,460	3,160	3,800	6,100		
2	2,160	3,300	4,950	6,400	7,700	12,300		
2 1/2	3,600	5,350	8,200	10,700	12,800	20,400		
3	6,500	9,600	15,000	19,500	23,300	37,200		
3 1/2	9,600	14,400	22,300	28,700	34,500	55,000		
4	13,700	20,500	31,600	40,500	49,200	78,500		
5	25,600	38,100	58,500	76,000	91,500	146,000		
6	42,000	62,500	96,000	125,000	150,000	238,000		

Table 4.5 Return Main and Riser Capacities for Low-Pressure Steam Systems—Pounds per Hour [2017F, Ch 22, Tbl 34]

	Pipe Size, In.	Pressure Drop per 100 ft																	
		1/32 psi (1/2 oz)			1/24 psi (2/3 oz)			1/6 psi (1 oz)			1/8 psi (2 oz)			1/4 psi (4 oz)			1/2 psi (8 oz)		
		Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
Return Main	3/4	–	-	-	-	-	42	-	-	100	-	-	142	-	-	200	-	-	283
	1	125	62	-	145	71	143	175	80	175	250	103	249	350	115	350	-	-	494
	1 1/4	213	130	-	248	149	244	300	168	300	425	217	426	600	241	600	-	-	848
	1 1/2	338	206	-	393	236	388	475	265	475	675	340	674	950	378	950	-	-	1,340
	2	700	470	-	810	535	815	1,000	575	1,000	1,400	740	1,420	2,000	825	2,000	-	-	2,830
	2 1/2	1,180	760	-	1,580	868	1,360	1,680	950	1,680	2,350	1,230	2,380	3,350	1,360	3,350	-	-	4,730
	3	1,880	1,460	-	2,130	1,560	2,180	2,680	1,750	2,680	3,750	2,250	3,800	5,350	2,500	5,350	-	-	7,560
	3 1/2	2,750	1,970	-	3,300	2,200	3,250	4,000	2,500	4,000	5,500	3,230	5,680	8,000	3,580	8,000	-	-	11,300
	4	3,880	2,930	-	4,580	3,350	4,500	5,500	3,750	5,500	7,750	4,830	7,810	11,000	5,380	11,000	-	-	15,500
	5	-	-	-	-	-	7,880	-	-	9,680	-	-	13,700	-	-	19,400	-	-	27,300
	6	-	-	-	-	-	12,600	-	-	15,500	-	-	22,000	-	-	31,000	-	-	43,800
Riser	3/4	-	48	-	-	48	143	-	48	175	-	48	249	-	48	350	-	-	494
	1	-	113	-	-	113	244	-	113	300	-	113	426	-	113	600	-	-	848
	1 1/4	-	248	-	-	248	388	-	248	475	-	248	674	-	248	950	-	-	1,340
	1 1/2	-	375	-	-	375	815	-	375	1,000	-	375	1,420	-	375	2,000	-	-	2,830
	2	-	750	-	-	750	1,360	-	750	1,680	-	750	2,380	-	750	3,350	-	-	4,730
	2 1/2	-	-	-	-	-	2,180	-	-	2,680	-	-	3,800	-	-	5,350	-	-	7,560
	3	-	-	-	-	-	3,250	-	-	4,000	-	-	5,680	-	-	8,000	-	-	11,300
	3 1/2	-	-	-	-	-	4,480	-	-	5,500	-	-	7,810	-	-	11,000	-	-	15,500
	4	-	-	-	-	-	7,880	-	-	9,680	-	-	13,700	-	-	19,400	-	-	27,300
	5	-	-	-	-	-	12,600	-	-	15,500	-	-	22,000	-	-	31,000	-	-	43,800

5. PIPING

Table 5.1 Steel Pipe Data [2017F, Ch 22, Tbl 16]

Nominal Size, in.	Pipe OD, in.	Schedule Number or Weight[a]	Wall Thickness t, in.	Inside Diameter d, in.	Surface Area		Cross Section		Weight		Working Pressure[c] ASTM A53 B to 400°F		
					Outside, ft²/ft	Inside, ft²/ft	Metal Area, in²	Flow Area, in²	Pipe, lb/ft	Water, lb/ft	Mfr. Process	Joint Type[b]	psig
1/4	0.540	40 ST	0.088	0.364	0.141	0.095	0.125	0.104	0.424	0.045	CW	T	188
		80 XS	0.119	0.302	0.141	0.079	0.157	0.072	0.535	0.031	CW	T	871
3/8	0.675	40 ST	0.091	0.493	0.177	0.129	0.167	0.191	0.567	0.083	CW	T	203
		80 XS	0.126	0.423	0.177	0.111	0.217	0.141	0.738	0.061	CW	T	820
1/2	0.840	40 ST	0.109	0.622	0.220	0.163	0.250	0.304	0.850	0.131	CW	T	214
		80 XS	0.147	0.546	0.220	0.143	0.320	0.234	1.087	0.101	CW	T	753
3/4	1.050	40 ST	0.113	0.824	0.275	0.216	0.333	0.533	1.13	0.231	CW	T	217
		80 XS	0.154	0.742	0.275	0.194	0.433	0.432	1.47	0.187	CW	T	681
1	1.315	40 ST	0.133	1.049	0.344	0.275	0.494	0.864	1.68	0.374	CW	T	226
		80 XS	0.179	0.957	0.344	0.251	0.639	0.719	2.17	0.311	CW	T	642
1 1/4	1.660	40 ST	0.140	1.380	0.435	0.361	0.669	1.50	2.27	0.647	CW	T	229
		80 XS	0.191	1.278	0.435	0.335	0.881	1.28	2.99	0.555	CW	T	594
1 1/2	1.900	40 ST	0.145	1.610	0.497	0.421	0.799	2.04	2.72	0.881	CW	T	231
		80 XS	0.200	1.500	0.497	0.393	1.068	1.77	3.63	0.765	CW	T	576
2	2.375	40 ST	0.154	2.067	0.622	0.541	1.07	3.36	3.65	1.45	CW	T	230
		80 XS	0.218	1.939	0.622	0.508	1.48	2.95	5.02	1.28	CW	T	551
2 1/2	2.875	40 ST	0.203	2.469	0.753	0.646	1.70	4.79	5.79	2.07	CW	W	533
		80 XS	0.276	2.323	0.753	0.608	2.25	4.24	7.66	1.83	CW	W	835

Table 5.1 Steel Pipe Data [2017F, Ch 22, Tbl 16] ***(Continued)***

Nominal Size, in.	Pipe OD, in.	Schedule Number or Weight[a]	Wall Thickness t, in.	Inside Diameter d, in.	Surface Area		Cross Section		Weight		Working Pressure[c] ASTM A53 B to 400°F		
					Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Pipe, lb/ft	Water, lb/ft	Mfr. Process	Joint Type[b]	psig
3	3.500	40 ST	0.216	3.068	0.916	0.803	2.23	7.39	7.57	3.20	CW	W	482
		80 XS	0.300	2.900	0.916	0.759	3.02	6.60	10.25	2.86	CW	W	767
4	4.500	40 ST	0.237	4.026	1.178	1.054	3.17	12.73	10.78	5.51	CW	W	430
		80 XS	0.337	3.826	1.178	1.002	4.41	11.50	14.97	4.98	CW	W	695
6	6.625	40 ST	0.280	6.065	1.734	1.588	5.58	28.89	18.96	12.50	ERW	W	696
		80 XS	0.432	5.761	1.734	1.508	8.40	26.07	28.55	11.28	ERW	W	1209
8	8.625	30	0.277	8.071	2.258	2.113	7.26	51.16	24.68	22.14	ERW	W	526
		40 ST	0.322	7.981	2.258	2.089	8.40	50.03	28.53	21.65	ERW	W	643
		80 XS	0.500	7.625	2.258	1.996	12.76	45.66	43.35	19.76	ERW	W	1106
10	10.75	30	0.307	10.136	2.814	2.654	10.07	80.69	34.21	34.92	ERW	W	485
		40 ST	0.365	10.020	2.814	2.623	11.91	78.85	40.45	34.12	ERW	W	606
		XS	0.500	9.750	2.814	2.552	16.10	74.66	54.69	32.31	ERW	W	887
		80	0.593	9.564	2.814	2.504	18.92	71.84	64.28	31.09	ERW	W	1081
12	12.75	30	0.330	12.090	3.338	3.165	12.88	114.8	43.74	49.68	ERW	W	449
		ST	0.375	12.000	3.338	3.141	14.58	113.1	49.52	48.94	ERW	W	528
		40	0.406	11.938	3.338	3.125	15.74	111.9	53.48	48.44	ERW	W	583
		XS	0.500	11.750	3.338	3.076	19.24	108.4	65.37	46.92	ERW	W	748
		80	0.687	11.376	3.338	2.978	26.03	101.6	88.44	43.98	ERW	W	1076

Table 5.1 Steel Pipe Data [2017F, Ch 22, Tbl 16] ***(Continued)***

Nominal Size, in.	Pipe OD, in.	Schedule Number or Weight[a]	Wall Thickness *t*, in.	Inside Diameter *d*, in.	Surface Area: Outside, ft^2/ft	Surface Area: Inside, ft^2/ft	Cross Section: Metal Area, in^2	Cross Section: Flow Area, in^2	Weight: Pipe, lb/ft	Weight: Water, lb/ft	Working Pressure[c] ASTM A53 B to 400°F: Mfr. Process	Joint Type[b]	psig
14	14.00	30 ST	0.375	13.250	3.665	3.469	16.05	137.9	54.53	59.67	ERW	W	481
		40	0.437	13.126	3.665	3.436	18.62	135.3	63.25	58.56	ERW	W	580
		XS	0.500	13.000	3.665	3.403	21.21	132.7	72.04	57.44	ERW	W	681
		80	0.750	12.500	3.665	3.272	31.22	122.7	106.05	53.11	ERW	W	1081
16	16.00	30 ST	0.375	15.250	4.189	3.992	18.41	182.6	62.53	79.04	ERW	W	421
		40 XS	0.500	15.000	4.189	3.927	24.35	176.7	82.71	76.47	ERW	W	596
18	18.00	ST	0.375	17.250	4.712	4.516	20.76	233.7	70.54	101.13	ERW	W	374
		30	0.437	17.126	4.712	4.483	24.11	230.3	81.91	99.68	ERW	W	451
		XS	0.500	17.000	4.712	4.450	27.49	227.0	93.38	98.22	ERW	W	530
		40	0.562	16.876	4.712	4.418	30.79	223.7	104.59	96.80	ERW	W	607
20	20.00	20 ST	0.375	19.250	5.236	5.039	23.12	291.0	78.54	125.94	ERW	W	337
		30 XS	0.500	19.000	5.236	4.974	30.63	283.5	104.05	122.69	ERW	W	477
		40	0.593	18.814	5.236	4.925	36.15	278.0	122.82	120.30	ERW	W	581

[a] Numbers are schedule numbers per ASME Standard B36.10M; ST = Standard Weight; XS = Extra Strong.
[b] T = Thread; W = Weld
[c] Working pressures were calculated per ASME B31.9 using furnace butt-weld (continuous weld, CW) pipe through 4 in. and electric resistance weld (ERW) thereafter. The allowance A has been taken as(1)12.5% of *t* for mill tolerance on pipe wall thickness, *plus*
(2)An arbitrary corrosion allowance of 0.025 in. for pipe sizes through NPS 2 and 0.065 in. from NPS 2 1/2 through 20, *plus*
(3)A thread cutting allowance for sizes through NPS 2.
Because the pipe wall thickness of threaded standard pipe is so small after deducting allowance A, the mechanical strength of the pipe is impaired. It is good practice to limit standard weight threaded pipe pressure to 90 psig for steam and 125 psig for water.

Table 5.2 Copper Tube Data [2017F, Ch 22, Tbl 17]

Nominal Diameter, in.	Type	Wall Thickness t, in.	Diameter		Surface Area		Cross Section		Weight		Working Pressure[a,b,c] ASTM B88 to 250°F	
			Outside D, in.	Inside d, in.	Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
1/4	K	0.035	0.375	0.305	0.098	0.080	0.037	0.073	0.145	0.032	851	1596
	L	0.030	0.375	0.315	0.098	0.082	0.033	0.078	0.126	0.034	730	1368
3/8	K	0.049	0.500	0.402	0.131	0.105	0.069	0.127	0.269	0.055	894	1676
	L	0.035	0.500	0.430	0.131	0.113	0.051	0.145	0.198	0.063	638	1197
	M	0.025	0.500	0.450	0.131	0.118	0.037	0.159	0.145	0.069	456	855
1/2	K	0.049	0.625	0.527	0.164	0.138	0.089	0.218	0.344	0.094	715	1341
	L	0.040	0.625	0.545	0.164	0.143	0.074	0.233	0.285	0.101	584	1094
	M	0.028	0.625	0.569	0.164	0.149	0.053	0.254	0.203	0.110	409	766
5/8	K	0.049	0.750	0.652	0.196	0.171	0.108	0.334	0.418	0.144	596	1117
	L	0.042	0.750	0.666	0.196	0.174	0.093	0.348	0.362	0.151	511	958
3/4	K	0.065	0.875	0.745	0.229	0.195	0.165	0.436	0.641	0.189	677	1270
	L	0.045	0.875	0.785	0.229	0.206	0.117	0.484	0.455	0.209	469	879
	M	0.032	0.875	0.811	0.229	0.212	0.085	0.517	0.328	0.224	334	625
1	K	0.065	1.125	0.995	0.295	0.260	0.216	0.778	0.839	0.336	527	988
	L	0.050	1.125	1.025	0.295	0.268	0.169	0.825	0.654	0.357	405	760
	M	0.035	1.125	1.055	0.295	0.276	0.120	0.874	0.464	0.378	284	532
1 1/4	K	0.065	1.375	1.245	0.360	0.326	0.268	1.217	1.037	0.527	431	808
	L	0.055	1.375	1.265	0.360	0.331	0.228	1.257	0.884	0.544	365	684
	M	0.042	1.375	1.291	0.360	0.338	0.176	1.309	0.682	0.566	279	522
	DWV	0.040	1.375	1.295	0.360	0.339	0.168	1.317	0.650	0.570	265	497

Table 5.2 Copper Tube Data [2017F, Ch 22, Tbl 17] ***(Continued)***

Nominal Diameter, in.	Type	Wall Thickness *t*, in.	Diameter		Surface Area		Cross Section		Weight		Working Pressure[a,b,c] ASTM B88 to 250°F	
			Outside *D*, in.	Inside *d*, in.	Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
1 1/2	K	0.072	1.625	1.481	0.425	0.388	0.351	1.723	1.361	0.745	404	758
	L	0.060	1.625	1.505	0.425	0.394	0.295	1.779	1.143	0.770	337	631
	M	0.049	1.625	1.527	0.425	0.400	0.243	1.831	0.940	0.792	275	516
	DWV	0.042	1.625	1.541	0.425	0.403	0.209	1.865	0.809	0.807	236	442
2	K	0.083	2.125	1.959	0.556	0.513	0.532	3.014	2.063	1.304	356	668
	L	0.070	2.125	1.985	0.556	0.520	0.452	3.095	1.751	1.339	300	573
	M	0.058	2.125	2.009	0.556	0.526	0.377	3.170	1.459	1.372	249	467
	DWV	0.042	2.125	2.041	0.556	0.534	0.275	3.272	1.065	1.416	180	338
2 1/2	K	0.095	2.625	2.435	0.687	0.637	0.755	4.657	2.926	2.015	330	619
	L	0.080	2.625	2.465	0.687	0.645	0.640	4.772	2.479	2.065	278	521
	M	0.065	2.625	2.495	0.687	0.653	0.523	4.889	2.026	2.116	226	423
3	K	0.109	3.125	2.907	0.818	0.761	1.033	6.637	4.002	2.872	318	596
	L	0.090	3.125	2.945	0.818	0.771	0.858	6.812	3.325	2.947	263	492
	M	0.072	3.125	2.981	0.818	0.780	0.691	6.979	2.676	3.020	210	394
	DWV	0.045	3.125	3.035	0.818	0.795	0.435	7.234	1.687	3.130	131	246
3 1/2	K	0.120	3.625	3.385	0.949	0.886	1.321	8.999	5.120	3.894	302	566
	L	0.100	3.625	3.425	0.949	0.897	1.107	9.213	4.291	3.987	252	472
	M	0.083	3.625	3.459	0.949	0.906	0.924	9.397	3.579	4.066	209	392
4	K	0.134	4.125	3.857	1.080	1.010	1.680	11.684	6.510	5.056	296	555
	L	0.110	4.125	3.905	1.080	1.022	1.387	11.977	5.377	5.182	243	456
	M	0.095	4.125	3.935	1.080	1.030	1.203	12.161	4.661	5.262	210	394
	DWV	0.058	4.125	4.009	1.080	1.050	0.741	12.623	2.872	5.462	128	240

Table 5.2 Copper Tube Data [2017F, Ch 22, Tbl 17] ***(Continued)***

Nominal Diameter, in.	Type	Wall Thickness *t*, in.	Diameter		Surface Area		Cross Section		Weight		Working Pressure[a,b,c] ASTM B88 to 250°F	
			Outside *D*, in.	Inside *d*, in.	Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
5	K	0.160	5.125	4.805	1.342	1.258	2.496	18.133	9.671	7.846	285	534
	L	0.125	5.125	4.875	1.342	1.276	1.963	18.665	7.609	8.077	222	417
	M	0.109	5.125	4.907	1.342	1.285	1.718	18.911	6.656	8.183	194	364
	DWV	0.072	5.125	4.981	1.342	1.304	1.143	19.486	4.429	8.432	128	240
6	K	0.192	6.125	5.741	1.603	1.503	3.579	25.886	13.867	11.201	286	536
	L	0.140	6.125	5.845	1.603	1.530	2.632	26.832	10.200	11.610	208	391
	M	0.122	6.125	5.881	1.603	1.540	2.301	27.164	8.916	11.754	182	341
	DWV	0.083	6.125	5.959	1.603	1.560	1.575	27.889	6.105	12.068	124	232
8	K	0.271	8.125	7.583	2.127	1.985	6.687	45.162	25.911	19.542	304	570
	L	0.200	8.125	7.725	2.127	2.022	4.979	46.869	19.295	20.280	224	421
	M	0.170	8.125	7.785	2.127	2.038	4.249	47.600	16.463	20.597	191	358
	DWV	0.109	8.125	7.907	2.127	2.070	2.745	49.104	10.637	21.247	122	229
10	K	0.338	10.125	9.449	2.651	2.474	10.392	70.123	40.271	30.342	304	571
	L	0.250	10.125	9.625	2.651	2.520	7.756	72.760	30.054	31.483	225	422
	M	0.212	10.125	9.701	2.651	2.540	6.602	73.913	25.584	31.982	191	358
12	K	0.405	12.125	11.315	3.174	2.962	14.912	100.554	57.784	43.510	305	571
	L	0.280	12.125	11.565	3.174	3.028	10.419	105.046	40.375	45.454	211	395
	M	0.254	12.125	11.617	3.174	3.041	9.473	105.993	36.706	45.863	191	358

[a]When using soldered or brazed fittings, the joint determines the limiting pressure.
[b]Working pressures were calculated using ASME Standard B31.9 allowable stresses. A 5% mill tolerance has been used on the wall thickness. Higher tube ratings can be calculated using the allowable stress for lower temperatures.
[c]If soldered or brazed fittings are used on hard drawn tubing, use the annealed ratings. Full-tube allowable pressures can be used with suitably rated flare or compression-type fittings.

Table 5.3 Properties of Pipe Materials[a] [2017F, Ch 22, Tbl 18]

Material			Tensile Strength, psi (at 73°F)	Hydrostatic[b] Design Stress, psi (at 73°F)		Upper Temperature Limit, °F		HDS[b] Upper Limit, psi	Specific Gravity[c]	Impact Strength, ft·lb/in (at 73°F)	Modulus of Elasticity, psi (at 73°F)	Coefficient of Expansion, in/10^6 in·°F	Thermal Conductivity, Btu·in/h·ft²·°F	Relative Pipe Cost[d]
Designation	Type and Grade	Cell No.		Mfr.	ASME B31	Mfr.	ASME B31							
Metals														
Copper	L	Drawn – hard	36,000		9,000		400	8,200	8.90		17,000,000	9.4	232	2.2
Steel	A53 B	ERW	60,000	12,800			800	9,200	7.80	30	27,500,000	6.31	26	1.0
Stainless steel	304	Drawn or Welded	73,200				350		7.90		28,000,000	9.6	8	2.0
Thermoplastics														
PVC 1120	T I,G1	12454-B	7,500	2,000	2,000	140	150	440	1.40	0.8	420,000	30.0	1.1	0.6
PVC 1200	T I,G2	12454-C			2,000		150				410,000	35.0		
PVC 2120	T II,G1	14333-D			2,000		150					30.0		
CPVC 4120	T IV,G1	23447-B	8,000	2,000	2,000	210	210	320	1.55	1.5	423,000	35.0	0.95	0.8
PE 2306	Gr. P23				630		140				90,000	80.0		
PE 3306	Gr. P34				630		160				130,000	70.0		
PE 3406	Gr. P33				630		180				150,000	60.0		
HDPE 3408	Gr. P34	355434-C	5,000	1,600	800	140	180	800	0.96	12	110,000	120.0	2.7	1.1
PP			5,000	705		212	210		0.91	1.3	120,000	60.0	1.3	2.9
ABS	Acrylonitrile copolymer	6-3-3	5,500			176			1.06	8.5	240,000	56.0	1.7	3.4
ABS 1210	T I,G2	5-2-2		1,000		180	640		250,000	55.0				
ABS 1316	T I,G3	3-5-5		1,600		180	1,000		340,000	40.0				
ABS 2112	T II,G1	4-4-5		1,250		180	800			40.0				
PVDF			7,000	1,275		280	275	306	1.78	3.8	125,000	79.0	0.8	2.6

Table 5.3 Properties of Pipe Materials[a] [2017F, Ch 22, Tbl 18] *(Continued)*

Material			Tensile Strength, psi (at 73°F)	Hydrostatic[b] Design Stress, psi (at 73°F)		Upper Temperature Limit, °F		HDS[b] Upper Limit, psi	Specific Gravity[c]	Impact Strength, ft·lb/in (at 73°F)	Modulus of Elasticity, psi (at 73°F)	Coefficient of Expansion, in/10^6 in·°F	Thermal Conductivity, Btu·in/h·ft²·°F	Relative Pipe Cost[d]
Designation	Type and Grade	Cell No.		Mfr.	ASME B31	Mfr.	ASME B31							
Thermosetting														
Epoxy-glass	RTRP-11AF		44,000	8,000		300	7,000				1,000,000	9 to 13	2.9	1.5
PEX	A,B,C[e]		3,200	630	200	180	79		0.94		75,000	90.0	3.2	0.7
Polyester-glass	RTRP-12EF		44,000	9,000	200	200	5,000				1,000,000	9 to 11	1.3	1.5

[a]Properties listed are for specific materials listed; each plastic has other formulations. Consult the manufacturer of the system chosen. These values are for comparative purposes.
[b]Hydrostatic design stress (HDS) is equivalent to allowable design stress
[c]Relative to water at 62.4 lb/ft^3.
[d]Based on cost for 2 in. pipe only, without factoring in fittings, joints, hangers, and labor.
[e]A, B, and C are the three manufacturing processes of PEX pipe. The classifications are not related to a ranking system.

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1]

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Chilled water	≤ 2	Steel Type F (CW)	Schedule 40	Thread	Cast iron	125	250	125
	2.5 to 12	Steel A or B, Type E (ERW)	Schedule 40	Weld	Wrought steel	Standard	250	400
				Flange	Wrought steel	150	250	250
					Cast iron	125	250	175
					Cast iron	250	250	400
		Copper, hard or soft	Type K or L	Solder	Wrought or cast Cu		100	370 Type K soft
				Flared (soft)				635 Type K hard
				Rolled groove (2 to 8 in.)				250 Type L soft
				Press-connect (0.5 to 4 in.)				435 Type L hard
				Push connect (0.5 to 2 in.)				
				Mechanical formed				
				Braze	Wrought or cast Cu		100	250 Type L soft
				Weld				370 Type K soft
		Copper, hard	Type M	Solder	Wrought or cast Cu		100	395 Type M hard
				Rolled groove (2 to 8 in.)				
				Press-connect (0.5 to 4 in.)				
				Push connect (0.5 to 2 in.)				
				Mechanical formed				
				Braze	Wrought or cast Cu		100	230 Type M soft
				Weld				
	0.375 to 1.0	PEX (barrier)	SDR-9	Crimp	Bronze		73	145
				Clamp	Brass			
				Expansion	Copper			
				Compression	Engineered plastic			
				Push fit				
				Proprietary				

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Chilled water (cont'd)	0.5 to 6	PE	Schedule 40,[f] 80, SDR	Thermal fusion, compression	PE		120 (140 limit for some applications)	Varies with pipe wall thickness, grade, schedule, size. Check manufacturer's documentation for design ratings 30 to 110 at 130°F
Heating and recirculating	2 and smaller	Steel Type F (CW)	Schedule 40	Thread	Cast iron	125	250	125
	0.25 to 12	Steel B Type E (ERW)	Schedule 40	Weld	Wrought steel	Standard	250	400
				Flange	Wrought steel	150	250	250
					Cast iron	125	250	125
					Cast iron	250	250	400
		Copper, hard or soft	Type K or L	Solder	Wrought or cast Cu		200	300 Type K soft
				Braze				635 Type K hard
				Flared (soft)				205 Type L soft
				Rolled groove (2 to 8 in.)				435 Type L hard
				Press-connect (0.5 to 4 in.)				
				Push connect (0.5 to 2 in.)				
				Mechanical formed				
				Braze	Wrought or cast Cu		200	300 Type K soft
				Weld				205 Type L soft
	0.25 to 12	Copper, hard	Type M	Solder	Wrought or cast Cu		200	395 Type M hard
				Rolled groove (2 to 8 in.)				
				Press-connect (0.5 to 4 in.)				
				Push connect (0.5 to 2 in.)				
				Mechanical formed				
				Braze	Wrought or cast Cu		200	200 Type M soft
				Weld				

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Heating and recirculating (cont'd)	0.375 to 1.0	PEX (barrier)	SDR-9	Crimp	Bronze		200	79
				Clamp	Brass			
				Expansion	Copper			
				Compression	Engineered plastic			
				Push fit				
				Proprietary				
Steam and condensate	2 and smaller	Steel Type F (CW) or S	Schedule 40[d]	Thread	Cast iron	125		90
				Thread	Malleable iron	150		90
				Socket	Forged steel	3000		90
		Steel B Type E (ERW) or S	Schedule 40[d]	Thread	Cast iron	125		100
				Thread	Malleable iron	150		125
				Socket	Forged steel	3000		400
		Steel B Type E (ERW) or S	Schedule 80	Thread	Cast iron	250		200
				Socket				
				Thread	Malleable iron	300		250
				Socket	Forged steel	3000		400
	2 to 12	Steel B Type E (ERW) or S	Schedule 40	Weld	Wrought steel	Standard		250
				Flange	Wrought steel	150		200
					Cast iron	125		100
		Steel B Type E (ERW) or S	Schedule 80	Weld	Wrought steel	XS		700
				Flange	Wrought steel	300		500
					Cast iron	250		200

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Ground-source heat pump	0.25 to 2	Copper, hard or soft	Type L or ACR	Flared or brazed	Wrought or cast Cu		200	205 Type L soft, 435 Type L hard, 240 ACR soft, 500 ACR hard
	0.375 to 1.0	PEX (barrier)	SDR-9	Crimp Clamp Expansion Compression Push fit Proprietary	Bronze Brass Copper Engineered plastic		180	100
Refrigerant	0.375 to 4.125	Steel B Type E (ERW)	Schedule 40	Weld			Wrought steel	
		Copper, hard	Type L or ACR	Braze	Wrought or Forged Cu		200	435 Type L hard, 240 ACR soft
Natural gas and LP	0.25 to 12	Copper, hard or soft	Type K or L	Solder Rolled groove (2 to 8 in.) Press-connect (0.5 to 4 in.) Push connect (0.5 to 2 in.) Mechanically formed	Wrought or cast Cu		100	370 Type K soft 635 Type K hard 250 Type L soft 435 Type L hard
				Braze Weld	Wrought or cast Cu		100	370 Type K soft 250 Type L soft
	0.375 to 4.125	Copper, hard	ACR	Solder	Wrought or cast Cu		100	500 Type ACR hard
				Braze	Wrought or cast Cu		100	290 Type ACR Soft
	0.375 to 1.0	PEX	SDR-9	Crimp Clamp Expansion Compression Push fit Proprietary	Bronze Brass Copper Engineered plastic		73	145

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Natural gas and LP (cont'd)	0.5 to 6	PE	Schedule 40, 80, SDR	Thermal fusion, compression	PE		120 (140 limit for some applications)	Depends on pipe, grade, schedule, size. Generally 30 to 110 at 130°F
	0.5 to 6	HDPE	SDR	Thermal fusion, compression	HDPE		120	Depends on pipe, grade, schedule, size. Generally 64 for SDR 11 at 120°F
Fuel oil, aboveground	2 to 12	Black Steel, B Type E (ERW) or S (seamless)	Schedule 40	Thread or weld	Black malleable iron Wrought steel weld Forged steel flanges	150 150		
	0.25 to 12	Copper, hard or soft	Type K or L	Solder Flared (soft) Rolled groove (2 to 8 in.) Press-connect (0.5 to 4 in.) Push connect (0.5 to 2 in.) Mechanical formed	Wrought or cast Cu		100	300 Type K soft 635 Type K hard 250 Type L soft 435 Type L hard
				Braze or weld	Wrought or cast Cu		100	300 Type K soft, 250 Type L soft
		Copper, hard	Type M	Solder Braze Rolled groove (2 to 8 in.) Press-connect (0.5 to 4 in.) Push connect (0.5 to 2 in.) Mechanical formed	Wrought or cast Cu		100	395 Type M hard

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Fuel oil, aboveground (cont'd)	0.25 to 12	ABS	Schedule 40,[f] 80, SDR	Solvent weld, thread, flange	ABS		160 limit	Depends on pipe class: approximately 50 at 160°F
	0.5 to 6	HDPE	SDR-9	Thermal fusion, compression	HDPE		120	Depends on pipe, grade, schedule, size. Generally 64 for SDR 11 at 120°F
Compressed air	≤2.5 and smaller	Black steel	Schedule 40	Thread	Black malleable iron	150	350	
	>2.5	Black steel	Schedule 40	Flange or weld	Black malleable iron	150	350	
	0.375 to 4.125	Copper, hard	ACR	Solder Flared (soft) Mechanical formed Braze	Wrought or cast Cu		200 200	240 ACR soft 500 ACR hard 240 ACR hard
	0.5 to 4	ABS HDPE	Schedule 40 Schedule 40, 80, SDR	Solvent weld	ABS HDPE		73	185
	0.375 to 1.0	PEX	SDR-9					
Potable water, inside building	0.25 to 12	Steel, galvanized	Schedule 40	Thread	Galv. cast iron Galv. cast iron	150 150	100 100	125 150
		Copper, hard or soft	Type K or L	Solder[c] Flared (soft) Rolled groove (2 to 8 in.) Press-connect (0.5 to 4 in.) Push connect (0.5 to 2 in.) Mechanical formed	Wrought or cast Cu		100	370 Type K soft 635 Type K hard 250 Type L soft 435 Type L hard
				Braze Weld	Wrought or cast Cu		100	370 Type K soft 250 Type L soft

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Potable water, inside building (cont'd)	0.25 to 12	Copper, hard	Type M	Solder[c] Rolled groove (2 to 8 in.) Press-connect (0.5 to 4 in.) Push connect (0.5 to 2 in.) Mechanical formed	Wrought or cast Cu		100	395 Type M hard
				Braze Weld	Wrought or cast Cu		100	230 Type M soft
	0.5 to 8	CPVC	Schedule 40,[f] 80		CPVC		210 Limit, 200 operating	
	0.375 to 1.0	PEX	SDR-9	Crimp Clamp Expansion Compression Push fit Proprietary	Bronze Brass Copper Engineered plastic		100	145
	0.5 to 6	PE	Schedule 40,[f] 80, SDR	Thermal fusion, compression	PE		120 (140 limit for some applications)	Depends on pipe, grade, schedule, size generally 30 to 110 at 130°F
	0.5 to 6	PP	Schedule 40,[f] 80, SDR	Thermal fusion, flange, Thread[e]	PP		180	50

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Water services, underground	Through 6	Ductile iron	Class 50	Mechanical joint	Cast iron		75	250
	0.25 to 12	Copper, hard or soft	Type K or L	Solder[c]	Wrought or cast Cu		100	370 Type K soft
				Flared (soft)				635 Type K hard
				Rolled groove (2 to 8 in.)				250 Type L soft
				Press-connect (0.5 to 4 in.)				435 Type L hard
				Push connect (0.5 to 2 in.)				
				Mechanical formed				
				Braze	Wrought or cast Cu		100	370 Type K soft
				Weld				250 Type L soft
				Flange	Bronze		100	
	0.25 to 12	Copper, hard	Type M	Solder[c]	Wrought or cast Cu		100	395 Type K hard
				Rolled groove (2 to 8 in.)				
				Press-connect (0.5 to 4 in.)				
				Push connect (0.5 to 2 in.)				
				Mechanical formed				
				Braze	Wrought or cast Cu		100	230 Type M soft
				Weld				
	0.375 to 1.0	PEX	SDR-9	Crimp	Bronze		73	145
				Clamp	Brass			
				Expansion	Copper			
				Compression	Engineered plastic			
				Push fit				
				Proprietary				
	0.25 to 20	PVC	Schedule 40, 80, 120, SDR	Solvent weld, thread,[f] thermal weld	PVC		150 limit, 140 operating	79 to 105, depending on schedule and size

Table 5.4 Common Applications of Pipe, Fittings, and Valves for Heating and Air Conditioning [2017F, Ch 22, Tbl 1] ***(Continued)***

Application	Size, in.	Material	Weight	Joint Type	Fitting Material	Class (When Applicable)	System[g] Temperature, °F	Maximum Pressure at Temperature,[a,b] psi
Drainage, waste, and vent (DWV)	1.25 to 8	Copper, hard	DWV	Solder	Wrought or cast Cu		100	250 DWV hard
	1.25 to 12	ABS	Schedule DWV, 40,[f] 80, SDR	Solvent weld, thread, flange	ABS		160 limit	Depends on pipe class: approximately 50 at 160°F
	1.25 to 20	PV	Schedule 40,[f] 80, 120, SDR	Solvent weld, thread, thermal weld	PVC		150 limit, 140 operating	79 to 105, depending on schedule and size

[a]Maximum allowable working pressures have been derated in this table. Higher system pressures can be used for lower temperatures and smaller pipe sizes. Pipe, fittings, joints, and valves must all be considered.
[b]Temperature and pressure relationships can vary based on pipe material composition, size, class, and schedule.
[c]Lead- and antimony-based solders are prohibited for potable water systems. Brazing should be used.
[d]Piping codes typically require thicker-walled pipe for threaded joints to maintain corrosion allowance and pressure ratings.
[e]All plumbing codes require both hot and cold water piping to have a 100 psi at 180°F rating.
[f]Threads are not recommended on Schedule 40 plastic pipe.
[g]Designer should confirm that all materials are suitably rated for intended operation.

Table 5.5 Thermal Expansion of Metal Pipe [2017F, Ch 22, Tbl 13]

	Saturated Steam Pressure, psig	Temperature, °F	Linear Thermal Expansion, in/100 ft		
			Carbon Steel	Type 304 Stainless Steel	Copper
		–30	–0.19	–0.30	–0.32
		–20	–0.12	–0.20	–0.21
		–10	–0.06	–0.10	–0.11
		0	0	0	0
		10	0.08	0.11	0.12
		20	0.15	0.22	0.24
Vacuum	–14.6	32	0.24	0.36	0.37
Vacuum	–14.6	40	0.30	0.45	0.45
Vacuum	–14.5	50	0.38	0.56	0.57
Vacuum	–14.4	60	0.46	0.67	0.68
Vacuum	–14.3	70	0.53	0.78	0.79
Vacuum	–14.2	80	0.61	0.90	0.90
Vacuum	–14.0	90	0.68	1.01	1.02
Vacuum	–13.7	100	0.76	1.12	1.13
Vacuum	–13.0	120	0.91	1.35	1.37
Vacuum	–11.8	140	1.06	1.57	1.59
Vacuum	–10.0	160	1.22	1.79	1.80
Vacuum	–7.2	180	1.37	2.02	2.05
Vacuum	–3.2	200	1.52	2.24	2.30
	0	212	1.62	2.38	2.43
	2.5	220	1.69	2.48	2.52
	10.3	240	1.85	2.71	2.76
	20.7	260	2.02	2.94	2.99
	34.6	280	2.18	3.17	3.22
	52.3	300	2.35	3.40	3.46
	75.0	320	2.53	3.64	3.70
	103.3	340	2.70	3.88	3.94
	138.3	360	2.88	4.11	4.18
	181.1	380	3.05	4.35	4.42
	232.6	400	3.23	4.59	4.87
	666.1	500	4.15	5.80	5.91
	1528	600	5.13	7.03	7.18
	3079	700	6.16	8.29	8.47
		800	7.23	9.59	9.79
		900	8.34	10.91	11.16
		1000	9.42	12.27	12.54

Table 5.6 Suggested Hanger Spacing and Rod Size for Straight Horizontal Runs
[2017F, Ch 22, Tbl 11]

NPS, in.	Hanger Spacing, ft			Rod Size, in.
	Standard Steel Pipe*		Copper Tube	
	Water	Steam	Water	
1/2	7	8	5	1/4
3/4	7	9	5	1/4
1	7	9	6	1/4
1 1/2	9	12	8	3/8
2	10	13	8	3/8
2 1/2	11	14	9	3/8
3	12	15	10	3/8
4	14	17	12	1/2
6	17	21	14	1/2
8	19	24	16	5/8
10	20	26	18	3/4
12	23	30	19	7/8
14	25	32		1
16	27	35		1
18	28	37		1 1/4
20	30	39		1 1/4

Source: Adapted from MSS Standard SP-69
*Spacing does not apply where span calculations are made or where concentrated loads are placed between supports such as flanges, valves, specialties, etc.

Table 5.7 Capacities of ASTM A36 Steel Threaded Rods
[2017F, Ch 22, Tbl 10]

Rod Diameter, in.	Root Area of Coarse Thread, in^2	Maximum Load,* lb
1/4	0.027	240
3/8	0.068	610
1/2	0.126	1130
5/8	0.202	1810
3/4	0.302	2710
7/8	0.419	3770
1	0.552	4960
1 1/4	0.889	8000

*Based on an allowable stress of 12,000 psi reduced by 25% using the root area in accordance with ASME Standard B31.1 and MSS Standard SP-58.

6. SERVICE WATER HEATING

Water heating energy use is second only to space conditioning in most residential buildings, and is also significant in many commercial and industrial settings. In some climates and applications, water heating is the largest energy use in a building. Moreover, quick availability of adequate amounts of hot water is an important factor in user satisfaction. Both water and energy waste can be significant in poorly designed service water-heating systems: from over- or undersizing pipes and equipment, from poor building layout, and from poor system design and operating strategies. Good service water-heating system design and operating practices can often reduce first costs as well as operating costs.

System Elements

A service water-heating system has (1) one or more heat energy sources, (2) heat transfer equipment, (3) a distribution system, and (4) terminal hot-water usage devices.

Heat energy sources may be (1) fuel combustion; (2) electrical conversion; (3) solar energy; (4) geothermal, air, or other environmental energy; and/or (5) recovered waste heat from sources such as flue gases, ventilation and air-conditioning systems, refrigeration cycles, and process waste discharge.

Heat transfer equipment is direct, indirect, or a combination of the two. For direct equipment, heat is derived from combustion of fuel or direct conversion of electrical energy into heat and is applied within the water-heating equipment. For indirect heat transfer equipment, heat energy is developed from remote heat sources (e.g., boilers; solar energy collection; air, geothermal, or other environmental source; cogeneration; refrigeration; waste heat) and is then transferred to the water in a separate piece of equipment. Storage tanks may be part of or associated with either type of heat transfer equipment.

Distribution systems transport hot water produced by water-heating equipment to terminal hot-water usage devices. Water consumed must be replenished from the building water service main. For locations where constant supply temperatures are desired, circulation piping or a means of heat maintenance must be provided.

Terminal hot-water usage devices are plumbing fixtures and equipment requiring hot water that may have periods of irregular flow, constant flow, and no flow. These patterns and their related water usage vary with different buildings, process applications, and personal preference.

Legionella pneumophila (Legionnaires' Disease)

Legionnaires' disease (a form of severe pneumonia) is caused by inhaling the bacteria *Legionella pneumophila*. It has been discovered in the service water systems of various buildings throughout the world.

Service water temperature in the 140°F range is recommended to limit the potential for *L. pneumophila* growth. This high temperature increases the potential for scalding, so care must be taken such as installing an anti-scald or mixing valve.

More information on this subject can be found in ASHRAE Standard 188 and ASHRAE Guideline 12.

Load Diversity

The greatest difficulty in designing water-heating systems comes from uncertainty about design hot-water loads, especially for buildings not yet built. Although it is fairly simple to test maximum flow rates of various hot-water fixtures and appliances, actual flow rates and durations are user-dependent. Moreover, the timing of different hot-water use events varies from day to day, with some overlap, but almost never will all fixtures be used simultaneously. As the number of hot-water-using fixtures and appliances grows, the percent of those fixtures used simultaneously decreases.

Some of the hot-water load information here is based on limited-scale field testing combined with statistical analysis to estimate load demand or **diversity** factors (percent of total possible load that is ever actually used at one time) versus number of end use points, number of people, etc. Much of the work to provide these diversity factors dates from the 1930s to the 1960s; it remains, however, the best information currently available (with a few exceptions, as noted). Of greatest concern is the fact that most of the data from those early studies were for fixtures that used water at much higher flow rates than modern energy-efficient fixtures (e.g., low-flow shower heads and sink aerators, energy-efficient washing machines and dishwashers). Using the older load diversity information usually results in a water-heating system that adequately serves the loads, but often results in substantial oversizing. Oversizing can be a deterrent to using modern high-efficiency water-heating equipment, which may have higher first cost per unit of capacity than less efficient equipment.

Table 6.1 Typical Residential Use of Hot Water [2015A, Ch 50, Tbl 3]

Use	High Flow, Gallons/Task	Low Flow (Water Savers Used), Gallons/Task	Ultralow Flow, Gallons/Task
Food preparation	5	3	3
Hand dish washing	4	4	3
Automatic dishwasher	15	15	3 to 10
Clothes washer	32	21	5 to 15
Shower or bath	20	15	10 to 15
Face and hand washing	4	2	1 to 2

Table 6.2 HUD-FHA Minimum Water Heater Capacities for One- and Two-Family Living Units [2015A, Ch 50, Tbl 4]

Number of Baths	1 to 1.5			2 to 2.5				3 to 3.5			
Number of Bedrooms	**1**	**2**	**3**	**2**	**3**	**4**	**5**	**3**	**4**	**5**	**6**
Gas[a]											
Storage, gal	20	30	30	30	40	40	50	40	50	50	50
1000 Btu/h input	27	36	36	36	36	38	47	38	38	47	50
1 h draw, gal	43	60	60	60	70	72	90	72	82	90	92
Recovery, gph	23	30	30	30	30	32	40	32	32	40	42
Electric[a]											
Storage, gal	20	30	40	40	50	50	66	50	66	66	80
kW input	2.5	3.5	4.5	4.5	5.5	5.5	5.5	5.5	5.5	5.5	5.5
1 h draw, gal	30	44	58	58	72	72	88	72	88	88	102
Recovery, gph	10	14	18	18	22	22	22	22	22	22	22
Oil[a]											
Storage, gal	30	30	30	30	30	30	30	30	30	30	30
1000 Btu/h input	70	70	70	70	70	70	70	70	70	70	70
1 h draw, gal	89	89	89	89	89	89	89	89	89	89	89
Recovery, gph	59	59	59	59	59	59	59	59	59	59	59
Tank-Type Indirect[b,c]											
I-W-H-rated draw, gal in 3 h, 100°F rise		40	40		66	66[e]	66	66	66	66	66
Manufacturer-rated draw, gal in 3 h, 100°F rise		49	49		75	75[e]	75	75	75	75	75
Tank capacity, gal		66	66		66	66[e]	82	66	82	82	82
Tankless-Type Indirect[c,d]											
I-W-H-rated draw, gpm, 100°F rise		2.75	2.75		3.25	3.25[e]	3.75	3.25	3.75	3.75	3.75
Manufacturer-rated draw, gal in 5 min, 100°F rise		15	15		25	25[e]	35	25	35	35	35

Note: Applies to tank-type water heaters only.

[a] Storage capacity, input, and recovery requirements indicated are typical and may vary with manufacturer. Any combination of requirements to produce stated 1 h draw is satisfactory.

[b] Boiler-connected water heater capacities (180°F boiler water, internal or external connection).

[c] Heater capacities and inputs are minimum allowable. Variations in tank size are permitted when recovery is based on 4 gph/kW at 100°F rise for electrical, AGA recovery ratings for gas, and IBR ratings for steam and hot-water heaters.

[d] Boiler-connected heater capacities (200°F boiler water, internal or external connection).

[e] Also for 1 to 1.5 baths and 4 bedrooms for indirect water heaters.

Table 6.3 Overall (OVL) and Peak Average Hot-Water Use [2015A, Ch 50, Tbl 5]

Group	Average Hot-Water Use, gal							
	Hourly		Daily		Weekly		Monthly	
	OVL	Peak	OVL	Peak	OVL	Peak	OVL	Peak
All families	2.6	4.6	62.4	67.1	436	495	1897	2034
"Typical" families	2.6	5.8	63.1	66.6	442	528	1921	2078

Table 6.4 Hot-Water Demands and Use for Various Types of Buildings*
[2015A, Ch 50, Tbl 6]

Type of Building	Maximum Hourly	Maximum Daily	Average Daily
Men's dormitories	3.8 gal/student	22.0 gal/student	13.1 gal/student
Women's dormitories	5.0 gal/student	26.5 gal/student	12.3 gal/student
Motels: Number of units[a]			
20 or less	6.0 gal/unit	35.0 gal/unit	20.0 gal/unit
60	5.0 gal/unit	25.0 gal/unit	14.0 gal/unit
100 or more	4.0 gal/unit	15.0 gal/unit	10.0 gal/unit
Nursing homes	4.5 gal/bed	30.0 gal/bed	18.4 gal/bed
Office buildings	0.4 gal/person	2.0 gal/person	1.0 gal/person
Food service establishments			
Type A: Full-meal restaurants and cafeterias	1.5 gal/max meals/h	11.0 gal/max meals/day	2.4 gal/average meals/day[b]
Type B: Drive-ins, grills, luncheonettes, sandwich, and snack shops	0.7 gal/max meals/h	6.0 gal/max meals/day	0.7 gal/average meals/day[b]
Apartment houses: Number of apartments			
20 or less	12.0 gal/apartment	80.0 gal/apartment	42.0 gal/apartment
50	10.0 gal/apartment	73.0 gal/apartment	40.0 gal/apartment
75	8.5 gal/apartment	66.0 gal/apartment	38.0 gal/apartment
100	7.0 gal/apartment	60.0 gal/apartment	37.0 gal/apartment
200 or more	5.0 gal/apartment	50.0 gal/apartment	35.0 gal/apartment
Elementary schools	0.6 gal/student	1.5 gal/student	0.6 gal/student[b]
Junior and senior high schools	1.0 gal/student	3.6 gal/student	1.8 gal/student[b]

*Data predate modern low-flow fixtures and appliances.
[a]Interpolate for intermediate values.[b]Per day of operation.

Table 6.5 Hot-Water Demand per Fixture for Various Types of Buildings [2015A, Ch 50, Tbl 10]
(Gallons of water per hour per fixture, calculated at a final temperature of 140°F)

	Apartment House	Club	Gymnasium	Hospital	Hotel	Industrial Plant	Office Building	Private Residence	School	YMCA
1. Basin, private lavatory	2	2	2	2	2	2	2	2	2	2
2. Basin, public lavatory	4	6	8	6	8	12	6	—	15	8
3. Bathtub[c]	20	20	30	20	20	—	—	20	—	30
4. Dishwasher[a]	15	50-150	—	50-150	50-200	20-100	—	15	20-100	20-100
5. Foot basin	3	3	12	3	3	12	—	3	3	12
6. Kitchen sink	10	20	—	20	30	20	20	10	20	20
7. Laundry, stationary tub	20	28	—	28	28	—	—	20	—	28
8. Pantry sink	5	10	—	10	10	—	10	5	10	10
9. Shower	30	150	225	75	75	225	30	30	225	225
10. Service sink	20	20	—	20	30	20	20	15	20	20
11. Hydrotherapeutic shower				400						
12. Hubbard bath				600						
13. Leg bath				100						
14. Arm bath				35						
15. Sitz bath				30						
16. Continuous-flow bath				165						
17. Circular wash sink				20	20	30	20		30	
18. Semicircular wash sink				10	10	15	10		15	
19. DEMAND FACTOR	0.30	0.30	0.40	0.25	0.25	0.40	0.30	0.30	0.40	0.40
20. STORAGE CAPACITY FACTOR[b]	1.25	0.90	1.00	0.60	0.80	1.00	2.00	0.70	1.00	1.00

Note: Data sources predate low-flow fixtures and appliances.
[a]Dishwasher requirements should be taken from this table or from manufacturers' data for model to be used, if known.
[b]Ratio of storage tank capacity to probable maximum demand/h. Storage capacity may be reduced where unlimited supply of steam is available from central street steam system or large boiler plant.
[c]Whirlpool baths require specific consideration based on capacity. They are not included in the bathtub category.

Table 6.6 Tankless Water Heater Output Heat Rates, Btu/h* [2015A, Ch 50, Tbl 15]

Flow Rate, gpm	Temperature Rise						
	10°F	25°F	50°F	55°F	75°F	77°F	100°F
0.1	504	1,260	2,520	2,772	3,780	3,881	5,040
0.5	2,520	6,300	12,600	13,860	18,900	19,404	25,200
1.0	5,040	12,600	25,200	27,720	37,800	38,808	50,400
1.5	7,560	18,900	37,800	41,580	56,700	58,212	75,600
2.0	10,080	25,200	50,400	55,440	75,600	776,196	100,800
2.5	12,600	31,500	63,000	69,300	94,500	97,020	126,000
3.0	15,120	37,800	75,600	83,160	113,400	116,424	151,200
3.5	17,640	44,100	88,200	97,020	132,300	135,828	176,400
4.0	20,160	50,400	100,800	110,880	151,200	155,232	201,600
4.5	22,680	56,700	113,400	124,740	170,100	174,636	226,800
5.0	25,200	63,000	126,000	138,600	189,000	194,040	252,000
6.0	30,240	75,600	151,200	166,320	226,800	232,848	302,400
7.0	35,280	88,200	176,400	194,040	264,600	271,656	352,800
8.0	40,320	100,800	201,600	221,760	302,400	310,464	403,200
9.0	45,360	113,400	226,800	249,480	340,200	349,272	453,600
10.0	50,400	126,000	252,000	277,200	378,000	388,080	504,000

*Divide table values by input efficiency to determine required heat input rate.

Table 6.7 Hot-Water Demand in Fixture Units (140°F Water) [2015A, Ch 50, Tbl 16]

	Apartments	Club	Gymnasium	Hospital	Hotels and Dormitories	Industrial Plant	Office Building	School	YMCA
Basin, private lavatory	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75
Basin, public lavatory	—	1	1	1	1	1	1	1	1
Bathtub	1.5	1.5	—	1.5	1.5	—	—	—	—
Dishwasher*	1.5	Five fixture units per 250 seating capacity							
Therapeutic bath	—	—	—	5	—	—	—	—	—
Kitchen sink	0.75	1.5	—	3	1.5	3	—	0.75	3
Pantry sink	—	2.5	—	2.5	2.5	—	—	2.5	2.5
Service sink	1.5	2.5	—	2.5	2.5	2.5	2.5	2.5	2.5
Shower	1.5	1.5	1.5	1.5	1.5	3.5	—	1.5	1.5
Circular wash fountain	—	2.5	2.5	2.5	—	4	—	2.5	2.5
Semicircular wash fountain	—	1.5	1.5	1.5	—	3	—	1.5	1.5

Note: Data predate modern low-flow fixtures and appliances.

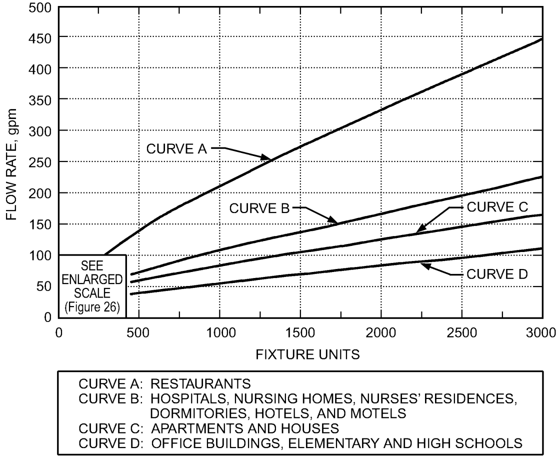


Figure 6.1 Modified Hunter Curve for Calculating Hot-Water Flow Rate [2015A, Ch 50, Fig 27]
(Data Predate Modern Low-Flow Fixtures and Appliances)

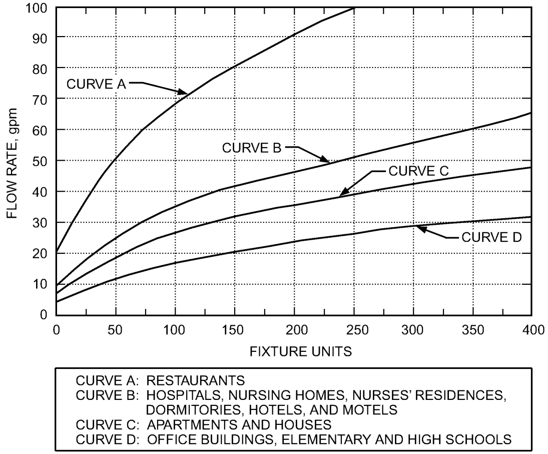


Figure 6.2 Enlarged Section of Modified Hunter Curve [2015A, Ch 50, Fig 28]
(Data Predate Modern Low-Flow Fixtures and Appliances)

Temperature Requirement

Typical temperature requirements for some services are shown in Table 6.8. A 140°F water temperature minimizes flue gas condensation in the equipment.

Other Safety Concerns

Regulatory agencies differ as to the selection of protective devices and methods of installation. It is therefore essential to check and comply with the manufacturer's instructions and the applicable local codes. In the absence of such instructions and codes, the following recommendations may be used as a guide:

- Water expands when it is heated. Although the water-heating system is initially under service pressure, the pressure rises rapidly if backflow is prevented by devices such as a check valve, pressure-reducing valve, or backflow preventer in the cold-water line or by temporarily shutting off the cold-water valve. When backflow is prevented, the pressure rise during heating may cause the safety relief valve to weep to relieve the pressure. However, if the safety relief valve is inadequate, inoperative, or missing, pressure rise may rupture the tank or cause other damage. Systems having this potential problem must be protected by a properly sized expansion tank located on the cold-water line downstream of and as close as practical to the device preventing backflow.
- Temperature-limiting devices (energy cutoff/high limit) prevent water temperatures from exceeding 210°F by stopping the flow of fuel or energy. These devices should be listed and labeled by a recognized certifying agency.
- Safety relief valves open when pressure exceeds the valve setting. These valves are typically applied to water-heating and hot-water supply boilers. The set pressure should not exceed the maximum allowable working pressure of the boiler. The heat input pressure steam rating (in Btu/h) should equal or exceed the maximum out-put rating for the boiler. The valves should comply with current applicable standards or the ASME *Boiler and Pressure Vessel Code*.
- Temperature and pressure safety relief valves also open if the water temperature reaches 210°F. These valves are typically applied to water heaters and hot-water storage tanks. The heat input temperature/steam rating (in Btu/h) should equal or exceed the heat input rating of the water heater. Combination temperature- and pressure-relief valves should be installed with the temperature-sensitive element located in the top 6 in. of the tank (i.e., where the water is hottest).
- To reduce scald hazards, discharge temperature at fixtures accessible to the occupant should not exceed 120°F. Thermostatically controlled mixing valves can be used to blend hot and cold water to maintain safe service hot-water temperatures.
- A relief valve should be installed in any part of the system containing a heat input device that can be isolated by valves. The heat input device may be solar water-heating panels, desuperheater water heaters, heat recovery devices, or similar equipment.

Table 6.8 Representative Hot-Water Temperatures [2015A, Ch 50, Tbl 19]

Use	Temperature, °F
Lavatory	
Hand washing	105
Shaving	115
Showers and tubs	110
Therapeutic baths	95
Commercial or institutional laundry, based on fabric	up to 180
Residential dish washing and laundry	140
Surgical scrubbing	110
Commercial spray-type dish washing[a]	
Single- or multiple-tank hood or rack type	
Wash	150 minimum
Final rinse	180 to 195
Single-tank conveyor type	
Wash	160 minimum
Final rinse	180 to 195
Single-tank rack or door type	
Single-temperature wash and rinse	165 minimum
Chemical sanitizing types[b]	140
Multiple-tank conveyor type	
Wash	150 minimum
Pumped rinse	160 minimum
Final rinse	180 to 195
Chemical sanitizing glass washer	
Wash	140
Rinse	75 minimum

[a]As required by NSF.
[b]See manufacturer for actual temperature required.

7. REFRIGERATION CYCLES

Refrigeration cycles transfer thermal energy from a region of low temperature T_R to one of higher temperature. Usually the higher-temperature heat sink is the ambient air or cooling water, at temperature T_0, the temperature of the surroundings.

The first and second laws of thermodynamics can be applied to individual components to determine mass and energy balances and the irreversibility of the components. This procedure is illustrated in later sections in this chapter.

Performance of a refrigeration cycle is usually described by a **coefficient of performance (COP)**, defined as the benefit of the cycle (amount of heat removed) divided by the required energy input to operate the cycle:

$$\text{COP} \equiv \frac{\text{Useful refrigerating effect}}{\text{Net energy supplied from external sources}} \tag{7.1}$$

For a mechanical vapor compression system, the net energy supplied is usually in the form of work, mechanical or electrical, and may include work to the compressor and fans or pumps. Thus,

$$\text{COP} = \frac{Q_{evap}}{W_{net}} \tag{7.2}$$

In an absorption refrigeration cycle, the net energy supplied is usually in the form of heat into the generator and work into the pumps and fans, or

$$\text{COP} = \frac{Q_{evap}}{Q_{gen} + W_{net}} \tag{7.3}$$

In many cases, work supplied to an absorption system is very small compared to the amount of heat supplied to the generator, so the work term is often neglected.

Applying the second law to an entire refrigeration cycle shows that a completely reversible cycle operating under the same conditions has the maximum possible COP. Departure of the actual cycle from an ideal reversible cycle is given by the **refrigerating efficiency**:

$$\eta_R = \frac{\text{COP}}{(\text{COP})_{rev}} \tag{7.4}$$

Heat into evaporator ${}_4Q_1 = m(h_1 - h_4)$Btu/min
Work of compression ${}_1W_2 = m(h_2 - h_1)$ with s = constant Btu/min
Heat out to condenser ${}_2Q_3 = m(h_2 - h_3)$ Btu/min
Expansion by throttling flow $h_3 = h_4$

Coefficient of performance

$$\text{COP} = \frac{{}_4Q_1}{{}_1W_2} = \frac{h_1 - h_4}{h_2 - h_1} \tag{7.5}$$

where

m	=	refrigerant flow rate, lb/min
h	=	enthalpy, Btu/lb
s	=	entropy, Btu/lb·°R

Theoretical compressor displacement, $D = m\, v_1$ ft^3/min
where v_1 = specific volume at suction, ft^3/lb.

For a given cycle, capacity in tons of refrigeration:

$$m = \frac{(\text{tons})(200 \text{ Btu/min} - \text{ton})}{h_1 - h_4} \tag{7.6}$$

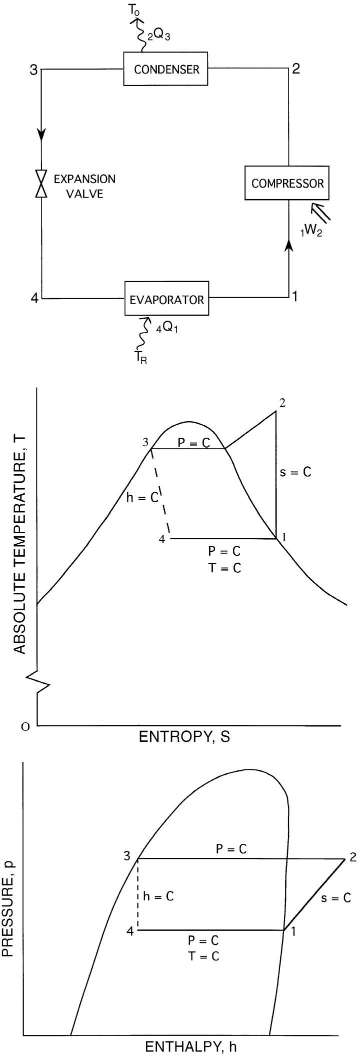


Figure 7.1 Theoretical Single-Stage Vapor Compression Refrigeration Cycle [2017F, Ch 2, Fig 8]

There are pressure drops in evaporator, condenser, and piping. There is power input to evaporator and condenser. There are heat gains and losses between refrigerant and environment. The liquid is subcooled; the suction vapor is superheated.

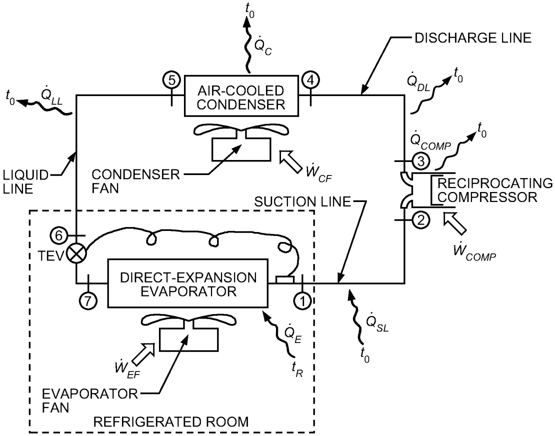


Figure 7.2 Schematic of Real, Direct-Expansion, Single-Stage Mechanical Vapor-Compression Refrigeration System [2017F, Ch 2, Fig 14]

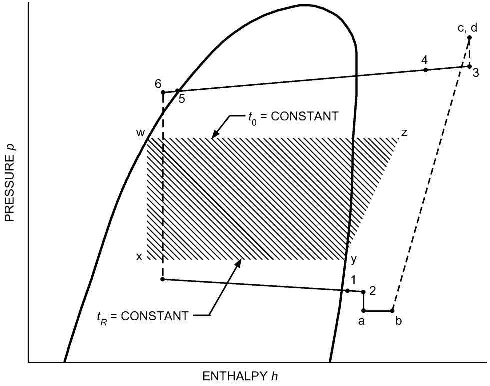


Figure 7.3 Pressure-Enthalpy Diagram of Actual System and Theoretical Single-Stage System Operating Between Same Inlet Air Temperatures t_R and t_0 [2017F, Ch 2, Fig 15]

8. REFRIGERANTS

Refrigerant Data

Table 8.1 Refrigerant Data and Safety Classifications
[2017F, Ch29, Tbl 1]

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point,[a] °F	Safety Group
Methane Series					
11	Trichlorofluoromethane	CCl_3F	137.4	75	A1
12	Dichlorodifluoromethane	CCl_2F_2	120.9	–22	A1
12B1	Bromochlorodifluoromethane	$CBrClF_2$	165.4	25	
13	Chlorotrifluoromethane	$CClF_3$	104.5	–115	A1
13B1	Bromotrifluoromethane	$CBrF_3$	148.9	–72	A1
14	Tetrafluoromethane (carbon tetrafluoride)	CF_4	88.0	–198	A1
21	Dichlorofluoromethane	$CHCl_2F$	102.9	48	B1
22	Chlorodifluoromethane	$CHClF_2$	86.5	–41	A1
23	Trifluoromethane	CHF_3	70.0	–116	A1
30	Dichloromethane (methylene chloride)	CH_2Cl_2	84.9	104	B2
31	Chlorofluoromethane	CH_2ClF	68.5	16	
32	Difluoromethane (methylene fluoride)	CH_2F_2	52.0	–62	A2L
40	Chloromethane (methyl chloride)	CH_3Cl	50.4	–12	B2
41	Fluoromethane (methyl fluoride)	CH_3F	34.0	–109	
50	Methane	CH_4	16.0	–259	A3
Ethane Series					
113	1,1,2-trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2	187.4	118	A1
114	1,2-dichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$	170.9	38	A1
115	Chloropentafluoroethane	$CClF_2CF_3$	154.5	–38	A1
116	Hexafluoroethane	CF_3CF_3	138.0	–109	A1
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	153.0	81	B1
124	2-chloro-1,1,1,2-tetrafluoroethane	$CHClFCF_3$	136.5	10	A1
125	Pentafluoroethane	CHF_2CF_3	120.0	–55	A1
134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3	102.0	–15	A1
141b	1,1-dichloro-1-fluoroethane	CH_3CCl_2F	117.0	90	
142b	1-chloro-1,1-difluoroethane	CH_3CClF_2	100.5	14	A2
143a	1,1,1-trifluoroethane	CH_3CF_3	84.0	–53	A2L
152a	1,1-difluoroethane	CH_3CHF_2	66.0	–11	A2
170	Ethane	CH_3CH_3	30.0	–128	A3
Ethers					
E170	Dimethyl ether	CH_3OCH_3	46.0	–13	A3
Propane Series					
218	Octafluoropropane	$CF_3CF_2CF_3$	188.0	–35	A1
227ea	1,1,1,2,3,3,3-heptafluoropropane	CF_3CHFCF_3	170.0	3	A1
236fa	1,1,1,3,3,3-hexafluoropropane	$CF_3CH_2CF_3$	152.0	29	A1
245fa	1,1,1,3,3-pentafluoropropane	$CF_3CH_2CHF_2$	134.0	59	B1
290	Propane	$CH_3CH_2CH_3$	44.0	–44	A3
Cyclic Organic Compounds (see 2017F, Ch 29, Tbl 2 for blends)					
C318	Octafluorocyclobutane	$–(CF_2)_4–$	200.0	21	A1

Table 8.1 Refrigerant Data and Safety Classifications
[2017F, Ch29, Tbl 1] **(*Continued*)**

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point,[a] °F	Safety Group
Miscellaneous Organic Compounds					
Hydrocarbons					
600	Butane	$CH_3CH_2CH_2CH_3$	58.1	31	A3
600a	2-methylpropane (isobutane)	$CH(CH_3)_2CH_3$	58.1	11	A3
601	Pentane	$CH_3(CH_2)_3CH_3$	72.15	97	A3
601a	2-methylbutane (isopentane)	$(CH_3)_2CHCH_2CH_3$	72.15	82	A3
Oxygen Compounds					
610	Ethyl ether	$CH_3CH_2OCH_2CH_3$	74.1	94	
611	Methyl formate	$HCOOCH_3$	60.0	89	B2
Sulfur Compounds					
620	(Reserved for future assignment)				
Nitrogen Compounds					
630	Methanamine (methyl amine)	CH_3NH_2	31.1	20	
631	Ethanamine (ethyl amine)	$CH_3CH_2(NH_2)$	45.1	62	
Inorganic Compounds					
702	Hydrogen	H_2	2.0	–423	A3
704	Helium	He	4.0	–452	A1
717	Ammonia	NH_3	17.0	–28	B2L
718	Water	H_2O	18.0	212	A1
720	Neon	Ne	20.2	–411	A1
728	Nitrogen	N_2	28.1	–320	A1
732	Oxygen	O_2	32.0	–297	
740	Argon	Ar	39.9	–303	A1
744	Carbon dioxide	CO_2	44.0	–109[c]	A1
744A	Nitrous oxide	N_2O	44.0	–129	
764	Sulfur dioxide	SO_2	64.1	14	B1
Unsaturated Organic Compounds					
1150	Ethene (ethylene)	$CH_2{=}CH_2$	28.1	–155	A3
1233zd(E)	Trans-1-chloro-3,3,3-trifluoro-1-propene	$CF_3CH{=}CHCl$	130.5	64	A1
1234yf	2,3,3,3-tetrafluoro-1-propene	$CF_3CF{=}CH_2$	114.0	–20.9	A2L
1234ze(E)	Trans-1,3,3,3-tetrafluoro-1-propene	$CF_3CH{=}CHF$	114.0	–2.2	A2L
1270	Propene (propylene)	$CH_3CH{=}CH_2$	42.1	–54	A3
1336mzz(Z)	Cis-1,1,1,4,4,4-hexafluoro-2-butene	$CF_3CH{=}CHCF_3$	164.1	92	A1

Source: ANSI/ASHRAE *Standard* 34-2010.
[a]Chemical name, chemical formula, molecular mass, and normal boiling point are not part of this standard.
[b]Preferred chemical name is followed by the popular name in parentheses.
[c]Sublimes.

The environmental effect of the chlorine in CFCs and HCFCs has resulted in CFCs no longer being manufactured and the manufacture of HCFCs being phased out.

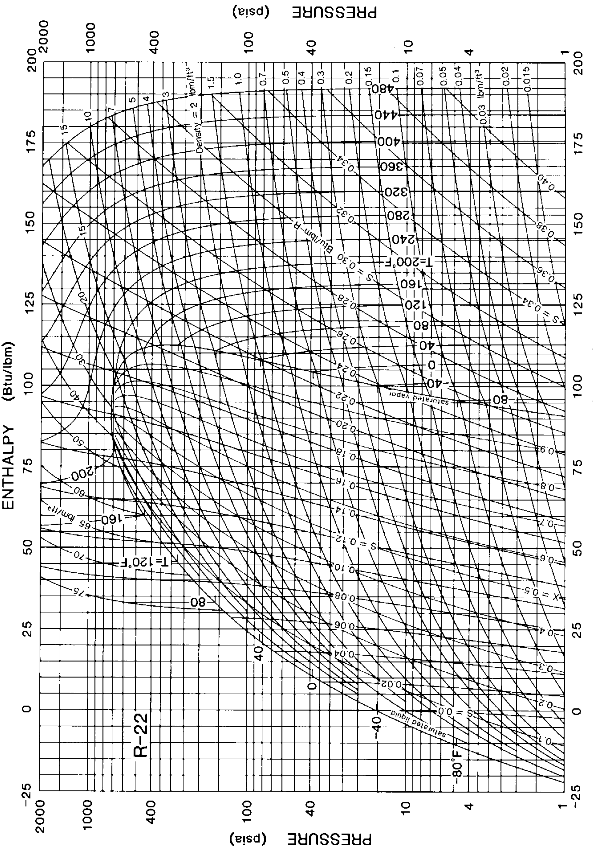


Figure 8.1 Refrigerant 22 (Chlorodifluoromethane)
Properties of Saturated Liquid and Saturated Vapor
[2017F, Ch 30, Fig 2]

Table 8.2 R-22 (Chlorodifluoromethane) Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-22]

Temp.,* °F	Pressure, psia	Density, lb/ft³ Liquid	Volume, ft³/lb Vapor	Enthalpy, Btu/lb		Entropy, Btu/lb·°F	
				Liquid	Vapor	Liquid	Vapor
–150	0.263	98.28	146.06	–28.119	87.566	–0.07757	0.29600
–140	0.436	97.36	90.759	–25.583	88.729	–0.06951	0.28808
–130	0.698	96.44	58.384	–23.046	89.899	–0.06170	0.28090
–120	1.082	95.52	38.745	–20.509	91.074	–0.05412	0.27439
–110	1.629	94.59	26.444	–17.970	92.252	–0.04675	0.26846
–100	2.388	93.66	18.511	–15.427	93.430	–0.03959	0.26307
–95	2.865	93.19	15.623	–14.154	94.018	–0.03608	0.26055
–90	3.417	92.71	13.258	–12.880	94.605	–0.03261	0.25815
–85	4.053	92.24	11.309	–11.604	95.191	–0.02918	0.25585
–80	4.782	91.76	9.6939	–10.326	95.775	–0.02580	0.25366
–75	5.615	91.28	8.3487	–9.046	96.357	–0.02245	0.25155
–70	6.561	90.79	7.2222	–7.763	96.937	–0.01915	0.24954
–65	7.631	90.31	6.2744	–6.477	97.514	–0.01587	0.24761
–60	8.836	89.82	5.4730	–5.189	98.087	–0.01264	0.24577
–55	10.190	89.33	4.7924	–3.897	98.657	–0.00943	0.24400
–50	11.703	88.83	4.2119	–2.602	99.224	–0.00626	0.24230
–45	13.390	88.33	3.7147	–1.303	99.786	–0.00311	0.24067
–41.46[b]	14.696	87.97	3.4054	–0.381	100.181	–0.00091	0.23955
–40	15.262	87.82	3.2872	0.000	100.343	0.00000	0.23910
–35	17.336	87.32	2.9181	1.308	100.896	0.00309	0.23759
–30	19.624	86.80	2.5984	2.620	101.443	0.00615	0.23615
–25	22.142	86.29	2.3204	3.937	101.984	0.00918	0.23475
–20	24.906	85.76	2.0778	5.260	102.519	0.01220	0.23341
–15	27.929	85.24	1.8656	6.588	103.048	0.01519	0.23211
–10	31.230	84.71	1.6792	7.923	103.570	0.01815	0.23086
–5	34.824	84.17	1.5150	9.263	104.085	0.02110	0.22965
0	38.728	83.63	1.3701	10.610	104.591	0.02403	0.22848
5	42.960	83.08	1.2417	11.964	105.090	0.02694	0.22735
10	47.536	82.52	1.1276	13.325	105.580	0.02983	0.22625
15	52.475	81.96	1.0261	14.694	106.061	0.03270	0.22519
20	57.795	81.39	0.9354	16.070	106.532	0.03556	0.22415
25	63.514	80.82	0.8543	17.455	106.994	0.03841	0.22315
30	69.651	80.24	0.7815	18.848	107.445	0.04124	0.22217
35	76.225	79.65	0.7161	20.250	107.884	0.04406	0.22121
40	83.255	79.05	0.6572	21.662	108.313	0.04686	0.22028
45	90.761	78.44	0.6040	23.083	108.729	0.04966	0.21936
50	98.763	77.83	0.5558	24.514	109.132	0.05244	0.21847
55	107.28	77.20	0.5122	25.956	109.521	0.05522	0.21758
60	116.33	76.57	0.4725	27.409	109.897	0.05798	0.21672
65	125.94	75.92	0.4364	28.874	110.257	0.06074	0.21586
70	136.13	75.27	0.4035	30.350	110.602	0.06350	0.21501
75	146.92	74.60	0.3734	31.839	110.929	0.06625	0.21417
80	158.33	73.92	0.3459	33.342	111.239	0.06899	0.21333
85	170.38	73.23	0.3207	34.859	111.530	0.07173	0.21250
90	183.09	72.52	0.2975	36.391	111.801	0.07447	0.21166
95	196.50	71.80	0.2762	37.938	112.050	0.07721	0.21083
100	210.61	71.06	0.2566	39.502	112.276	0.07996	0.20998
105	225.46	70.30	0.2385	41.084	112.478	0.08270	0.20913
110	241.06	69.52	0.2217	42.686	112.653	0.08545	0.20827
115	257.45	68.72	0.2062	44.308	112.799	0.08821	0.20739
120	274.65	67.90	0.1918	45.952	112.914	0.09098	0.20649
125	292.69	67.05	0.1785	47.621	112.996	0.09376	0.20557
130	311.58	66.18	0.1660	49.316	113.040	0.09656	0.20462
135	331.37	65.27	0.1544	51.041	113.043	0.09937	0.20364
140	352.08	64.32	0.1435	52.798	113.000	0.10222	0.20261
145	373.74	63.34	0.1334	54.591	112.907	0.10509	0.20153
150	396.38	62.31	0.1238	56.425	112.756	0.10800	0.20040
155	420.04	61.22	0.1149	58.305	112.539	0.11096	0.19919
160	444.75	60.07	0.1064	60.240	112.247	0.11397	0.19790
165	470.56	58.84	0.0984	62.237	111.866	0.11705	0.19650
170	497.50	57.53	0.0907	64.309	111.378	0.12022	0.19497
175	525.62	56.10	0.0834	66.474	110.760	0.12350	0.19328
180	554.98	54.52	0.0764	68.757	109.976	0.12693	0.19136
185	585.63	52.74	0.0695	71.196	108.972	0.13056	0.18916
190	617.64	50.67	0.0626	73.859	107.654	0.13450	0.18651
195	651.12	48.14	0.0556	76.875	105.835	0.13893	0.18316
200	686.20	44.68	0.0479	80.593	103.010	0.14437	0.17835
205.06[c]	723.74	32.70	0.0306	91.208	91.208	0.16012	0.16012

*Temperatures on ITS-90 scale [b]Normal boiling point [c]Critical point

Table 8.3 Superheated Vapor Thermodynamic Properties of R-22

Temp., °F	Pressure = 30 psia Sat. Temp. = –11.85°F			Pressure = 60 psia Sat. Temp. = 21.94°F		
	V	***h***	***s***	***V***	***h***	***s***
–10	1.760	103.92	0.2325			
30	1.943	109.92	0.2453	0.9271	108.35	0.2271
60	2.078	114.55	0.2545	1.001	113.17	0.2367
100	2.255	120.92	0.2663	1.096	119.74	0.2488
150	2.473	129.17	0.2804	1.212	128.19	0.2633
Temp., °F	**Pressure = 75 psia Sat. Temp. = 34.06°F**			**Pressure = 90 psia Sat. Temp. = 44.47°F**		
	V	***h***	***s***	***V***	***h***	***s***
30	0.7851	107.81	0.2229			
60	0.7847	112.45	0.2306	0.6401	111.69	0.2253
100	0.8639	119.13	0.2429	0.7088	118.50	0.2379
150	0.9591	127.69	0.2576	0.7906	127.18	0.2528
Temp., °F	**Pressure = 135 psia Sat. Temp. = 69.39°F**			**Pressure = 180 psia Sat. Temp. = 88.72°F**		
	V	***h***	***s***	***V***	***h***	***s***
100	0.4492	116.50	0.2260	0.3177	114.29	0.2164
150	0.5092	125.59	0.2416	0.3678	123.90	0.2329
200	0.5655	134.79	0.2561	0.4132	133.45	0.2479
250	0.6193	144.20	0.2698	0.4558	143.10	0.2620
300	0.6713	153.84	0.2829	0.4965	152.93	0.2754
Temp., °F	**Pressure = 200 psia Sat. Temp. = 96.17°F**			**Pressure = 220 psia Sat. Temp. = 103.09°F**		
	V	***h***	***s***	***V***	***h***	***s***
100	0.2776	113.22	0.2126			
150	0.3251	123.11	0.2295	0.2900	122.30	0.2263
200	0.3674	132.83	0.2448	0.3299	132.20	0.2419
250	0.4067	142.60	0.2591	0.3666	142.09	0.2564
300	0.4441	152.52	0.2726	0.4012	152.10	0.2700
Temp., °F	**Pressure = 240 psia Sat. Temp. = 109.57°F**			**Pressure = 260 psia Sat. Temp. = 115.66°F**		
	V	***h***	***s***	***V***	***h***	***s***
150	0.2606	121.45	0.2232	0.2356	120.58	0.2203
200	0.2985	131.56	0.2392	0.2720	130.90	0.2366
250	0.3330	141.58	0.2538	0.3046	141.06	0.2514
300	0.3654	151.69	0.2676	0.3351	151.27	0.2653

V = vapor volume, ft^3/lb
h = enthalpy, Btu/lb
s = entropy, Btu/lb·°F

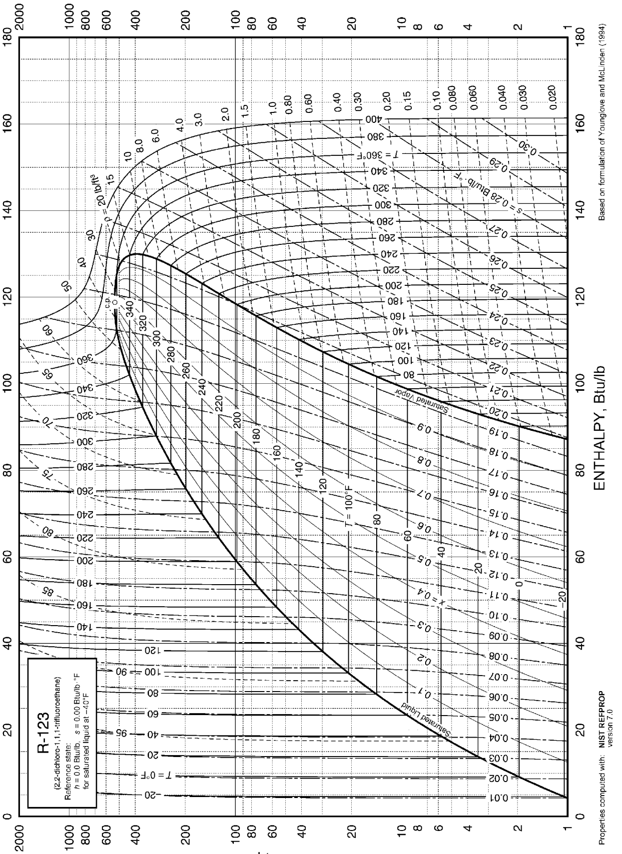


Figure 8.2 Pressure-Enthalpy Diagram for Refrigerant 123 [2017F, Ch 30, Fig 5]

Table 8.4 R-123 (2,2-Dichloro-1,1,1-Trifluoroethane) Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-123]

Temp.,* °F	Pressure, psia	Density, lb/ft³ Liquid	Volume, ft³/lb Vapor	Enthalpy, Btu/lb Liquid	Enthalpy, Btu/lb Vapor	Entropy, Btu/lb·°F Liquid	Entropy, Btu/lb·°F Vapor	c_p/c_v Vapor
–140	0.003	108.90	7431.6	–22.241	71.783	–0.06050	0.23363	1.1237
–130	0.006	108.12	3871.0	–20.033	72.974	–0.05370	0.22843	1.1212
–120	0.011	107.35	2111.6	–17.826	74.187	–0.04710	0.22379	1.1187
–110	0.020	106.57	1201.0	–15.619	75.421	–0.04070	0.21966	1.1165
–100	0.036	105.80	709.46	–13.410	76.676	–0.03447	0.21600	1.1144
–90	0.060	105.03	433.83	–11.195	77.950	–0.02840	0.21275	1.1124
–80	0.097	104.26	273.77	–8.975	79.244	–0.02247	0.20989	1.1106
–70	0.154	103.48	177.81	–6.746	80.556	–0.01668	0.20737	1.1090
–60	0.236	102.70	118.57	–4.509	81.885	–0.01101	0.20516	1.1075
–50	0.354	101.92	80.999	–2.260	83.231	–0.00545	0.20323	1.1061
–40	0.519	101.13	56.576	0.000	84.592	0.00000	0.20157	1.1050
–30	0.744	100.34	40.333	2.272	85.967	0.00535	0.20014	1.1040
–20	1.046	99.54	29.299	4.558	87.355	0.01061	0.19892	1.1032
–10	1.445	98.73	21.655	6.857	88.754	0.01578	0.19790	1.1026
0	1.963	97.92	16.264	9.170	90.163	0.02086	0.19706	1.1022
5	2.274	97.51	14.174	10.332	90.871	0.02337	0.19670	1.1021
10	2.625	97.10	12.396	11.498	91.582	0.02587	0.19638	1.1020
15	3.019	96.69	10.878	12.667	92.294	0.02834	0.19609	1.1020
20	3.460	96.28	9.5779	13.840	93.008	0.03080	0.19585	1.1020
25	3.952	95.86	8.4595	15.017	93.723	0.03324	0.19563	1.1021
30	4.499	95.44	7.4943	16.198	94.440	0.03566	0.19544	1.1023
35	5.106	95.02	6.6586	17.382	95.158	0.03806	0.19529	1.1025
40	5.778	94.60	5.9327	18.570	95.877	0.04045	0.19517	1.1028
45	6.519	94.17	5.3002	19.762	96.597	0.04282	0.19507	1.1031
50	7.334	93.74	4.7474	20.958	97.317	0.04518	0.19500	1.1035
55	8.229	93.31	4.2629	22.158	98.038	0.04752	0.19495	1.1040
60	9.208	92.88	3.8371	23.362	98.760	0.04984	0.19493	1.1046
65	10.278	92.44	3.4617	24.570	99.481	0.05215	0.19493	1.1052
70	11.445	92.01	3.1301	25.782	100.203	0.05444	0.19495	1.1059
75	12.713	91.56	2.8362	26.998	100.924	0.05673	0.19499	1.1067
80	14.090	91.12	2.5753	28.218	101.645	0.05899	0.19505	1.1075
82.08[b]	14.696	90.94	2.4753	28.728	101.945	0.05993	0.19508	1.1079
85	15.580	90.67	2.3429	29.443	102.365	0.06124	0.19513	1.1085
90	17.192	90.22	2.1356	30.671	103.085	0.06348	0.19522	1.1095
95	18.931	89.77	1.9503	31.904	103.804	0.06571	0.19534	1.1106
100	20.804	89.31	1.7841	33.141	104.521	0.06792	0.19546	1.1119
105	22.819	88.85	1.6349	34.383	105.238	0.07012	0.19560	1.1132
110	24.980	88.39	1.5006	35.628	105.953	0.07231	0.19576	1.1146
115	27.297	87.92	1.3795	36.879	106.666	0.07449	0.19593	1.1162
120	29.776	87.45	1.2701	38.134	107.377	0.07665	0.19611	1.1178
125	32.425	86.98	1.1710	39.393	108.086	0.07881	0.19630	1.1196
130	35.251	86.50	1.0812	40.657	108.792	0.08095	0.19650	1.1215
135	38.261	86.01	0.9996	41.926	109.497	0.08308	0.19671	1.1236
140	41.464	85.52	0.9253	43.200	110.198	0.08520	0.19693	1.1258
145	44.868	85.03	0.8577	44.479	110.896	0.08732	0.19716	1.1281
150	48.479	84.53	0.7959	45.763	111.591	0.08942	0.19739	1.1306
160	56.360	83.52	0.6876	48.347	112.970	0.09359	0.19788	1.1362
170	65.173	82.49	0.5965	50.953	114.333	0.09773	0.19839	1.1426
180	74.986	81.43	0.5195	53.583	115.678	0.10184	0.19892	1.1499
190	85.868	80.34	0.4539	56.237	117.001	0.10592	0.19945	1.1583
200	97.892	79.23	0.3979	58.918	118.300	0.10997	0.19999	1.1681
210	111.13	78.08	0.3497	61.627	119.572	0.11400	0.20053	1.1793
220	125.66	76.89	0.3080	64.367	120.813	0.11801	0.20106	1.1925
230	141.56	75.66	0.2719	67.141	122.019	0.12201	0.20158	1.2079
240	158.91	74.38	0.2404	69.952	123.184	0.12599	0.20207	1.2262
250	177.80	73.04	0.2128	72.805	124.303	0.12997	0.20254	1.2482
260	198.31	71.64	0.1885	75.704	125.367	0.13396	0.20296	1.2749
270	220.53	70.16	0.1670	78.655	126.368	0.13795	0.20334	1.3079
280	244.58	68.60	0.1479	81.666	127.294	0.14196	0.20365	1.3496
290	270.54	66.92	0.1309	84.749	128.128	0.14600	0.20387	1.4035
300	298.53	65.11	0.1155	87.916	128.851	0.15010	0.20398	1.4755
310	328.69	63.12	0.1016	91.188	129.431	0.15426	0.20395	1.5762
320	361.16	60.91	0.0889	94.594	129.822	0.15853	0.20372	1.7258
330	396.11	58.37	0.0770	98.186	129.950	0.16297	0.20320	1.9693
340	433.76	55.33	0.0658	102.059	129.670	0.16769	0.20222	2.4318
350	474.41	51.32	0.0544	106.459	128.628	0.17298	0.20036	3.6383
360	518.66	43.97	0.0403	112.667	125.064	0.18039	0.19551	14.6330
362.63[c]	531.10	34.34	0.0291	118.800	118.800	0.18779	0.18779	∞

*Temperatures on ITS-90 scale   [b]Normal boiling point   [c]Critical point

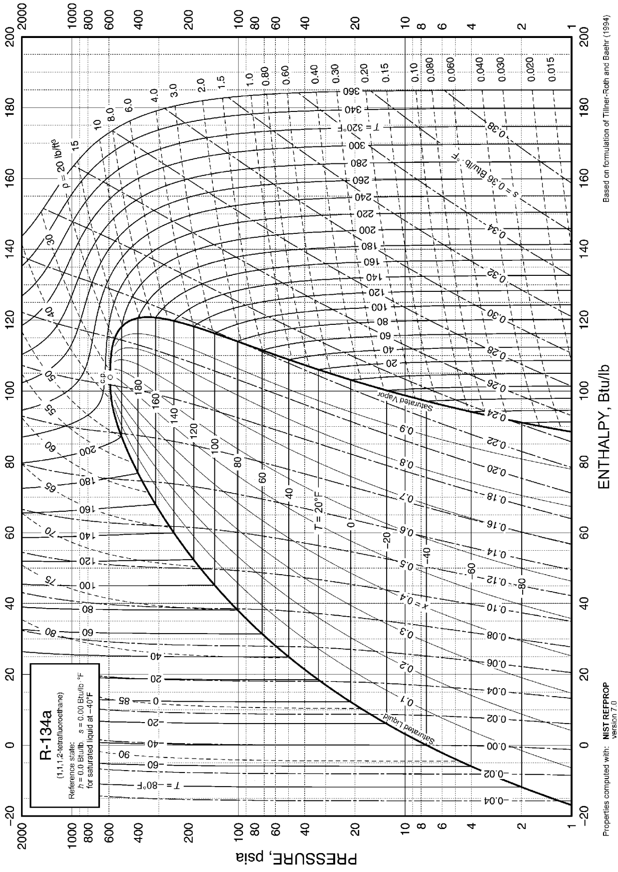


Figure 8.3 Pressure-Enthalpy Diagram for Refrigerant 134a [2017F, Ch 30, Fig 8]

Table 8.5 R-134a (1,1,1,2-Tetrafluoroethane) Properties of Saturated Liquid and Saturated Vapor

[2017F, Ch 30, Tbl R-134a]

Temp.,* °F	Pressure, psia	Density, lb/ft^3	Volume, ft^3/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F		Specific Heat c_p, Btu/lb·°F		c_p/c_v
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor
–153.94[a]	0.057	99.33	568.59	–32.992	80.362	–0.09154	0.27923	0.2829	0.1399	1.1637
–150	0.072	98.97	452.12	–31.878	80.907	–0.08791	0.27629	0.2830	0.1411	1.1623
–140	0.129	98.05	260.63	–29.046	82.304	–0.07891	0.26941	0.2834	0.1443	1.1589
–130	0.221	97.13	156.50	–26.208	83.725	–0.07017	0.26329	0.2842	0.1475	1.1559
–120	0.365	96.20	97.481	–23.360	85.168	–0.06166	0.25784	0.2853	0.1508	1.1532
–110	0.583	95.27	62.763	–20.500	86.629	–0.05337	0.25300	0.2866	0.1540	1.1509
–100	0.903	94.33	41.637	–17.626	88.107	–0.04527	0.24871	0.2881	0.1573	1.1490
–90	1.359	93.38	28.381	–14.736	89.599	–0.03734	0.24490	0.2898	0.1607	1.1475
–80	1.993	92.42	19.825	–11.829	91.103	–0.02959	0.24152	0.2916	0.1641	1.1465
–75	2.392	91.94	16.711	–10.368	91.858	–0.02577	0.23998	0.2925	0.1658	1.1462
–70	2.854	91.46	14.161	–8.903	92.614	–0.02198	0.23854	0.2935	0.1676	1.1460
–65	3.389	90.97	12.060	–7.432	93.372	–0.01824	0.23718	0.2945	0.1694	1.1459
–60	4.002	90.49	10.321	–5.957	94.131	–0.01452	0.23590	0.2955	0.1713	1.1460
–55	4.703	90.00	8.8733	–4.476	94.890	–0.01085	0.23470	0.2965	0.1731	1.1462
–50	5.501	89.50	7.6621	–2.989	95.650	–0.00720	0.23358	0.2976	0.1751	1.1466
–45	6.406	89.00	6.6438	–1.498	96.409	–0.00358	0.23252	0.2987	0.1770	1.1471
–40	7.427	88.50	5.7839	0.000	97.167	0.00000	0.23153	0.2999	0.1790	1.1478
–35	8.576	88.00	5.0544	1.503	97.924	0.00356	0.23060	0.3010	0.1811	1.1486
–30	9.862	87.49	4.4330	3.013	98.679	0.00708	0.22973	0.3022	0.1832	1.1496
–25	11.299	86.98	3.9014	4.529	99.433	0.01058	0.22892	0.3035	0.1853	1.1508
–20	12.898	86.47	3.4449	6.051	100.184	0.01406	0.22816	0.3047	0.1875	1.1521
–15	14.671	85.95	3.0514	7.580	100.932	0.01751	0.22744	0.3060	0.1898	1.1537
–14.93[b]	14.696	85.94	3.0465	7.600	100.942	0.01755	0.22743	0.3061	0.1898	1.1537
–10	16.632	85.43	2.7109	9.115	101.677	0.02093	0.22678	0.3074	0.1921	1.1554
–5	18.794	84.90	2.4154	10.657	102.419	0.02433	0.22615	0.3088	0.1945	1.1573
0	21.171	84.37	2.1579	12.207	103.156	0.02771	0.22557	0.3102	0.1969	1.1595
5	23.777	83.83	1.9330	13.764	103.889	0.03107	0.22502	0.3117	0.1995	1.1619
10	26.628	83.29	1.7357	15.328	104.617	0.03440	0.22451	0.3132	0.2021	1.1645
15	29.739	82.74	1.5623	16.901	105.339	0.03772	0.22403	0.3147	0.2047	1.1674
20	33.124	82.19	1.4094	18.481	106.056	0.04101	0.22359	0.3164	0.2075	1.1705
25	36.800	81.63	1.2742	20.070	106.767	0.04429	0.22317	0.3181	0.2103	1.1740
30	40.784	81.06	1.1543	21.667	107.471	0.04755	0.22278	0.3198	0.2132	1.1777
35	45.092	80.49	1.0478	23.274	108.167	0.05079	0.22241	0.3216	0.2163	1.1818
40	49.741	79.90	0.9528	24.890	108.856	0.05402	0.22207	0.3235	0.2194	1.1862
45	54.749	79.32	0.8680	26.515	109.537	0.05724	0.22174	0.3255	0.2226	1.1910
50	60.134	78.72	0.7920	28.150	110.209	0.06044	0.22144	0.3275	0.2260	1.1961
55	65.913	78.11	0.7238	29.796	110.871	0.06362	0.22115	0.3297	0.2294	1.2018
60	72.105	77.50	0.6625	31.452	111.524	0.06680	0.22088	0.3319	0.2331	1.2079
65	78.729	76.87	0.6072	33.120	112.165	0.06996	0.22062	0.3343	0.2368	1.2145
70	85.805	76.24	0.5572	34.799	112.796	0.07311	0.22037	0.3368	0.2408	1.2217
75	93.351	75.59	0.5120	36.491	113.414	0.07626	0.22013	0.3394	0.2449	1.2296
80	101.39	74.94	0.4710	38.195	114.019	0.07939	0.21989	0.3422	0.2492	1.2382
85	109.93	74.27	0.4338	39.913	114.610	0.08252	0.21966	0.3451	0.2537	1.2475
90	119.01	73.58	0.3999	41.645	115.186	0.08565	0.21944	0.3482	0.2585	1.2578
95	128.65	72.88	0.3690	43.392	115.746	0.08877	0.21921	0.3515	0.2636	1.2690
100	138.85	72.17	0.3407	45.155	116.289	0.09188	0.21898	0.3551	0.2690	1.2813
105	149.65	71.44	0.3148	46.934	116.813	0.09500	0.21875	0.3589	0.2747	1.2950
110	161.07	70.69	0.2911	48.731	117.317	0.09811	0.21851	0.3630	0.2809	1.3101
115	173.14	69.93	0.2693	50.546	117.799	0.10123	0.21826	0.3675	0.2875	1.3268
120	185.86	69.14	0.2493	52.382	118.258	0.10435	0.21800	0.3723	0.2948	1.3456
125	199.28	68.32	0.2308	54.239	118.690	0.10748	0.21772	0.3775	0.3026	1.3666
130	213.41	67.49	0.2137	56.119	119.095	0.11062	0.21742	0.3833	0.3112	1.3903
135	228.28	66.62	0.1980	58.023	119.468	0.11376	0.21709	0.3897	0.3208	1.4173
140	243.92	65.73	0.1833	59.954	119.807	0.11692	0.21673	0.3968	0.3315	1.4481
145	260.36	64.80	0.1697	61.915	120.108	0.12010	0.21634	0.4048	0.3435	1.4837
150	277.61	63.83	0.1571	63.908	120.366	0.12330	0.21591	0.4138	0.3571	1.5250
155	295.73	62.82	0.1453	65.936	120.576	0.12653	0.21542	0.4242	0.3729	1.5738
160	314.73	61.76	0.1343	68.005	120.731	0.12979	0.21488	0.4362	0.3914	1.6318
165	334.65	60.65	0.1239	70.118	120.823	0.13309	0.21426	0.4504	0.4133	1.7022
170	355.53	59.47	0.1142	72.283	120.842	0.13644	0.21356	0.4675	0.4400	1.7889
175	377.41	58.21	0.1051	74.509	120.773	0.13985	0.21274	0.4887	0.4733	1.8984
180	400.34	56.86	0.0964	76.807	120.598	0.14334	0.21180	0.5156	0.5159	2.0405
185	424.36	55.38	0.0881	79.193	120.294	0.14693	0.21069	0.5512	0.5729	2.2321
190	449.52	53.76	0.0801	81.692	119.822	0.15066	0.20935	0.6012	0.6532	2.5041
195	475.91	51.91	0.0724	84.343	119.123	0.15459	0.20771	0.6768	0.7751	2.9192
200	503.59	49.76	0.0647	87.214	118.097	0.15880	0.20562	0.8062	0.9835	3.6309
205	532.68	47.08	0.0567	90.454	116.526	0.16353	0.20275	1.0830	1.4250	5.1360
210	563.35	43.20	0.0477	94.530	113.746	0.16945	0.19814	2.1130	3.0080	10.5120
213.91[c]	588.75	31.96	0.0313	103.894	103.894	0.18320	0.18320	∞	∞	∞

*Temperatures on ITS-90 scale [a]Triple point [b]Normal boiling point [c]Critical point

Refrigerants

Table 8.6 Superheated Vapor Thermodynamic Properties of R-134a

Temp, °F	Pressure = 15 psia Sat. temp. = –14.25°F			Pressure = 30 psia Sat. temp. = 15.39°F		
	V	*h*	*s*	*V*	*h*	*s*
0	3.118	103.35	0.2324			
20	3.268	107.07	0.2403	1.584	106.18	0.2255
40	3.417	110.88	0.2481	1.663	110.06	0.2335
60	3.565	114.79	0.2558	1.741	114.03	0.2413
80	3.712	118.79	0.2633	1.818	118.08	0.2489
100	3.858	122.87	0.2708	1.895	122.22	0.2564
120	4.004	127.05	0.2781	1.971	126.44	0.2638
140	4.149	131.31	0.2853	2.046	130.75	0.2711
Temp, °F	**Pressure = 45 psia Sat. temp. = 34.94°F**			**Pressure = 60 psia Sat. temp. = 49.94°F**		
	V	*h*	*s*	*V*	*h*	*s*
40	1.077	109.20	0.2243			
60	1.132	113.24	0.2323	0.8269	112.41	0.2255
80	1.187	117.36	0.2400	0.8699	116.60	0.2334
100	1.240	121.55	0.2477	0.9120	120.86	0.2412
120	1.293	125.82	0.2552	0.9533	125.18	0.2488
140	1.345	130.17	0.2625	0.9940	129.58	0.2562
Temp, °F	**Pressure = 150 psia Sat. temp. = 105.14°F**			**Pressure = 200 psia Sat. temp. = 125.19°F**		
	V	*h*	*s*	*V*	*h*	*s*
125	0.3433	122.06	0.2274			
150	0.3692	128.08	0.2375	0.2596	125.69	0.2289
175	0.3937	134.13	0.2472	0.2807	132.07	0.2391
200	0.4171	140.23	0.2566	0.3003	138.42	0.2489
225	0.4397	146.41	0.2658	0.3189	144.80	0.2584
250	0.4616	152.66	0.2748	0.3366	151.23	0.2676
Temp, °F	**Pressure = 250 psia Sat. temp. = 141.79°F**			**Pressure = 300 psia Sat. temp. = 156.07°F**		
	V	*h*	*s*	*V*	*h*	*s*
150	0.1920	122.93	0.2210			
175	0.2118	129.79	0.2320	0.1646	127.20	0.2252
200	0.2295	136.47	0.2423	0.1817	134.35	0.2362
225	0.2460	143.10	0.2522	0.1969	141.29	0.2466
250	0.2614	149.73	0.2617	0.2110	148.15	0.2564
275	0.2761	156.37	0.2709	0.2242	154.99	0.2659
300	0.2902	163.07	0.2798	0.2367	161.84	0.2750

V = vapor volume, ft^3/lb
h = enthalpy, Btu/lb
s = entropy, Btu/lb·°F

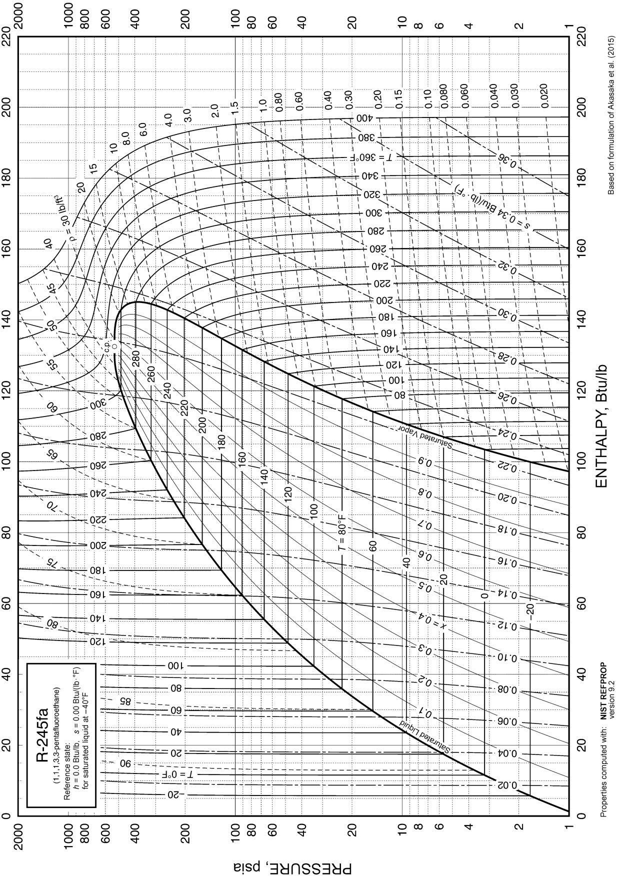


Figure 8.4 Pressure-Enthalpy Diagram for Refrigerant 245fa [2017F, Ch 30, Fig 11]

Table 8.7 Refrigerant 245fa (1,1,1,3,3-Pentafluoropropane) Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-245fa]

Temp., °F	Pressure, psia	Density, lb/ft³ Liquid	Volume, ft³/lb Vapor	Enthalpy, Btu/lb Liquid	Enthalpy, Btu/lb Vapor	Entropy, Btu/lb·°F Liquid	Entropy, Btu/lb·°F Vapor	Specific Heat c_p, Btu/lb·°F Liquid	Specific Heat c_p, Btu/lb·°F Vapor	c_p/c_v Vapor
–50	0.578	94.55	56.518	–2.840	94.771	–0.00685	0.23142	0.2830	0.1734	1.0958
–45	0.702	94.14	47.084	–1.422	95.622	–0.00341	0.23062	0.2840	0.1748	1.0953
–40	0.847	93.74	39.436	–0.000	96.478	–0.00000	0.22989	0.2849	0.1763	1.0949
–35	1.018	93.32	33.201	1.427	97.337	0.00338	0.22923	0.2859	0.1778	1.0945
–30	1.216	92.91	28.090	2.859	98.201	0.00673	0.22863	0.2869	0.1792	1.0942
–25	1.446	92.50	23.879	4.296	99.069	0.01006	0.22809	0.2879	0.1807	1.0939
–20	1.711	92.08	20.391	5.739	99.940	0.01335	0.22761	0.2889	0.1822	1.0937
–15	2.015	91.67	17.489	7.186	100.815	0.01663	0.22718	0.2900	0.1837	1.0936
–10	2.363	91.25	15.062	8.640	101.692	0.01987	0.22681	0.2911	0.1853	1.0935
–5	2.759	90.83	13.024	10.099	102.573	0.02310	0.22649	0.2923	0.1868	1.0934
0	3.208	90.40	11.305	11.563	103.456	0.02630	0.22621	0.2934	0.1884	1.0935
5	3.717	89.98	9.8498	13.034	104.342	0.02948	0.22598	0.2946	0.1900	1.0936
10	4.289	89.55	8.6119	14.511	105.230	0.03264	0.22579	0.2958	0.1916	1.0937
15	4.931	89.12	7.5553	15.994	106.120	0.03578	0.22565	0.2971	0.1933	1.0940
20	5.650	88.69	6.6500	17.483	107.011	0.03890	0.22554	0.2983	0.1949	1.0943
25	6.452	88.26	5.8715	18.979	107.904	0.04199	0.22547	0.2996	0.1966	1.0947
30	7.343	87.82	5.1998	20.481	108.798	0.04507	0.22544	0.3009	0.1983	1.0951
35	8.332	87.38	4.6183	21.990	109.694	0.04814	0.22543	0.3023	0.2001	1.0957
40	9.424	86.94	4.1132	23.506	110.590	0.05118	0.22546	0.3037	0.2018	1.0963
45	10.629	86.49	3.6731	25.029	111.486	0.05421	0.22552	0.3051	0.2036	1.0970
50	11.954	86.04	3.2885	26.560	112.383	0.05722	0.22561	0.3065	0.2055	1.0978
55	13.407	85.59	2.9515	28.097	113.280	0.06022	0.22573	0.3079	0.2073	1.0987
59.09[b]	14.696	85.22	2.7065	29.359	114.013	0.06265	0.22584	0.3092	0.2089	1.0995
60	14.997	85.13	2.6552	29.642	114.176	0.06320	0.22587	0.3094	0.2092	1.0997
65	16.733	84.67	2.3941	31.195	115.072	0.06616	0.22603	0.3109	0.2112	1.1008
70	18.624	84.21	2.1634	32.755	115.968	0.06911	0.22622	0.3125	0.2131	1.1021
75	20.679	83.75	1.9590	34.323	116.862	0.07205	0.22643	0.3140	0.2151	1.1034
80	22.909	83.27	1.7775	35.899	117.755	0.07498	0.22666	0.3156	0.2172	1.1048
85	25.322	82.80	1.6159	37.483	118.646	0.07789	0.22690	0.3173	0.2193	1.1064
90	27.930	82.32	1.4717	39.076	119.536	0.08079	0.22717	0.3189	0.2214	1.1082
95	30.742	81.84	1.3427	40.677	120.423	0.08368	0.22745	0.3206	0.2236	1.1100
100	33.769	81.35	1.2271	42.287	121.308	0.08655	0.22775	0.3224	0.2258	1.1120
105	37.022	80.85	1.1233	43.905	122.190	0.08942	0.22806	0.3241	0.2281	1.1142
110	40.513	80.35	1.0299	45.533	123.069	0.09228	0.22838	0.3260	0.2304	1.1166
115	44.251	79.85	0.9456	47.169	123.945	0.09512	0.22872	0.3278	0.2328	1.1192
120	48.249	79.33	0.8695	48.815	124.816	0.09796	0.22907	0.3297	0.2353	1.1219
125	52.518	78.82	0.8005	50.471	125.684	0.10078	0.22943	0.3317	0.2378	1.1249
130	57.071	78.29	0.7380	52.137	126.546	0.10360	0.22979	0.3337	0.2405	1.1282
135	61.918	77.76	0.6812	53.812	127.404	0.10641	0.23016	0.3358	0.2431	1.1317
140	67.074	77.22	0.6295	55.498	128.255	0.10922	0.23054	0.3379	0.2459	1.1355
145	72.550	76.67	0.5824	57.195	129.101	0.11201	0.23093	0.3401	0.2488	1.1396
150	78.358	76.12	0.5393	58.902	129.939	0.11480	0.23132	0.3424	0.2518	1.1440
155	84.512	75.55	0.5000	60.621	130.771	0.11758	0.23171	0.3447	0.2549	1.1489
160	91.025	74.98	0.4639	62.351	131.594	0.12036	0.23210	0.3472	0.2581	1.1542
165	97.911	74.39	0.4308	64.093	132.409	0.12313	0.23250	0.3497	0.2615	1.1599
170	105.18	73.80	0.4003	65.847	133.214	0.12590	0.23289	0.3524	0.2650	1.1662
175	112.85	73.19	0.3723	67.614	134.009	0.12867	0.23328	0.3551	0.2688	1.1731
180	120.94	72.58	0.3465	69.395	134.793	0.13143	0.23367	0.3580	0.2727	1.1806
185	129.45	71.95	0.3227	71.189	135.565	0.13419	0.23405	0.3611	0.2768	1.1889
190	138.41	71.30	0.3007	72.997	136.324	0.13695	0.23442	0.3643	0.2812	1.1981
195	147.82	70.64	0.2803	74.820	137.068	0.13970	0.23479	0.3677	0.2859	1.2082
200	157.71	69.97	0.2614	76.659	137.797	0.14246	0.23514	0.3713	0.2910	1.2194
205	168.08	69.28	0.2438	78.514	138.508	0.14522	0.23548	0.3752	0.2964	1.2320
210	178.96	68.57	0.2275	80.386	139.200	0.14798	0.23581	0.3794	0.3023	1.2460
215	190.36	67.84	0.2123	82.277	139.872	0.15075	0.23612	0.3838	0.3087	1.2617
220	202.30	67.08	0.1981	84.187	140.521	0.15352	0.23641	0.3887	0.3157	1.2795
225	214.80	66.31	0.1849	86.117	141.145	0.15630	0.23667	0.3940	0.3234	1.2998
230	227.87	65.50	0.1725	88.070	141.742	0.15909	0.23691	0.3998	0.3320	1.3229
235	241.53	64.67	0.1609	90.047	142.307	0.16189	0.23712	0.4062	0.3417	1.3496
240	255.81	63.80	0.1500	92.049	142.837	0.16470	0.23729	0.4134	0.3526	1.3806
245	270.72	62.90	0.1398	94.080	143.329	0.16753	0.23742	0.4215	0.3652	1.4169
250	286.28	61.96	0.1301	96.141	143.777	0.17038	0.23751	0.4307	0.3798	1.4599
260	319.47	59.92	0.1124	100.372	144.514	0.17616	0.23750	0.4540	0.4174	1.5741
270	355.57	57.63	0.0964	104.780	144.980	0.18209	0.23718	0.4874	0.4735	1.7498
280	394.83	54.97	0.0818	109.431	145.063	0.18824	0.23641	0.5410	0.5663	2.0476
290	437.52	51.72	0.0680	114.455	144.549	0.19479	0.23493	0.6449	0.7492	2.6453
300	484.03	47.29	0.0543	120.218	142.886	0.20219	0.23203	0.962	1.292	4.439
308.95[c]	529.53	32.43	0.0308	132.508	132.508	0.21798	0.21798	∞	∞	∞

*Temperatures on ITS-90 scale [b]Normal boiling point [c]Critical point

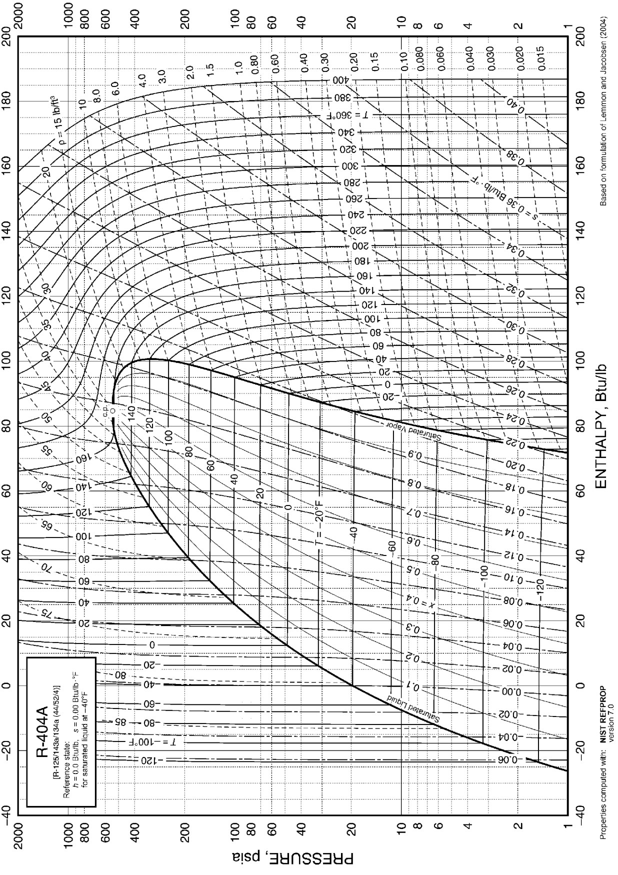


Figure 8.5 Pressure-Enthalpy Diagram for Refrigerant 404A [2017F, Ch 30, Fig 15]

Table 8.8 R-404A [R-125/143a/134a (44/52/4)]
Properties of Liquid on Bubble Line and Vapor on Dew Line [2017F, Ch 30, Tbl R-404A]

Pressure,	Temperature,* °F		Density, lb/ft³	Volume, ft³/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F	
psia	Bubble	Dew	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
1	–129.56	–127.50	89.61	36.2311	–26.33	71.76	–0.07039	0.22616
1.5	–120.05	–118.11	88.64	24.7754	–23.56	73.11	–0.06215	0.22201
2	–112.90	–111.03	87.92	18.9245	–21.49	74.14	–0.05611	0.21920
2.5	–107.10	–105.29	87.33	15.3578	–19.81	74.98	–0.05129	0.21710
3	–102.18	–100.42	86.83	12.9493	–18.38	75.69	–0.04727	0.21544
4	–94.08	–92.40	86.01	9.8941	–16.02	76.86	–0.04076	0.21292
5	–87.49	–85.87	85.33	8.0300	–14.10	77.82	–0.03555	0.21106
6	–81.89	–80.32	84.76	6.7705	–12.46	78.64	–0.03119	0.20960
7	–77.00	–75.46	84.25	5.8607	–11.02	79.35	–0.02742	0.20841
8	–72.64	–71.14	83.80	5.1716	–9.74	79.98	–0.02409	0.20741
10	–65.08	–63.64	83.01	4.1954	–7.51	81.07	–0.01839	0.20581
12	–58.65	–57.25	82.34	3.5353	–5.60	82.00	–0.01360	0.20457
14	–53.01	–51.65	81.74	3.0582	–3.91	82.81	–0.00944	0.20357
14.7[b]	–51.20	–49.85	81.55	2.9217	–3.37	83.07	–0.00812	0.20326
16	–47.98	–46.65	81.20	2.6968	–2.41	83.53	–0.00577	0.20273
18	–43.42	–42.11	80.71	2.4132	–1.03	84.18	–0.00246	0.20203
20	–39.24	–37.96	80.26	2.1845	0.23	84.78	0.00055	0.20141
22	–35.37	–34.11	79.83	1.9960	1.40	85.32	0.00332	0.20088
24	–31.77	–30.53	79.44	1.8379	2.50	85.83	0.00588	0.20041
26	–28.39	–27.17	79.06	1.7033	3.53	86.30	0.00827	0.19998
28	–25.21	–24.01	78.71	1.5873	4.51	86.75	0.01051	0.19960
30	–22.20	–21.02	78.37	1.4863	5.44	87.16	0.01263	0.19925
32	–19.34	–18.17	78.05	1.3974	6.32	87.56	0.01463	0.19894
34	–16.62	–15.46	77.74	1.3187	7.16	87.93	0.01653	0.19864
36	–14.01	–12.87	77.44	1.2484	7.97	88.29	0.01834	0.19838
38	–11.52	–10.39	77.15	1.1852	8.75	88.62	0.02007	0.19813
40	–9.12	–8.01	76.87	1.1281	9.50	88.95	0.02172	0.19790
42	–6.81	–5.71	76.60	1.0763	10.22	89.26	0.02331	0.19768
44	–4.59	–3.50	76.34	1.0290	10.92	89.56	0.02484	0.19748
46	–2.44	–1.36	76.09	0.9857	11.60	89.84	0.02632	0.19729
48	–0.36	0.71	75.84	0.9459	12.25	90.12	0.02774	0.19711
50	1.65	2.71	75.60	0.9091	12.89	90.38	0.02911	0.19694
55	6.43	7.47	75.03	0.8285	14.41	91.01	0.03237	0.19655
60	10.89	11.90	74.48	0.7609	15.84	91.58	0.03539	0.19621
65	15.07	16.07	73.97	0.7033	17.19	92.11	0.03822	0.19590
70	19.02	20.00	73.47	0.6537	18.47	92.61	0.04088	0.19562
75	22.76	23.72	72.99	0.6104	19.69	93.07	0.04339	0.19537
80	26.32	27.27	72.54	0.5724	20.86	93.50	0.04578	0.19514
85	29.71	30.64	72.09	0.5387	21.98	93.91	0.04804	0.19492
90	32.96	33.88	71.67	0.5085	23.05	94.30	0.05021	0.19471
95	36.07	36.98	71.25	0.4815	24.09	94.66	0.05229	0.19452
100	39.07	39.96	70.84	0.4570	25.10	95.00	0.05428	0.19434
110	44.73	45.60	70.06	0.4145	27.01	95.64	0.05804	0.19400
120	50.02	50.86	69.32	0.3789	28.82	96.21	0.06155	0.19368
130	54.99	55.81	68.60	0.3485	30.53	96.73	0.06485	0.19338
140	59.68	60.48	67.90	0.3222	32.16	97.20	0.06795	0.19309
150	64.13	64.91	67.23	0.2994	33.73	97.62	0.07090	0.19281
160	68.36	69.13	66.57	0.2793	35.23	98.01	0.07371	0.19253
170	72.40	73.15	65.93	0.2614	36.68	98.37	0.07639	0.19226
180	76.26	76.99	65.30	0.2454	38.08	98.69	0.07896	0.19198
190	79.97	80.68	64.68	0.2311	39.44	98.98	0.08143	0.19170
200	83.53	84.23	64.07	0.2181	40.76	99.25	0.08381	0.19143
220	90.27	90.94	62.87	0.1955	43.29	99.70	0.08833	0.19085
240	96.57	97.21	61.70	0.1764	45.70	100.05	0.09259	0.19026
260	102.48	103.09	60.53	0.1601	48.02	100.32	0.09663	0.18962
280	108.06	108.64	59.37	0.1460	50.25	100.51	0.10047	0.18895
300	113.34	113.90	58.20	0.1336	52.42	100.61	0.10417	0.18823
320	118.36	118.89	57.03	0.1226	54.54	100.64	0.10773	0.18745
340	123.14	123.65	55.83	0.1127	56.61	100.58	0.11118	0.18660
360	127.71	128.19	54.61	0.1038	58.65	100.43	0.11456	0.18566
380	132.09	132.54	53.35	0.0956	60.67	100.20	0.11787	0.18464
400	136.28	136.71	52.03	0.0881	62.68	99.85	0.12114	0.18349
450	146.07	146.42	48.36	0.0713	67.80	98.42	0.12934	0.17987
500	154.97	155.22	43.51	0.0556	73.49	95.51	0.13833	0.17416
548.24[c]	162.50	162.50	35.84	0.0279	80.85	80.85	0.14987	0.14987

*Temperatures on ITS-90 scale [b]Bubble and dew points at one standard atmosphere [c]Critical point

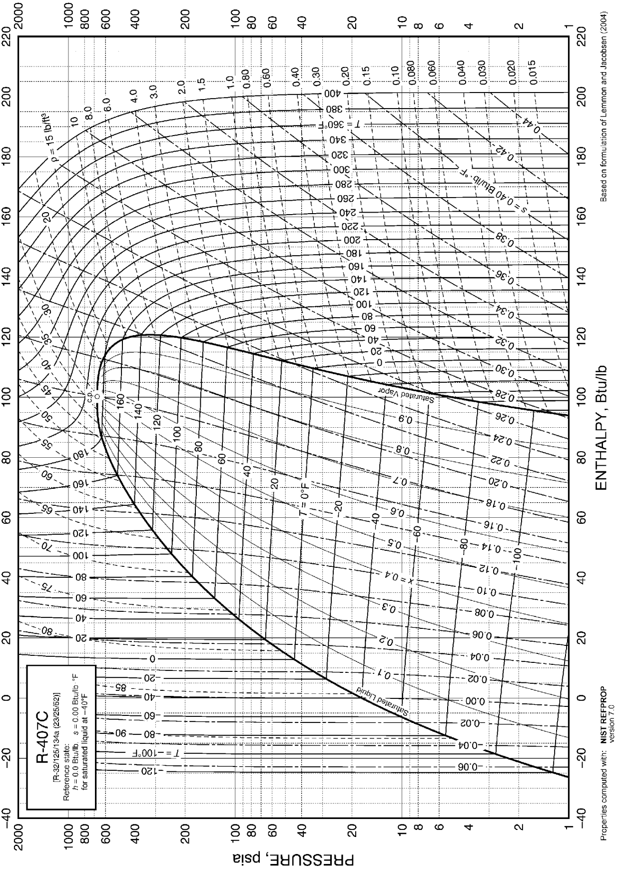


Figure 8.6 Pressure-Enthalpy Diagram for Refrigerant 407C [2017F, Ch 30, Fig 16]

Table 8.9 R-407C [R-32/125/134a (23/25/52)] Properties of Liquid on Bubble Line and Vapor on Dew Line [2017F, Ch 30, Tbl R-407C]

Pressure,	Temp.,* °F		Density, lb/ft^3	Volume, ft^3/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F		Specific Heat c_p, Btu/lb·°F		c_p/c_v
psia	Bubble	Dew	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor
1	–125.19	–111.30	94.24	43.0887	–26.34	93.96	–0.07002	0.28254	0.3065	0.1568	1.183
1.5	–115.58	–101.85	93.28	29.4430	–23.40	95.34	–0.06135	0.27716	0.3063	0.1600	1.182
2	–108.36	–94.75	92.55	22.4776	–21.18	96.37	–0.05499	0.27346	0.3063	0.1624	1.181
2.5	–102.52	–88.99	91.97	18.2333	–19.39	97.21	–0.04994	0.27066	0.3065	0.1644	1.181
3	–97.57	–84.12	91.47	15.3685	–17.87	97.92	–0.04572	0.26841	0.3068	0.1662	1.181
4	–89.43	–76.11	90.64	11.7361	–15.37	99.09	–0.03889	0.26495	0.3074	0.1693	1.181
5	–82.81	–69.61	89.97	9.5211	–13.34	100.03	–0.03345	0.26234	0.3081	0.1719	1.182
6	–77.20	–64.09	89.40	8.0252	–11.60	100.83	–0.02889	0.26025	0.3087	0.1742	1.182
7	–72.30	–59.27	88.89	6.9450	–10.09	101.52	–0.02496	0.25852	0.3094	0.1762	1.183
8	–67.94	–54.97	88.44	6.1272	–8.74	102.13	–0.02149	0.25705	0.3100	0.1781	1.184
10	–60.38	–47.55	87.66	4.9690	–6.39	103.19	–0.01556	0.25464	0.3112	0.1814	1.186
12	–53.96	–41.23	86.98	4.1864	–4.38	104.08	–0.01059	0.25272	0.3123	0.1844	1.188
14	–48.34	–35.71	86.39	3.6210	–2.62	104.85	–0.00629	0.25114	0.3133	0.1871	1.189
14.7[b]	–46.53	–33.93	86.19	3.4593	–2.06	105.10	–0.00492	0.25065	0.3137	0.1880	1.190
16	–43.32	–30.78	85.85	3.1928	–1.05	105.54	–0.00249	0.24979	0.3143	0.1896	1.191
18	–38.77	–26.31	85.36	2.8570	0.39	106.15	0.00092	0.24863	0.3153	0.1919	1.193
20	–34.61	–22.23	84.91	2.5862	1.70	106.71	0.00402	0.24760	0.3162	0.1941	1.195
22	–30.76	–18.45	84.50	2.3632	2.92	107.22	0.00687	0.24668	0.3172	0.1961	1.197
24	–27.18	–14.93	84.10	2.1761	4.06	107.70	0.00950	0.24586	0.3180	0.1981	1.199
26	–23.83	–11.64	83.73	2.0169	5.13	108.14	0.01196	0.24510	0.3189	0.1999	1.201
28	–20.66	–8.54	83.38	1.8798	6.15	108.55	0.01426	0.24442	0.3197	0.2017	1.203
30	–17.67	–5.60	83.05	1.7603	7.10	108.93	0.01643	0.24378	0.3205	0.2034	1.205
32	–14.84	–2.82	82.73	1.6553	8.02	109.30	0.01848	0.24319	0.3213	0.2051	1.207
34	–12.13	–0.17	82.43	1.5622	8.89	109.64	0.02042	0.24265	0.3221	0.2067	1.209
36	–9.55	2.37	82.14	1.4791	9.72	109.97	0.02227	0.24213	0.3229	0.2083	1.211
38	–7.07	4.79	81.85	1.4045	10.53	110.28	0.02404	0.24165	0.3236	0.2098	1.213
40	–4.70	7.12	81.58	1.3371	11.30	110.58	0.02573	0.24120	0.3244	0.2113	1.215
42	–2.41	9.37	81.32	1.2759	12.04	110.86	0.02735	0.24077	0.3251	0.2127	1.217
44	–0.20	11.53	81.06	1.2201	12.76	111.13	0.02891	0.24036	0.3258	0.2141	1.219
46	1.93	13.61	80.82	1.1690	13.46	111.39	0.03041	0.23998	0.3265	0.2155	1.221
48	3.98	15.63	80.58	1.1220	14.13	111.64	0.03186	0.23961	0.3272	0.2169	1.223
50	5.98	17.58	80.34	1.0786	14.79	111.88	0.03326	0.23926	0.3279	0.2182	1.225
55	10.71	22.21	79.78	0.9835	16.34	112.44	0.03656	0.23844	0.3296	0.2214	1.230
60	15.13	26.53	79.25	0.9037	17.81	112.96	0.03963	0.23771	0.3313	0.2246	1.235
65	19.27	30.58	78.75	0.8359	19.19	113.44	0.04250	0.23703	0.3329	0.2276	1.240
70	23.18	34.40	78.27	0.7774	20.49	113.88	0.04519	0.23641	0.3346	0.2305	1.245
75	26.88	38.02	77.82	0.7264	21.74	114.29	0.04773	0.23584	0.3362	0.2333	1.250
80	30.39	41.46	77.38	0.6816	22.92	114.67	0.05014	0.23530	0.3378	0.2361	1.255
85	33.75	44.73	76.95	0.6419	24.06	115.03	0.05243	0.23480	0.3393	0.2389	1.260
90	36.96	47.87	76.54	0.6064	25.16	115.37	0.05462	0.23432	0.3409	0.2416	1.266
95	40.04	50.87	76.15	0.5746	26.21	115.68	0.05671	0.23387	0.3424	0.2442	1.271
100	43.00	53.75	75.76	0.5458	27.23	115.98	0.05871	0.23344	0.3440	0.2468	1.276
110	48.60	59.21	75.02	0.4959	29.16	116.53	0.06250	0.23265	0.3471	0.2520	1.287
120	53.83	64.30	74.32	0.4540	30.99	117.03	0.06602	0.23191	0.3502	0.2570	1.298
130	58.75	69.08	73.64	0.4183	32.72	117.47	0.06932	0.23122	0.3533	0.2621	1.310
140	63.39	73.59	72.99	0.3875	34.36	117.88	0.07244	0.23058	0.3564	0.2671	1.321
150	67.79	77.86	72.37	0.3607	35.94	118.24	0.07538	0.22997	0.3596	0.2721	1.334
160	71.98	81.92	71.76	0.3372	37.45	118.57	0.07818	0.22938	0.3628	0.2772	1.346
170	75.97	85.79	71.17	0.3163	38.90	118.87	0.08086	0.22882	0.3660	0.2824	1.359
180	79.80	89.49	70.59	0.2976	40.30	119.15	0.08341	0.22828	0.3693	0.2876	1.373
190	83.47	93.04	70.02	0.2808	41.66	119.39	0.08587	0.22776	0.3727	0.2929	1.387
200	87.00	96.45	69.47	0.2656	42.97	119.61	0.08823	0.22725	0.3761	0.2983	1.401
220	93.69	102.90	68.40	0.2393	45.49	119.99	0.09271	0.22625	0.3832	0.3095	1.432
240	99.94	108.92	67.35	0.2171	47.88	120.29	0.09691	0.22529	0.3907	0.3213	1.466
260	105.82	114.56	66.33	0.1982	50.17	120.52	0.10088	0.22434	0.3986	0.3338	1.502
280	111.37	119.88	65.33	0.1819	52.36	120.68	0.10464	0.22340	0.4070	0.3473	1.542
300	116.64	124.91	64.34	0.1676	54.48	120.78	0.10824	0.22246	0.4161	0.3618	1.586
320	121.66	129.69	63.37	0.1550	56.53	120.82	0.11168	0.22152	0.4260	0.3777	1.635
340	126.45	134.24	62.39	0.1438	58.53	120.80	0.11500	0.22056	0.4368	0.3951	1.689
360	131.03	138.58	61.42	0.1337	60.47	120.73	0.11821	0.21958	0.4487	0.4143	1.750
380	135.43	142.73	60.44	0.1246	62.38	120.61	0.12132	0.21857	0.4620	0.4358	1.819
400	139.66	146.71	59.46	0.1163	64.25	120.42	0.12435	0.21753	0.4769	0.4600	1.897
450	149.59	155.98	56.92	0.0984	68.84	119.71	0.13167	0.21473	0.5248	0.5373	2.151
500	158.73	164.41	54.21	0.0835	73.37	118.56	0.13879	0.21152	0.5982	0.6546	2.541
550	167.22	172.09	51.15	0.0706	78.00	116.83	0.14595	0.20765	0.7284	0.8572	3.217
600	175.17	179.07	47.39	0.0586	83.04	114.18	0.15363	0.20253	1.0271	1.2973	4.683
650	182.79	185.22	41.60	0.0457	89.56	109.19	0.16351	0.19401	2.4146	3.0022	10.265
673.36[c]	186.94	186.94	31.59	0.0317	99.99	99.99	0.17797	0.17797	—	—	—

*Temperatures on ITS-90 scale [b]Bubble and dew points at one standard atmosphere [c]Critical point

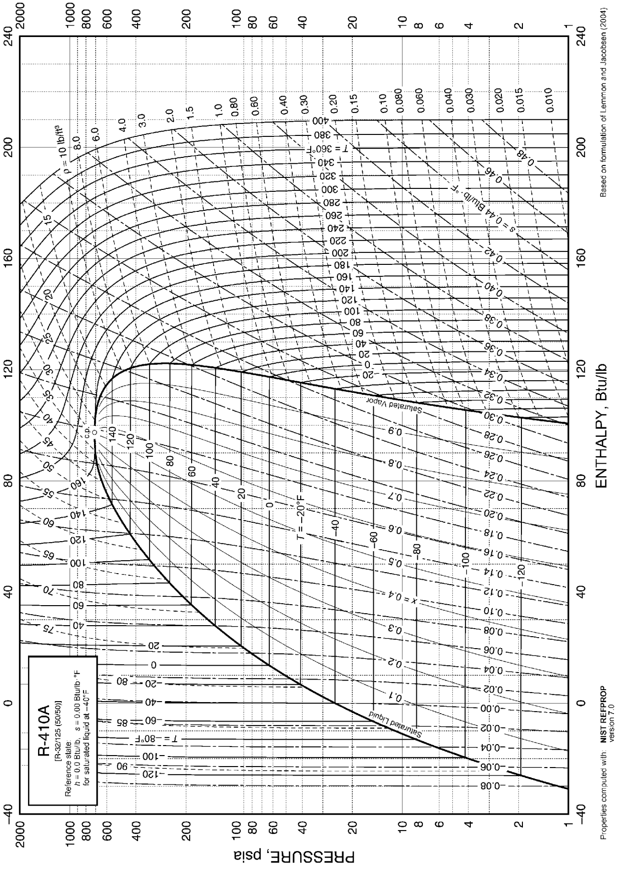


Figure 8.7 Pressure-Enthalpy Diagram for Refrigerant 410A [2017F, Ch 30, Fig 17]

Table 8.10 R-410A [R-32/125 (50/50)] Properties of Liquid on Bubble Line and Vapor on Dew Line [2017F, Ch 30, Tbl R-410A]

Pressure, psia	Temp.,* °F		Density, lb/ft³	Volume, ft³/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F		Specific Heat c_p, Btu/lb·°F		c_p/c_v
	Bubble	Dew	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor
1	–135.16	–134.98	92.02	47.6458	–30.90	100.62	–0.08330	0.32188	0.3215	0.1568	1.228
1.5	–126.03	–125.87	91.10	32.5774	–27.97	101.90	–0.07439	0.31477	0.3212	0.1600	1.227
2	–119.18	–119.02	90.41	24.8810	–25.76	102.86	–0.06786	0.30981	0.3213	0.1626	1.227
2.5	–113.63	–113.48	89.84	20.1891	–23.98	103.63	–0.06267	0.30602	0.3214	0.1648	1.228
3	–108.94	–108.78	89.36	17.0211	–22.47	104.27	–0.05834	0.30296	0.3216	0.1668	1.228
4	–101.22	–101.07	88.57	13.0027	–19.98	105.33	–0.05133	0.29820	0.3221	0.1703	1.229
5	–94.94	–94.80	87.92	10.5514	–17.96	106.18	–0.04574	0.29455	0.3226	0.1733	1.230
6	–89.63	–89.48	87.36	8.8953	–16.24	106.89	–0.04107	0.29162	0.3231	0.1760	1.232
7	–84.98	–84.84	86.87	7.6992	–14.74	107.50	–0.03704	0.28916	0.3236	0.1785	1.233
8	–80.85	–80.71	86.44	6.7935	–13.40	108.05	–0.03349	0.28705	0.3241	0.1807	1.234
10	–73.70	–73.56	85.67	5.5105	–11.08	108.97	–0.02743	0.28356	0.3251	0.1848	1.237
12	–67.62	–67.48	85.02	4.6434	–9.10	109.75	–0.02235	0.28075	0.3261	0.1884	1.240
14	–62.31	–62.16	84.44	4.0168	–7.36	110.42	–0.01795	0.27840	0.3270	0.1917	1.243
14.70[b]	–60.60	–60.46	84.26	3.8375	–6.80	110.63	–0.01655	0.27766	0.3274	0.1928	1.244
16	–57.56	–57.42	83.93	3.5423	–5.80	111.01	–0.01407	0.27638	0.3279	0.1947	1.245
18	–53.27	–53.13	83.45	3.1699	–4.39	111.54	–0.01059	0.27461	0.3288	0.1975	1.248
20	–49.34	–49.19	83.02	2.8698	–3.09	112.01	–0.00743	0.27305	0.3297	0.2002	1.251
22	–45.70	–45.56	82.61	2.6225	–1.89	112.45	–0.00452	0.27164	0.3305	0.2027	1.254
24	–42.32	–42.18	82.23	2.4151	–0.77	112.85	–0.00184	0.27036	0.3313	0.2050	1.256
26	–39.15	–39.01	81.87	2.2386	0.28	113.22	0.00067	0.26919	0.3321	0.2073	1.259
28	–36.17	–36.02	81.54	2.0865	1.27	113.56	0.00301	0.26811	0.3329	0.2094	1.261
30	–33.35	–33.20	81.21	1.9540	2.22	113.88	0.00522	0.26711	0.3337	0.2115	1.264
32	–30.68	–30.53	80.90	1.8375	3.11	114.19	0.00730	0.26617	0.3345	0.2135	1.267
34	–28.13	–27.98	80.61	1.7343	3.97	114.47	0.00928	0.26530	0.3352	0.2154	1.269
36	–25.69	–25.54	80.33	1.6422	4.79	114.74	0.01116	0.26448	0.3360	0.2173	1.272
38	–23.36	–23.20	80.05	1.5594	5.57	115.00	0.01296	0.26371	0.3367	0.2191	1.274
40	–21.12	–20.96	79.79	1.4847	6.33	115.24	0.01467	0.26297	0.3374	0.2208	1.277
42	–18.96	–18.81	79.54	1.4168	7.06	115.47	0.01632	0.26228	0.3382	0.2226	1.279
44	–16.89	–16.73	79.29	1.3549	7.76	115.69	0.01791	0.26162	0.3389	0.2242	1.282
46	–14.88	–14.73	79.05	1.2982	8.45	115.90	0.01943	0.26098	0.3396	0.2259	1.284
48	–12.94	–12.79	78.82	1.2460	9.11	116.10	0.02090	0.26038	0.3403	0.2275	1.287
50	–11.07	–10.91	78.59	1.1979	9.75	116.30	0.02232	0.25980	0.3410	0.2290	1.289
55	–6.62	–6.45	78.05	1.0925	11.27	116.75	0.02568	0.25845	0.3427	0.2328	1.295
60	–2.46	–2.30	77.54	1.0040	12.70	117.16	0.02880	0.25722	0.3445	0.2365	1.301
65	1.43	1.60	77.06	0.9287	14.05	117.53	0.03171	0.25610	0.3462	0.2400	1.308
70	5.10	5.27	76.60	0.8638	15.33	117.88	0.03444	0.25505	0.3478	0.2434	1.314
75	8.58	8.75	76.15	0.8073	16.54	118.20	0.03702	0.25408	0.3495	0.2467	1.320
80	11.88	12.06	75.73	0.7576	17.70	118.49	0.03946	0.25316	0.3512	0.2499	1.326
85	15.03	15.21	75.32	0.7135	18.81	118.77	0.04178	0.25231	0.3528	0.2531	1.333
90	18.05	18.22	74.93	0.6742	19.88	119.02	0.04400	0.25149	0.3545	0.2562	1.339
95	20.93	21.11	74.54	0.6389	20.91	119.26	0.04611	0.25072	0.3561	0.2592	1.345
100	23.71	23.89	74.17	0.6070	21.90	119.48	0.04815	0.24999	0.3578	0.2622	1.352
110	28.96	29.14	73.46	0.5515	23.79	119.89	0.05198	0.24862	0.3611	0.2681	1.365
120	33.86	34.05	72.78	0.5051	25.57	120.24	0.05555	0.24736	0.3644	0.2738	1.378
130	38.46	38.65	72.13	0.4655	27.25	120.56	0.05890	0.24618	0.3678	0.2795	1.392
140	42.80	42.99	71.51	0.4314	28.85	120.83	0.06205	0.24508	0.3712	0.2852	1.406
150	46.91	47.11	70.90	0.4016	30.38	121.08	0.06503	0.24403	0.3746	0.2908	1.420
160	50.82	51.02	70.32	0.3755	31.85	121.29	0.06787	0.24304	0.3781	0.2965	1.435
170	54.56	54.76	69.75	0.3523	33.27	121.48	0.07057	0.24210	0.3816	0.3022	1.451
180	58.13	58.33	69.20	0.3316	34.63	121.65	0.07316	0.24119	0.3851	0.3080	1.467
190	61.55	61.76	68.66	0.3130	35.95	121.79	0.07565	0.24031	0.3888	0.3139	1.483
200	64.84	65.05	68.13	0.2962	37.22	121.91	0.07804	0.23946	0.3925	0.3200	1.500
220	71.07	71.28	67.10	0.2669	39.67	122.09	0.08258	0.23783	0.4001	0.3325	1.537
240	76.89	77.10	66.11	0.2424	41.99	122.20	0.08683	0.23628	0.4081	0.3457	1.576
260	82.35	82.57	65.14	0.2215	44.21	122.25	0.09084	0.23478	0.4165	0.3599	1.619
280	87.51	87.73	64.19	0.2034	46.34	122.24	0.09464	0.23333	0.4255	0.3751	1.665
300	92.40	92.61	63.26	0.1876	48.40	122.18	0.09827	0.23190	0.4350	0.3915	1.716
320	97.04	97.26	62.34	0.1736	50.38	122.07	0.10175	0.23049	0.4452	0.4094	1.772
340	101.48	101.69	61.42	0.1613	52.31	121.91	0.10509	0.22909	0.4564	0.4290	1.833
360	105.71	105.93	60.52	0.1501	54.19	121.70	0.10832	0.22769	0.4685	0.4507	1.901
380	109.78	109.99	59.61	0.1401	56.03	121.44	0.11145	0.22629	0.4820	0.4747	1.977
400	113.68	113.89	58.70	0.1310	57.83	121.13	0.11450	0.22488	0.4971	0.5016	2.063
450	122.82	123.01	56.39	0.1114	62.23	120.14	0.12182	0.22124	0.5443	0.5857	2.333
500	131.19	131.38	53.97	0.0952	66.54	118.80	0.12888	0.21732	0.6143	0.7083	2.728
550	138.93	139.09	51.32	0.0814	70.89	117.02	0.13590	0.21295	0.7303	0.9059	3.367
600	146.12	146.25	48.24	0.0690	75.47	114.59	0.14320	0.20777	0.9603	1.2829	4.579
692.78[c]	158.40	158.40	34.18	0.0293	90.97	90.97	0.16781	0.16781	—	—	—

*Temperatures on ITS-90 scale [b]Bubble and dew points at one standard atmosphere [c]Critical point

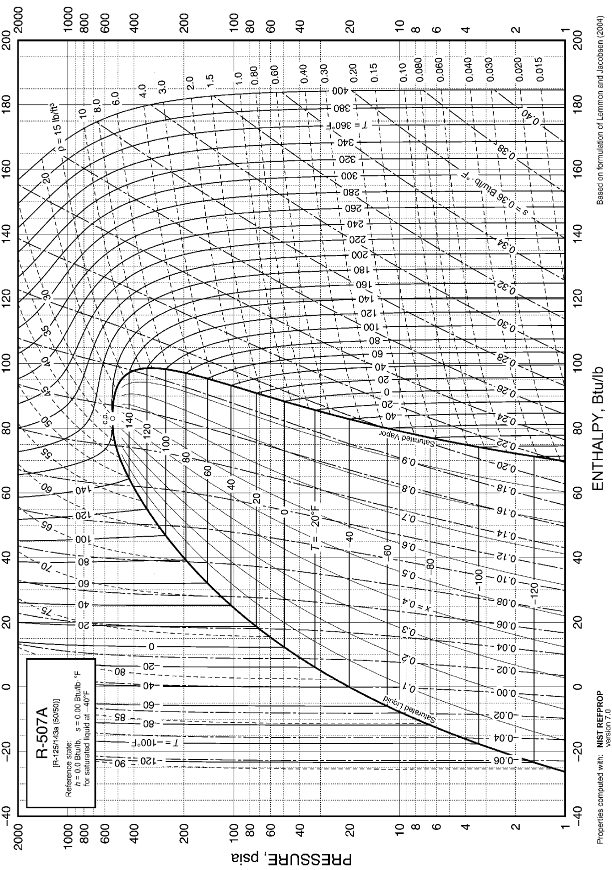


Figure 8.8 Pressure-Enthalpy Diagram for Refrigerant 507A [2017F, Ch 30, Fig 18]

Refrigerants

Table 8.11 R-507A [R-125/143a (50/50)] Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-507A]

Temp.,* °F	Pressure,** psia	Density, lb/ft³ Liquid	Volume, ft³/lb Vapor	Enthalpy, Btu/lb Liquid	Enthalpy, Btu/lb Vapor	Entropy, Btu/lb·°F Liquid	Entropy, Btu/lb·°F Vapor	Specific Heat c_p, Btu/lb·°F Liquid	Specific Heat c_p, Btu/lb·°F Vapor	c_p/c_v Vapor
–150	0.386	92.41	86.952	–32.027	67.009	–0.08831	0.23154	0.2919	0.1470	1.1650
–145	0.497	91.88	68.522	–30.571	67.711	–0.08365	0.22872	0.2904	0.1487	1.1637
–140	0.634	91.36	54.501	–29.121	68.416	–0.07908	0.22607	0.2893	0.1504	1.1626
–135	0.801	90.84	43.729	–27.677	69.126	–0.07460	0.22358	0.2885	0.1522	1.1616
–130	1.004	90.32	35.377	–26.235	69.838	–0.07019	0.22125	0.2879	0.1540	1.1607
–125	1.249	89.80	28.844	–24.796	70.554	–0.06586	0.21906	0.2876	0.1558	1.1599
–120	1.541	89.29	23.692	–23.359	71.272	–0.06160	0.21701	0.2874	0.1576	1.1593
–115	1.887	88.77	19.596	–21.921	71.993	–0.05740	0.21509	0.2874	0.1595	1.1588
–110	2.295	88.26	16.315	–20.484	72.716	–0.05326	0.21328	0.2875	0.1614	1.1584
–105	2.773	87.75	13.669	–19.045	73.440	–0.04918	0.21159	0.2878	0.1633	1.1581
–100	3.329	87.23	11.521	–17.604	74.166	–0.04515	0.21001	0.2882	0.1652	1.1580
–95	3.974	86.72	9.7644	–16.161	74.892	–0.04117	0.20852	0.2887	0.1672	1.1581
–90	4.715	86.20	8.3201	–14.716	75.619	–0.03723	0.20713	0.2893	0.1692	1.1583
–85	5.566	85.68	7.1254	–13.266	76.346	–0.03335	0.20583	0.2900	0.1712	1.1586
–80	6.535	85.16	6.1316	–11.813	77.073	–0.02950	0.20462	0.2908	0.1733	1.1592
–75	7.636	84.64	5.3004	–10.356	77.800	–0.02569	0.20348	0.2917	0.1754	1.1599
–70	8.879	84.11	4.6018	–8.894	78.525	–0.02192	0.20242	0.2926	0.1776	1.1607
–65	10.280	83.58	4.0116	–7.427	79.248	–0.01819	0.20143	0.2937	0.1798	1.1618
–60	11.849	83.05	3.5108	–5.954	79.970	–0.01449	0.20050	0.2948	0.1821	1.1631
–55	13.603	82.51	3.0839	–4.475	80.690	–0.01082	0.19963	0.2960	0.1844	1.1646
–52.13[b]	14.696	82.20	2.8676	–3.625	81.101	–0.00873	0.19916	0.2967	0.1858	1.1655
–50	15.554	81.97	2.7184	–2.990	81.406	–0.00719	0.19882	0.2972	0.1868	1.1663
–45	17.719	81.43	2.4043	–1.499	82.119	–0.00358	0.19807	0.2985	0.1893	1.1682
–40	20.112	80.88	2.1331	0.000	82.829	0.00000	0.19737	0.30000	0.1918	1.1704
–35	22.750	80.33	1.8983	1.506	83.534	0.00355	0.19671	0.3014	0.1944	1.1728
–30	25.649	79.77	1.6941	3.020	84.235	0.00708	0.19610	0.3030	0.1971	1.1755
–25	28.827	79.20	1.5160	4.541	84.931	0.01058	0.19553	0.3046	0.1998	1.1785
–20	32.300	78.63	1.3601	6.071	85.621	0.01407	0.19500	0.3063	0.2026	1.1818
–15	36.086	78.05	1.2231	7.610	86.304	0.01753	0.19450	0.3081	0.2056	1.1854
–10	40.203	77.46	1.1025	9.158	86.981	0.02097	0.19404	0.3100	0.2086	1.1894
–5	44.671	76.87	0.9960	10.716	87.651	0.02439	0.19360	0.3119	0.2117	1.1938
0	49.508	76.27	0.9016	12.284	88.313	0.02779	0.19319	0.3140	0.2149	1.1986
5	54.733	75.66	0.8177	13.862	88.966	0.03118	0.19281	0.3161	0.2183	1.2038
10	60.367	75.04	0.7430	15.452	89.610	0.03455	0.19245	0.3184	0.2218	1.2095
15	66.429	74.41	0.6763	17.052	90.245	0.03791	0.19211	0.3208	0.2254	1.2157
20	72.941	73.77	0.6165	18.665	90.868	0.04126	0.19179	0.3233	0.2291	1.2226
25	79.923	73.12	0.5629	20.290	91.480	0.04459	0.19148	0.3260	0.2330	1.2301
30	87.396	72.45	0.5146	21.929	92.079	0.04791	0.19118	0.3288	0.2371	1.2384
35	95.384	71.78	0.4711	23.581	92.664	0.05123	0.19089	0.3318	0.2414	1.2476
40	103.91	71.09	0.4318	25.249	93.234	0.05454	0.19061	0.3350	0.2460	1.2577
45	112.99	70.38	0.3962	26.931	93.788	0.05784	0.19032	0.3384	0.2508	1.2690
50	122.65	69.66	0.3638	28.630	94.324	0.06114	0.19004	0.3421	0.2560	1.2816
55	132.92	68.92	0.3344	30.346	94.840	0.06444	0.18976	0.3460	0.2616	1.2956
60	143.82	68.16	0.3076	32.080	95.336	0.06773	0.18946	0.3503	0.2676	1.3113
65	155.38	67.39	0.2832	33.834	95.808	0.07103	0.18916	0.3549	0.2742	1.3289
70	167.62	66.58	0.2608	35.609	96.255	0.07434	0.18884	0.3599	0.2814	1.3488
75	180.56	65.76	0.2403	37.406	96.675	0.07764	0.18850	0.3654	0.2894	1.3713
80	194.24	64.90	0.2214	39.228	97.065	0.08096	0.18814	0.3715	0.2983	1.3970
85	208.68	64.02	0.2041	41.076	97.421	0.08429	0.18775	0.3783	0.3083	1.4265
90	223.92	63.10	0.1880	42.952	97.740	0.08764	0.18732	0.3858	0.3196	1.4606
95	239.97	62.14	0.1732	44.860	98.019	0.09101	0.18686	0.3944	0.3325	1.5003
100	256.88	61.14	0.1595	46.803	98.251	0.09441	0.18634	0.4043	0.3475	1.5471
105	274.68	60.09	0.1468	48.784	98.431	0.09784	0.18576	0.4157	0.3650	1.6029
110	293.40	58.99	0.1349	50.809	98.551	0.10130	0.18511	0.4291	0.3858	1.6706
115	313.08	57.82	0.1238	52.885	98.600	0.10482	0.18438	0.4453	0.4112	1.7541
120	333.77	56.57	0.1134	55.018	98.568	0.10840	0.18354	0.4652	0.4427	1.8597
125	355.50	55.22	0.1036	57.221	98.435	0.11206	0.18256	0.4904	0.4833	1.9972
130	378.33	53.76	0.0943	59.509	98.177	0.11583	0.18141	0.5237	0.5375	2.1831
135	402.31	52.15	0.0855	61.903	97.759	0.11973	0.18003	0.5700	0.6142	2.4480
140	427.52	50.32	0.0769	64.439	97.125	0.12382	0.17833	0.6399	0.7313	2.8546
145	454.04	48.19	0.0684	67.182	96.173	0.12821	0.17616	0.7590	0.9326	3.5556
150	481.99	45.55	0.0597	70.265	94.697	0.13311	0.17318	1.0130	1.3606	5.0420
155	511.55	41.76	0.0499	74.107	92.081	0.13918	0.16842	1.9550	2.8693	10.2379
159.12[c]	537.40	30.64	0.0326	83.010	83.010	0.15339	0.15339	∞	∞	∞

*Temperatures on ITS-90 scale
**Small deviations from azeotropic behavior occur at some conditions; tabulated pressures are average of bubble and dew-point pressures

[b]Normal boiling point
[c] Critical point

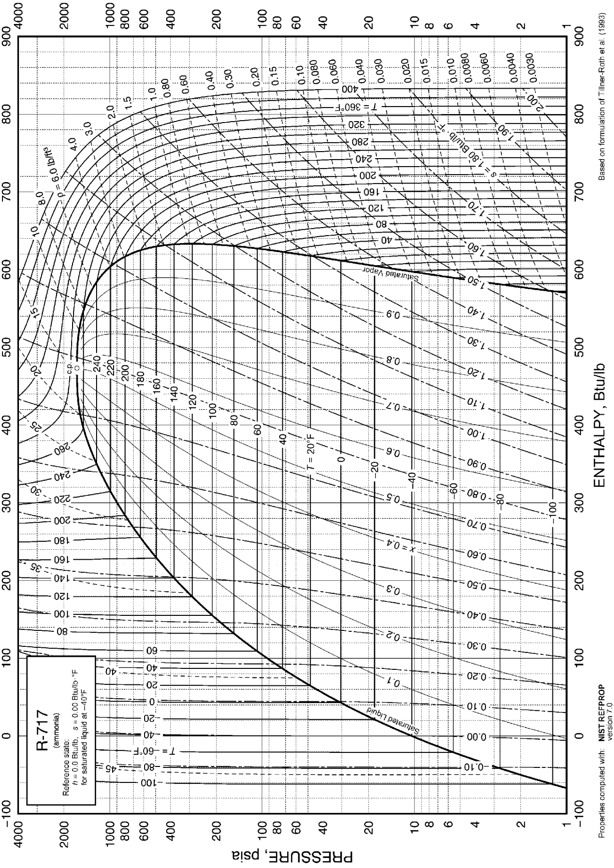


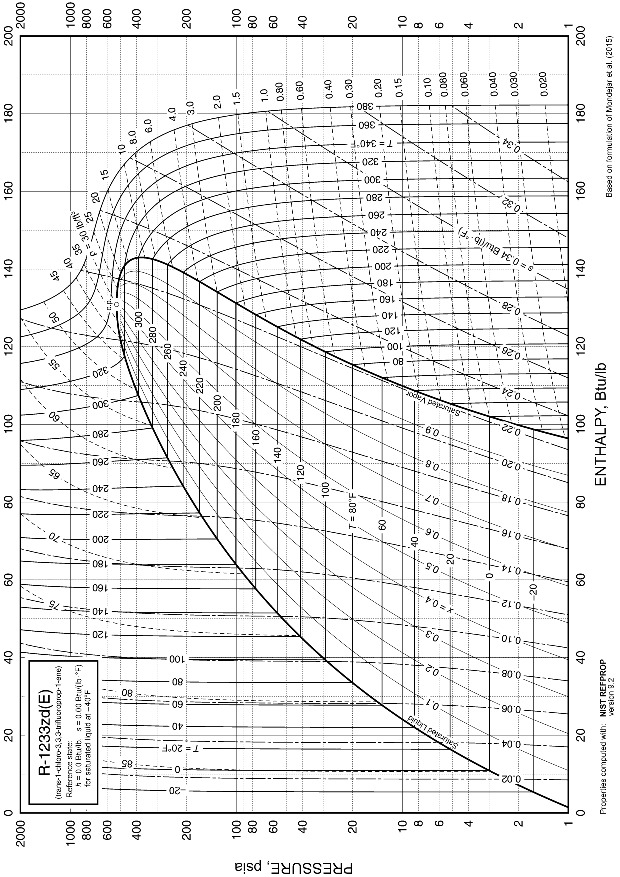
Figure 8.9 Pressure-Enthalpy Diagram for Refrigerant 717 (Ammonia) [2017F, Ch 30, Fig 19]

Table 8.12 R-717 (Ammonia) Properties of Saturated Liquid and Saturated Vapor

[2017F, Ch 30, Tbl R-717]

Temp.,* °F	Pressure, psia	Density, lb/ft³ Liquid	Volume, ft³/lb Vapor	Enthalpy, Btu/lb Liquid	Enthalpy, Btu/lb Vapor	Entropy, Btu/lb·°F Liquid	Entropy, Btu/lb·°F Vapor	Specific Heat c_p, Btu/lb·°F Liquid	Specific Heat c_p, Btu/lb·°F Vapor	c_p/c_v Vapor
–107.78[a]	0.883	45.75	249.92	–69.830	568.765	–0.18124	1.63351	1.0044	0.4930	1.3252
–100	1.237	45.47	182.19	–61.994	572.260	–0.15922	1.60421	1.0100	0.4959	1.3262
–90	1.864	45.09	124.12	–51.854	576.688	–0.13142	1.56886	1.0176	0.5003	1.3278
–80	2.739	44.71	86.546	–41.637	581.035	–0.10416	1.53587	1.0254	0.5056	1.3296
–70	3.937	44.31	61.647	–31.341	585.288	–0.07741	1.50503	1.0331	0.5118	1.3319
–60	5.544	43.91	44.774	–20.969	589.439	–0.05114	1.47614	1.0406	0.5190	1.3346
–50	7.659	43.50	33.105	–10.521	593.476	–0.02534	1.44900	1.0478	0.5271	1.3379
–40	10.398	43.08	24.881	0.000	597.387	0.00000	1.42347	1.0549	0.5364	1.3419
–30	13.890	42.66	18.983	10.592	601.162	0.02491	1.39938	1.0617	0.5467	1.3465
–27.99[b]	14.696	42.57	18.007	12.732	601.904	0.02987	1.39470	1.0631	0.5490	1.3475
–25	15.962	42.45	16.668	15.914	602.995	0.03720	1.38784	1.0651	0.5524	1.3491
–20	18.279	42.23	14.684	21.253	604.789	0.04939	1.37660	1.0684	0.5583	1.3520
–15	20.858	42.01	12.976	26.609	606.544	0.06148	1.36567	1.0716	0.5646	1.3550
–10	23.723	41.79	11.502	31.982	608.257	0.07347	1.35502	1.0749	0.5711	1.3584
–5	26.895	41.57	10.226	37.372	609.928	0.08536	1.34463	1.0782	0.5781	1.3619
0	30.397	41.34	9.1159	42.779	611.554	0.09715	1.33450	1.0814	0.5853	1.3657
5	34.253	41.12	8.1483	48.203	613.135	0.10885	1.32462	1.0847	0.5929	1.3698
10	38.487	40.89	7.3020	53.644	614.669	0.12045	1.31496	1.0880	0.6009	1.3742
15	43.126	40.66	6.5597	59.103	616.154	0.13197	1.30552	1.0914	0.6092	1.3789
20	48.194	40.43	5.9067	64.579	617.590	0.14340	1.29629	1.0948	0.6179	1.3840
25	53.720	40.20	5.3307	70.072	618.974	0.15474	1.28726	1.0983	0.6271	1.3894
30	59.730	39.96	4.8213	75.585	620.305	0.16599	1.27842	1.1019	0.6366	1.3951
35	66.255	39.72	4.3695	81.116	621.582	0.17717	1.26975	1.1056	0.6465	1.4012
40	73.322	39.48	3.9680	86.666	622.803	0.18827	1.26125	1.1094	0.6569	1.4078
45	80.962	39.24	3.6102	92.237	623.967	0.19929	1.25291	1.1134	0.6678	1.4147
50	89.205	38.99	3.2906	97.828	625.072	0.21024	1.24472	1.1175	0.6791	1.4222
55	98.083	38.75	3.0045	103.441	626.115	0.22111	1.23667	1.1218	0.6909	1.4301
60	107.63	38.50	2.7479	109.076	627.097	0.23192	1.22875	1.126	0.703	1.438
65	117.87	38.25	2.5172	114.734	628.013	0.24266	1.22095	1.131	0.716	1.447
70	128.85	37.99	2.3094	120.417	628.864	0.25334	1.21327	1.136	0.730	1.457
75	140.59	37.73	2.1217	126.126	629.647	0.26396	1.20570	1.141	0.744	1.467
80	153.13	37.47	1.9521	131.861	630.359	0.27452	1.19823	1.147	0.758	1.478
85	166.51	37.21	1.7983	137.624	630.999	0.28503	1.19085	1.153	0.774	1.490
90	180.76	36.94	1.6588	143.417	631.564	0.29549	1.18356	1.159	0.790	1.502
95	195.91	36.67	1.5319	149.241	632.052	0.30590	1.17634	1.166	0.807	1.515
100	212.01	36.40	1.4163	155.098	632.460	0.31626	1.16920	1.173	0.824	1.529
105	229.09	36.12	1.3108	160.990	632.785	0.32659	1.16211	1.180	0.843	1.544
110	247.19	35.83	1.2144	166.919	633.025	0.33688	1.15508	1.188	0.862	1.561
115	266.34	35.55	1.1262	172.887	633.175	0.34713	1.14809	1.197	0.883	1.578
120	286.60	35.26	1.0452	178.896	633.232	0.35736	1.14115	1.206	0.905	1.597
125	307.98	34.96	0.9710	184.949	633.193	0.36757	1.13423	1.216	0.928	1.617
130	330.54	34.66	0.9026	191.049	633.053	0.37775	1.12733	1.227	0.952	1.638
135	354.32	34.35	0.8397	197.199	632.807	0.38792	1.12044	1.239	0.978	1.662
140	379.36	34.04	0.7817	203.403	632.451	0.39808	1.11356	1.251	1.006	1.687
145	405.70	33.72	0.7280	209.663	631.978	0.40824	1.10666	1.265	1.035	1.715
150	433.38	33.39	0.6785	215.984	631.383	0.41840	1.09975	1.280	1.067	1.745
155	462.45	33.06	0.6325	222.370	630.659	0.42857	1.09281	1.296	1.101	1.778
160	492.95	32.72	0.5899	228.827	629.798	0.43875	1.08582	1.313	1.138	1.813
165	524.94	32.37	0.5504	235.359	628.791	0.44896	1.07878	1.333	1.178	1.853
170	558.45	32.01	0.5136	241.973	627.630	0.45919	1.07167	1.354	1.222	1.896
175	593.53	31.64	0.4793	248.675	626.302	0.46947	1.06447	1.377	1.270	1.944
180	630.24	31.26	0.4473	255.472	624.797	0.47980	1.05717	1.403	1.322	1.998
185	668.63	30.87	0.4174	262.374	623.100	0.49019	1.04974	1.432	1.381	2.058
190	708.74	30.47	0.3895	269.390	621.195	0.50066	1.04217	1.465	1.446	2.126
195	750.64	30.05	0.3633	276.530	619.064	0.51121	1.03443	1.502	1.519	2.203
200	794.38	29.62	0.3387	283.809	616.686	0.52188	1.02649	1.543	1.602	2.290
205	840.03	29.17	0.3156	291.240	614.035	0.53267	1.01831	1.591	1.697	2.392
210	887.64	28.70	0.2938	298.842	611.081	0.54360	1.00986	1.646	1.806	2.509
215	937.28	28.21	0.2733	306.637	607.788	0.55472	1.00109	1.711	1.935	2.648
220	989.03	27.69	0.2538	314.651	604.112	0.56605	0.99193	1.788	2.088	2.814
225	1042.96	27.15	0.2354	322.918	599.996	0.57763	0.98232	1.882	2.272	3.015
230	1099.14	26.57	0.2178	331.483	595.371	0.58953	0.97216	1.999	2.501	3.265
235	1157.69	25.95	0.2010	340.404	590.142	0.60182	0.96133	2.148	2.790	3.582
240	1218.68	25.28	0.1849	349.766	584.183	0.61462	0.94966	2.346	3.171	4.000
245	1282.24	24.55	0.1693	359.695	577.309	0.62809	0.93690	2.624	3.693	4.575
250	1348.49	23.72	0.1540	370.391	569.240	0.64249	0.92269	3.047	4.460	5.420
260	1489.71	21.60	0.1233	395.943	547.139	0.67662	0.88671	5.273	8.106	9.439
270.05[c]	1643.71	14.05	0.0712	473.253	473.253	0.78093	0.78093	∞	∞	∞

*Temperatures on ITS-90 scale [a]Triple point [b]Normal boiling point [c]Critical point



Refrigerants

Figure 8.10 Pressure-Enthalpy Diagram for Refrigerant 1233zd(E) [2017F, Ch 30, Fig 12]

Table 8.13 Refrigerant 1233zd(E) (trans-1-chloro-3,3,3-trifluoroprop-1-ene) Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-1234zd(E)]

Temp., °F	Pressure, psia	Density, lb/ft^3	Volume, ft^3/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F		Specific Heat, c_p Btu/lb·°F		c_p / c_v
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor
–60	0.387	89.41	84.710	–5.327	92.383	–0.01300	0.23147	0.2646	0.1573	1.1086
–50	0.572	88.68	58.669	–2.673	93.947	–0.00644	0.22940	0.2663	0.1603	1.1069
–40	0.827	87.94	41.517	–0.000	95.532	–0.00000	0.22764	0.2681	0.1632	1.1054
–30	1.171	87.20	29.964	2.690	97.137	0.00633	0.22614	0.2699	0.1661	1.1043
–20	1.628	86.46	22.022	5.398	98.758	0.01256	0.22490	0.2717	0.1689	1.1034
–15	1.906	86.09	18.999	6.759	99.575	0.01564	0.22437	0.2726	0.1704	1.1030
–10	2.223	85.71	16.457	8.125	100.394	0.01869	0.22388	0.2735	0.1718	1.1028
–5	2.581	85.33	14.310	9.495	101.217	0.02172	0.22345	0.2744	0.1732	1.1026
0	2.986	84.96	12.488	10.870	102.043	0.02472	0.22307	0.2753	0.1746	1.1025
5	3.441	84.57	10.937	12.249	102.871	0.02771	0.22273	0.2762	0.1760	1.1024
10	3.952	84.19	9.6112	13.634	103.701	0.03067	0.22244	0.2772	0.1774	1.1024
15	4.522	83.81	8.4736	15.022	104.533	0.03361	0.22218	0.2781	0.1789	1.1026
20	5.157	83.42	7.4939	16.416	105.367	0.03652	0.22197	0.2791	0.1803	1.1027
25	5.862	83.03	6.6474	17.815	106.202	0.03942	0.22179	0.2800	0.1817	1.1030
30	6.643	82.64	5.9135	19.218	107.038	0.04230	0.22164	0.2810	0.1832	1.1034
35	7.505	82.25	5.2752	20.626	107.874	0.04516	0.22153	0.2819	0.1846	1.1038
40	8.455	81.85	4.7183	22.040	108.711	0.04799	0.22145	0.2829	0.1861	1.1043
45	9.498	81.45	4.2310	23.458	109.549	0.05081	0.22140	0.2839	0.1876	1.1050
50	10.641	81.05	3.8033	24.881	110.386	0.05362	0.22138	0.2849	0.1891	1.1057
55	11.890	80.65	3.4269	26.310	111.223	0.05640	0.22138	0.2859	0.1907	1.1065
60	13.252	80.24	3.0947	27.744	112.059	0.05917	0.22141	0.2870	0.1922	1.1074
64.87[b]	14.696	79.84	2.8079	29.146	112.873	0.06185	0.22146	0.2880	0.1937	1.1083
65	14.735	79.83	2.8008	29.183	112.894	0.06191	0.22147	0.2880	0.1938	1.1084
70	16.345	79.42	2.5402	30.627	113.728	0.06465	0.22154	0.2891	0.1954	1.1095
75	18.090	79.00	2.3084	32.077	114.561	0.06736	0.22164	0.2902	0.1970	1.1107
80	19.977	78.58	2.1018	33.532	115.392	0.07007	0.22175	0.2913	0.1987	1.1120
85	22.013	78.16	1.9172	34.993	116.222	0.07275	0.22188	0.2924	0.2004	1.1135
90	24.208	77.73	1.7519	36.460	117.049	0.07542	0.22204	0.2935	0.2021	1.1150
95	26.569	77.30	1.6037	37.933	117.874	0.07808	0.22220	0.2947	0.2038	1.1167
100	29.104	76.86	1.4703	39.412	118.696	0.08072	0.22239	0.2959	0.2056	1.1186
105	31.821	76.42	1.3502	40.896	119.515	0.08335	0.22258	0.2971	0.2075	1.1205
110	34.730	75.98	1.2418	42.387	120.331	0.08597	0.22279	0.2984	0.2093	1.1226
115	37.838	75.53	1.1437	43.885	121.144	0.08857	0.22301	0.2997	0.2113	1.1249
120	41.155	75.08	1.0548	45.389	121.953	0.09116	0.22325	0.3010	0.2132	1.1274
125	44.690	74.62	0.9741	46.900	122.758	0.09374	0.22349	0.3024	0.2152	1.1300
130	48.452	74.16	0.9007	48.417	123.558	0.09631	0.22374	0.3038	0.2173	1.1328
135	52.449	73.69	0.8339	49.942	124.354	0.09887	0.22400	0.3052	0.2194	1.1358
140	56.693	73.21	0.7729	51.474	125.145	0.10142	0.22427	0.3067	0.2216	1.1391
145	61.191	72.73	0.7171	53.014	125.931	0.10396	0.22455	0.3083	0.2239	1.1426
150	65.955	72.25	0.6661	54.561	126.711	0.10649	0.22483	0.3099	0.2262	1.1463
155	70.993	71.75	0.6193	56.117	127.485	0.10900	0.22511	0.3116	0.2286	1.1503
160	76.316	71.25	0.5763	57.680	128.253	0.11152	0.22540	0.3133	0.2311	1.1546
165	81.934	70.75	0.5368	59.253	129.013	0.11402	0.22570	0.3151	0.2337	1.1593
170	87.858	70.23	0.5004	60.834	129.767	0.11652	0.22599	0.3170	0.2364	1.1643
175	94.098	69.71	0.4669	62.424	130.512	0.11901	0.22629	0.3190	0.2392	1.1697
180	100.66	69.17	0.4359	64.024	131.249	0.12149	0.22658	0.3211	0.2421	1.1755
185	107.57	68.63	0.4072	65.634	131.977	0.12397	0.22688	0.3233	0.2451	1.1819
190	114.82	68.08	0.3807	67.254	132.695	0.12644	0.22717	0.3256	0.2483	1.1887
195	122.43	67.52	0.3561	68.886	133.403	0.12891	0.22746	0.3280	0.2517	1.1962
200	130.42	66.95	0.3333	70.528	134.099	0.13138	0.22774	0.3306	0.2552	1.2043
210	147.55	65.77	0.2923	73.850	135.455	0.13630	0.22829	0.3363	0.2629	1.2230
220	166.31	64.54	0.2567	77.225	136.754	0.14123	0.22881	0.3427	0.2717	1.2456
230	186.81	63.24	0.2256	80.660	137.988	0.14615	0.22928	0.3503	0.2818	1.2733
240	209.15	61.87	0.1983	84.161	139.142	0.15110	0.22968	0.3591	0.2938	1.3079
250	233.46	60.42	0.1742	87.741	140.202	0.15608	0.23000	0.3698	0.3084	1.3522
260	259.84	58.86	0.1528	91.410	141.147	0.16110	0.23021	0.3831	0.3267	1.4103
270	288.46	57.17	0.1337	95.189	141.946	0.16618	0.23026	0.4002	0.3506	1.4894
280	319.45	55.31	0.1165	99.102	142.560	0.17137	0.23012	0.4231	0.3835	1.6020
290	352.98	53.22	0.1009	103.188	142.927	0.17670	0.22971	0.4563	0.4319	1.7731
300	389.27	50.82	0.0864	107.515	142.944	0.18227	0.22890	0.5093	0.5111	2.0591
310	428.56	47.89	0.0727	112.212	142.420	0.18822	0.22746	0.6110	0.6642	2.6221
320	471.16	43.95	0.0590	117.623	140.880	0.19498	0.22481	0.896	1.086	4.191
330	517.63	36.02	0.0415	125.873	135.691	0.20521	0.21765	5.598	7.067	26.548
331.61[c]	525.57	29.98	0.0334	130.891	130.891	0.21150	0.21150	∞	∞	∞

[b]Normal boiling point [c]Critical point

Refrigerants

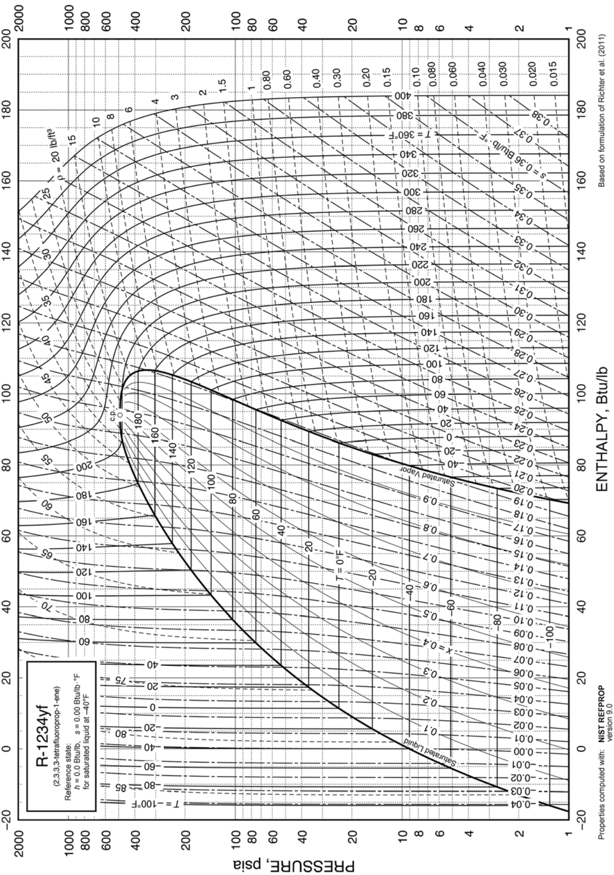


Figure 8.11 Pressure-Enthalpy Diagram for Refrigerant 1234yf [2017F, Ch 30, Fig 19]

Refrigerants

Table 8.14 Refrigerant 1234yf (2,3,3,3-tetrafluoroprop-1-ene) Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-1234yf]

Temp., °F	Pressure, psia	Density, lb/ft^3 Liquid	Volume, ft^3/lb Vapor	Enthalpy, Btu/lb Liquid	Enthalpy, Btu/lb Vapor	Entropy, Btu/lb·°F Liquid	Entropy, Btu/lb·°F Vapor	Specific Heat, c_p Btu/lb·°F Liquid	Specific Heat, c_p Btu/lb·°F Vapor	c_p / c_v Vapor
–60	5.111	82.49	7.1955	–5.458	76.593	–0.01330	0.19200	0.2688	0.1776	1.1241
–55	5.932	82.03	6.2622	–4.109	77.395	–0.00995	0.19146	0.2707	0.1796	1.1243
–50	6.855	81.58	5.4710	–2.749	78.198	–0.00662	0.19097	0.2727	0.1817	1.1247
–45	7.889	81.11	4.7974	–1.380	79.002	–0.00330	0.19055	0.2746	0.1838	1.1252
–40	9.046	80.65	4.2215	–0.000	79.808	–0.00000	0.19017	0.2766	0.1859	1.1258
–35	10.333	80.18	3.7271	1.390	80.614	0.00328	0.18984	0.2787	0.1880	1.1265
–30	11.761	79.71	3.3012	2.790	81.420	0.00655	0.18956	0.2807	0.1903	1.1274
–25	13.341	79.23	2.9329	4.200	82.226	0.00981	0.18932	0.2828	0.1925	1.1285
–21.07[b]	14.696	78.85	2.6781	5.315	82.859	0.01236	0.18916	0.2844	0.1943	1.1294
–20	15.084	78.75	2.6132	5.621	83.032	0.01305	0.18912	0.2848	0.1948	1.1297
–15	17.001	78.26	2.3349	7.053	83.837	0.01628	0.18896	0.2870	0.1971	1.1310
–10	19.104	77.77	2.0917	8.495	84.641	0.01949	0.18883	0.2891	0.1995	1.1325
–5	21.404	77.28	1.8786	9.948	85.444	0.02269	0.18874	0.2912	0.2019	1.1342
0	23.914	76.78	1.6913	11.412	86.244	0.02588	0.18868	0.2934	0.2043	1.1361
5	26.647	76.27	1.5262	12.887	87.043	0.02906	0.18865	0.2956	0.2068	1.1381
10	29.615	75.76	1.3802	14.374	87.839	0.03223	0.18865	0.2979	0.2094	1.1404
15	32.831	75.24	1.2508	15.871	88.632	0.03538	0.18867	0.3001	0.2120	1.1429
20	36.309	74.72	1.1357	17.381	89.422	0.03853	0.18872	0.3024	0.2147	1.1457
25	40.062	74.19	1.0332	18.902	90.208	0.04166	0.18878	0.3048	0.2174	1.1486
30	44.105	73.65	0.9416	20.434	90.989	0.04479	0.18887	0.3072	0.2202	1.1519
35	48.451	73.11	0.8596	21.979	91.765	0.04790	0.18898	0.3096	0.2231	1.1555
40	53.116	72.55	0.7860	23.536	92.536	0.05101	0.18910	0.3121	0.2261	1.1594
45	58.113	71.99	0.7198	25.106	93.301	0.05411	0.18924	0.3147	0.2291	1.1637
50	63.459	71.42	0.6601	26.688	94.059	0.05720	0.18939	0.3173	0.2323	1.1685
55	69.167	70.84	0.6062	28.283	94.810	0.06029	0.18955	0.3199	0.2355	1.1736
60	75.255	70.25	0.5573	29.891	95.552	0.06337	0.18972	0.3227	0.2389	1.1793
65	81.737	69.65	0.5130	31.513	96.285	0.06644	0.18989	0.3255	0.2425	1.1856
70	88.629	69.04	0.4728	33.149	97.008	0.06951	0.19007	0.3285	0.2462	1.1926
75	95.949	68.42	0.4361	34.799	97.720	0.07257	0.19025	0.3315	0.2501	1.2002
80	103.71	67.78	0.4027	36.463	98.420	0.07563	0.19044	0.3346	0.2543	1.2087
85	111.94	67.14	0.3721	38.142	99.106	0.07869	0.19062	0.3379	0.2587	1.2181
90	120.64	66.47	0.3441	39.837	99.779	0.08174	0.19079	0.3413	0.2635	1.2286
95	129.84	65.80	0.3185	41.548	100.435	0.08479	0.19096	0.3450	0.2686	1.2402
100	139.55	65.10	0.2949	43.275	101.075	0.08784	0.19112	0.3488	0.2742	1.2533
105	149.80	64.39	0.2732	45.021	101.696	0.09090	0.19126	0.3530	0.2802	1.2679
110	160.60	63.66	0.2532	46.784	102.296	0.09395	0.19140	0.3574	0.2867	1.2843
115	171.97	62.92	0.2347	48.568	102.874	0.09701	0.19151	0.3623	0.2940	1.3028
120	183.93	62.14	0.2176	50.373	103.428	0.10008	0.19160	0.3676	0.3019	1.3239
125	196.51	61.35	0.2017	52.201	103.955	0.10315	0.19167	0.3735	0.3107	1.3479
130	209.72	60.52	0.1870	54.054	104.452	0.10624	0.19171	0.3801	0.3206	1.3756
135	223.59	59.66	0.1733	55.935	104.916	0.10934	0.19171	0.3875	0.3318	1.4077
140	238.13	58.77	0.1606	57.845	105.342	0.11246	0.19167	0.3959	0.3446	1.4453
145	253.39	57.83	0.1487	59.789	105.726	0.11561	0.19158	0.4055	0.3594	1.4898
150	269.37	56.84	0.1375	61.769	106.061	0.11879	0.19144	0.4167	0.3767	1.5432
155	286.11	55.80	0.1270	63.792	106.340	0.12200	0.19122	0.4300	0.3974	1.6082
160	303.64	54.68	0.1172	65.861	106.554	0.12526	0.19093	0.4459	0.4227	1.6891
165	321.99	53.49	0.1078	67.986	106.690	0.12857	0.19053	0.4655	0.4544	1.7922
170	341.19	52.21	0.0990	70.175	106.731	0.13196	0.19001	0.4906	0.4956	1.9275
175	361.28	50.80	0.0905	72.445	106.653	0.13543	0.18933	0.5241	0.5513	2.1127
180	382.32	49.24	0.0823	74.816	106.421	0.13903	0.18844	0.5717	0.6314	2.3809
185	404.35	47.47	0.0743	77.328	105.976	0.14281	0.18725	0.6458	0.7571	2.8031
190	427.45	45.39	0.0662	80.050	105.213	0.14688	0.18561	0.7788	0.9837	3.5641
195	451.72	42.73	0.0578	83.145	103.888	0.15147	0.18315	1.094	1.517	5.344
200	477.33	38.53	0.0475	87.241	101.103	0.15752	0.17853	2.821	4.285	14.439
202.46[c]	490.55	29.69	0.0337	93.995	93.995	0.16763	0.16763	∞	∞	∞

[b]Normal boiling point [c]Critical point

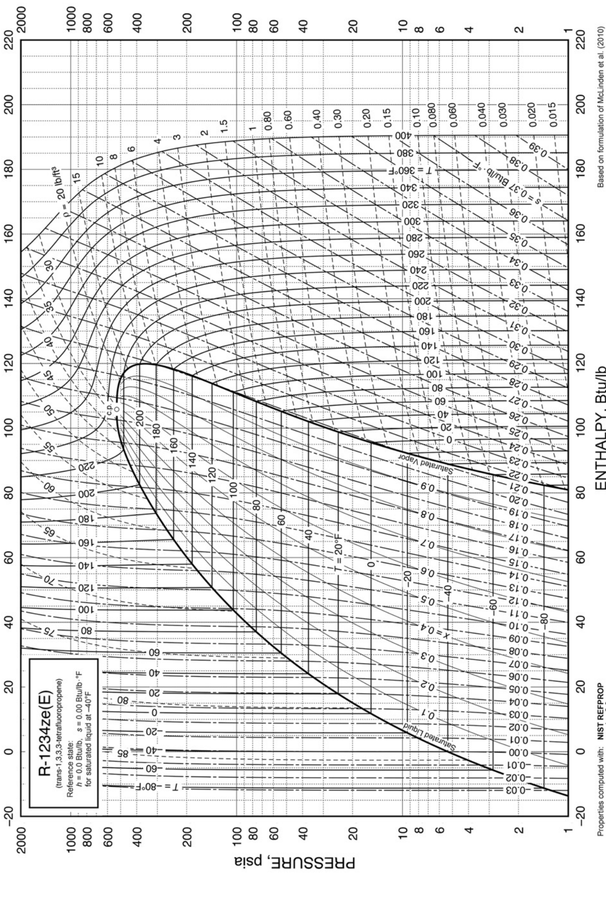


Figure 8.12 Pressure-Enthalpy Diagram for Refrigerant 1234ze(E) [2017F, Ch 30, Fig 13]

Table 8.15 Refrigerant 1234ze(E) (Trans-1,3,3,3-Tetrafluoropropene) Properties of Saturated Liquid and Saturated Vapor [2017F, Ch 30, Tbl R-1234ze(E)]

Temp., °F	Pressure, psia	Density, lb/ft^3	Volume, ft^3/lb	Enthalpy, Btu/lb		Entropy, Btu/lb·°F		Specific Heat, c_p Btu/lb·°F		c_p / c_v
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Vapor
–60	2.839	86.03	13.083	–5.955	85.763	–0.01452	0.21496	0.2962	0.1797	1.1146
–55	3.347	85.59	11.219	–4.472	86.613	–0.01084	0.21425	0.2968	0.1812	1.1145
–50	3.926	85.14	9.6638	–2.985	87.465	–0.00719	0.21360	0.2976	0.1827	1.1146
–45	4.584	84.70	8.3603	–1.494	88.319	–0.00358	0.21301	0.2983	0.1843	1.1148
–40	5.330	84.25	7.2623	–0.000	89.172	–0.00000	0.21248	0.2991	0.1858	1.1150
–35	6.170	83.80	6.3334	1.499	90.027	0.00355	0.21201	0.2999	0.1873	1.1154
–30	7.113	83.34	5.5439	3.002	90.881	0.00706	0.21159	0.3008	0.1889	1.1159
–25	8.170	82.88	4.8703	4.509	91.735	0.01054	0.21121	0.3017	0.1905	1.1165
–20	9.347	82.42	4.2931	6.022	92.588	0.01400	0.21088	0.3027	0.1921	1.1172
–15	10.657	81.96	3.7966	7.539	93.440	0.01742	0.21060	0.3037	0.1938	1.1181
–10	12.107	81.50	3.3680	9.062	94.290	0.02082	0.21035	0.3047	0.1955	1.1191
–5	13.710	81.03	2.9967	10.590	95.138	0.02419	0.21015	0.3058	0.1972	1.1202
–2.15[b]	14.696	80.76	2.8072	11.463	95.621	0.02610	0.21004	0.3064	0.1982	1.1209
0	15.476	80.55	2.6738	12.123	95.985	0.02754	0.20997	0.3069	0.1989	1.1214
5	17.416	80.07	2.3922	13.663	96.828	0.03086	0.20983	0.3081	0.2007	1.1228
10	19.542	79.59	2.1458	15.208	97.669	0.03415	0.20973	0.3093	0.2026	1.1244
15	21.865	79.11	1.9295	16.760	98.506	0.03743	0.20965	0.3105	0.2044	1.1261
20	24.398	78.62	1.7391	18.318	99.339	0.04068	0.20959	0.3118	0.2064	1.1280
25	27.153	78.12	1.5710	19.883	100.168	0.04391	0.20956	0.3131	0.2083	1.1301
30	30.143	77.62	1.4221	21.455	100.993	0.04713	0.20956	0.3145	0.2104	1.1324
35	33.382	77.11	1.2900	23.034	101.812	0.05032	0.20957	0.3160	0.2125	1.1349
40	36.882	76.60	1.1725	24.620	102.627	0.05349	0.20961	0.3175	0.2146	1.1376
45	40.658	76.09	1.0676	26.215	103.435	0.05665	0.20966	0.3191	0.2169	1.1405
50	44.723	75.56	0.9738	27.817	104.237	0.05979	0.20973	0.3208	0.2192	1.1437
55	49.093	75.03	0.8897	29.428	105.032	0.06291	0.20981	0.3225	0.2216	1.1472
60	53.780	74.49	0.8141	31.048	105.820	0.06602	0.20991	0.3243	0.2240	1.1510
65	58.802	73.95	0.7461	32.677	106.600	0.06912	0.21001	0.3262	0.2266	1.1551
70	64.172	73.40	0.6848	34.315	107.371	0.07220	0.21013	0.3282	0.2293	1.1596
75	69.905	72.84	0.6293	35.964	108.134	0.07527	0.21025	0.3303	0.2321	1.1645
80	76.019	72.27	0.5790	37.623	108.887	0.07833	0.21038	0.3325	0.2350	1.1698
85	82.528	71.69	0.5334	39.293	109.630	0.08138	0.21052	0.3348	0.2380	1.1757
90	89.450	71.10	0.4919	40.974	110.362	0.08442	0.21066	0.3373	0.2412	1.1820
95	96.800	70.50	0.4541	42.668	111.081	0.08745	0.21079	0.3399	0.2446	1.1889
100	104.60	69.89	0.4196	44.374	111.789	0.09048	0.21093	0.3426	0.2481	1.1966
105	112.85	69.26	0.3881	46.093	112.482	0.09350	0.21107	0.3455	0.2519	1.2049
110	121.59	68.63	0.3592	47.826	113.161	0.09651	0.21120	0.3486	0.2559	1.2142
115	130.83	67.97	0.3327	49.573	113.823	0.09952	0.21132	0.3519	0.2602	1.2245
120	140.58	67.31	0.3084	51.336	114.469	0.10253	0.21144	0.3554	0.2648	1.2359
125	150.87	66.62	0.2860	53.115	115.095	0.10554	0.21154	0.3592	0.2697	1.2486
130	161.72	65.92	0.2653	54.911	115.701	0.10854	0.21164	0.3632	0.2752	1.2629
135	173.14	65.20	0.2462	56.726	116.285	0.11155	0.21171	0.3676	0.2811	1.2789
140	185.15	64.46	0.2286	58.560	116.845	0.11457	0.21176	0.3724	0.2877	1.2971
145	197.78	63.69	0.2122	60.414	117.379	0.11759	0.21179	0.3777	0.2950	1.3176
150	211.04	62.90	0.1971	62.291	117.884	0.12061	0.21180	0.3835	0.3032	1.3412
155	224.97	62.08	0.1829	64.192	118.357	0.12365	0.21177	0.3899	0.3124	1.3682
160	239.58	61.22	0.1698	66.120	118.794	0.12670	0.21171	0.3972	0.3229	1.3994
165	254.89	60.33	0.1575	68.076	119.193	0.12977	0.21160	0.4054	0.3350	1.4360
170	270.94	59.40	0.1461	70.065	119.548	0.13287	0.21145	0.4148	0.3490	1.4791
175	287.74	58.43	0.1353	72.090	119.853	0.13598	0.21124	0.4258	0.3655	1.5307
180	305.33	57.40	0.1252	74.156	120.101	0.13914	0.21096	0.4387	0.3852	1.5934
185	323.74	56.30	0.1157	76.268	120.282	0.14233	0.21061	0.4544	0.4092	1.6709
190	343.00	55.14	0.1067	78.436	120.384	0.14558	0.21015	0.4737	0.4392	1.7689
195	363.15	53.88	0.0981	80.669	120.391	0.14890	0.20958	0.4983	0.4779	1.8967
200	384.22	52.51	0.0899	82.982	120.281	0.15231	0.20885	0.5309	0.5297	2.0694
205	406.27	50.99	0.0820	85.398	120.019	0.15584	0.20793	0.5764	0.6031	2.3154
210	429.35	49.28	0.0743	87.952	119.554	0.15954	0.20673	0.6450	0.7152	2.6920
215	453.52	47.28	0.0666	90.703	118.794	0.16350	0.20513	0.7613	0.9082	3.3385
220	478.88	44.79	0.0587	93.782	117.552	0.16789	0.20287	1.007	1.319	4.702
225	505.55	41.25	0.0497	97.579	115.267	0.17329	0.19912	1.893	2.784	9.484
228.85[c]	527.20	30.54	0.0327	106.210	106.210	0.18569	0.18569	∞	∞	∞

[b]Normal boiling point

[c]Critical point

Table 8.16 Comparative Refrigerant Performance per Ton of Refrigeration [2017F, Ch 29, Tbl 8]

Refrigerant No.	Chemical Name or Composition (% by mass)	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb	Refrigerant Circulated, lb/min	Liquid Circulated, gal/min	Specific Volume of Suction Gas, ft^3/lb	Compressor Displacement, ft^3/min	Power Consumption, hp	Coefficient of Performance	Compressor Discharge Temp., °F
Evaporator –25°F/Condenser 86°F												
744	Carbon dioxide	195.7	1046.2	5.35	56.8	3.52	0.711	0.457	1.61	2.779	1.698	196.3
170	Ethane	146.8	675.1	4.6	66.0	3.03	1.314	0.878	2.66	2.805	1.681	136.2
1270	Propylene	28.8	189.3	6.57	115.7	1.73	0.416	3.63	6.28	1.637	2.88	120.3
507A	R-125/143a (50/50)	28.8	211.7	7.34	43.5	4.60	0.54	1.52	6.98	1.833	2.573	100.6
404A	R-125/143a/134a (44/52/4)	27.6	206.1	7.46	45.1	4.44	0.521	1.61	7.13	1.817	2.595	102.1
502	R-22/115 (48.8/51.2)	26.5	189.2	7.14	42.1	4.76	0.48	1.48	7.06	1.722	2.739	106.3
22	Chlorodifluoromethane	22.1	172.9	7.81	66.8	3.00	0.307	2.32	6.95	1.589	2.967	149.8
717	Ammonia	16.0	169.3	10.61	463.9	0.43	0.087	16.7	7.19	1.569	3.007	285.6
Evaporator 20°F/Condenser 86°F												
744	Carbon dioxide	421.9	1046.2	2.48	55.7	3.59	0.726	0.203	0.73	1.342	3.514	142.3
170	Ethane	293.6	675.1	2.3	70.1	2.85	1.238	0.421	1.20	1.314	3.588	115.8
32	Difluoromethane	94.7	279.6	2.95	111.2	1.80	0.229	0.902	1.62	0.797	5.924	139.4
410A	R-32/125 (50/50)	93.2	273.6	2.94	73.5	2.72	0.316	0.651	1.77	0.815	5.78	115.8
507A	R-125/143a (50/50)	72.9	211.7	2.9	49.4	4.05	0.476	0.616	2.50	0.848	5.564	93.5
404A	R-125/143a/134a (44/52/4)	70.5	206.1	2.92	51.1	3.92	0.46	0.649	2.54	0.842	5.598	94.3
1270	Propylene	69.1	189.3	2.74	126.6	1.58	0.381	1.58	2.50	0.79	5.975	102.8
502	R-22/115 (48.8/51.2)	66.3	189.2	2.86	47.1	4.25	0.429	0.619	2.63	0.813	5.799	95.8
22	Chlorodifluoromethane	57.8	172.9	2.99	71.3	2.80	0.287	0.935	2.62	0.772	6.105	118.0
407C	R-32/125/134a (23/25/52)	57.5	183.7	3.19	71.9	2.78	0.296	0.942	2.62	0.795	5.93	111.0
290	Propane	55.8	156.5	2.8	124.1	1.61	0.399	1.89	3.05	0.787	5.987	94.8
717	Ammonia	48.2	169.3	3.51	478.5	0.42	0.084	5.91	2.47	0.754	6.254	179.8
1234yf	2,3,3,3-tetrafluoropropene*	36.3	113.6	3.13	51.8	3.86	0.43	1.15	4.44	0.809	5.835	86.0

Table 8.16 Comparative Refrigerant Performance per Ton of Refrigeration [2017F, Ch 29, Tbl 8] ***(Continued)***

Refrigerant No.	Chemical Name or Composition (% by mass)	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb	Refrigerant Circulated, lb/min	Liquid Circulated, gal/min	Specific Volume of Suction Gas, ft^3/lb	Compressor Displacement, ft^3/min	Power Consumption, hp	Coefficient of Performance	Compressor Discharge Temp., °F
134a	Tetrafluoroethane	33.1	111.7	3.37	65.8	3.04	0.307	1.41	4.28	0.778	6.063	94.7
1234ze(E)	Trans-1,3,3,3-tetrafluoropropene*	24.4	83.9	3.44	60.0	3.33	0.349	1.74	5.81	0.782	6.03	86.0
600a	Isobutane*	17.9	58.7	3.29	119.5	1.67	0.368	4.78	7.99	0.764	6.171	86.0
Evaporator 45°F/Condenser 86°F												
32	Difluoromethane	147.7	279.6	1.89	112.2	1.78	0.223	0.577	1.03	0.445	10.602	116.4
410A	R-32/125 (50/50)	145.0	273.6	1.89	75.2	2.66	0.308	0.416	1.11	0.455	10.379	103.7
502	R-22/115 (48.8/51.2)	102.0	189.2	1.85	49.6	4.03	0.407	0.404	1.63	0.451	10.474	91.8
407C	R-32/125/134a (23/25/52)	92.8	183.7	1.98	74.7	2.68	0.284	0.588	1.57	0.443	10.655	102.7
22	Chlorodifluoromethane	90.8	172.9	1.9	73.5	2.72	0.279	0.604	1.64	0.433	10.885	104.5
290	Propane	85.3	156.5	1.84	130.7	1.53	0.379	1.26	1.92	0.439	10.743	90.7
717	Ammonia	81.0	169.3	2.09	484.9	0.41	0.083	3.61	1.49	0.421	11.186	137.4
500	R-12/152a (73.8/26.2)	66.5	127.6	1.92	64.7	3.09	0.331	0.725	2.24	0.432	10.925	94.2
1234yf	2,3,3,3-tetrafluoropropene*	58.1	113.6	1.96	55.5	3.61	0.402	0.726	2.62	0.444	10.623	86.0
12	Dichlorodifluoromethane	56.3	107.9	1.92	54.6	3.67	0.34	0.719	2.64	0.429	11.004	91.6
134a	Tetrafluoroethane	54.7	111.7	2.04	69.2	2.89	0.292	0.868	2.51	0.433	10.903	90.6
1234ze(E)	Trans-1,3,3,3-tetrafluoropropene*	40.6	83.9	2.06	64.1	3.12	0.327	1.07	3.34	0.433	10.899	86.0
600a	Isobutane*	29.2	58.7	2.01	127.4	1.57	0.345	3.01	4.72	0.425	11.084	86.0
600	Butane*	19.5	41.1	2.11	140.5	1.42	0.301	4.57	6.50	0.42	11.226	86.0
123	Dichlorotrifluoroethane	6.5	15.9	2.44	66.9	2.99	0.246	5.3	15.85	0.414	11.397	86.0
113	Trichlorotrifluoroethane*	3.1	7.9	2.57	59.2	3.38	0.26	9.41	31.81	0.413	11.409	86.0

*Superheat required
Source: Data from NIST CYCLE_D 4.0, zero subcool, zero superheat unless noted, no line losses, 100% efficiencies, average temperatures.

Table 8.17 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 404A (Single- or High-Stage Applications)

[2014R, Ch 1, Tbl 6]

Line Size	Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.55 psi)						Liquid Lines		
	Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD	–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
	Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
	0.64	0.97	1.41	1.96	2.62	3.44	3.55	3.55	3.55	3.55	3.55	3.55		Δp = 3.6	Δp = 17.4
1/2	0.05	0.09	0.15	0.24	0.36	0.53	0.56	0.61	0.65	0.70	0.75	0.79	1.3	2.6	6.09
5/8	0.09	0.16	0.28	0.44	0.68	1.00	1.04	1.14	1.23	1.31	1.40	1.48	2.1	4.9	11.39
3/4	0.15	0.28	0.47	0.76	1.15	1.70	1.77	1.93	2.09	2.23	2.38	2.51	3.1	8.1	18.87
7/8	0.24	0.43	0.73	1.17	1.78	2.63	2.73	2.98	3.22	3.44	3.66	3.87	4.4	12.8	29.81
1 1/8	0.49	0.88	1.49	2.37	3.61	5.31	5.52	6.01	6.49	6.96	7.40	7.81	7.5	25.9	60.17
1 3/8	0.86	1.54	2.59	4.13	6.28	9.23	9.60	10.46	11.29	12.10	12.87	13.58	11.4	45.2	104.41
1 5/8	1.36	2.44	4.10	6.53	9.92	14.57	15.14	16.49	17.80	19.07	20.28	21.41	16.1	71.4	164.68
2 1/8	2.83	5.07	8.52	13.53	20.51	30.06	31.29	34.08	36.80	39.43	41.93	44.26	28.0	147.9	339.46
2 5/8	5.03	8.97	15.07	23.88	36.16	52.96	55.04	59.95	64.74	69.36	73.76	77.85	43.2	261.2	597.42
3 1/8	8.05	14.34	24.02	38.05	57.56	84.33	87.66	95.48	103.11	110.47	117.48	124.00	61.7	416.2	950.09
3 5/8	11.98	21.31	35.73	56.53	85.39	125.18	129.88	141.46	152.76	163.67	174.05	183.71	83.5	618.4	1407.96
4 1/8	16.93	30.09	50.32	79.66	120.39	176.20	182.83	199.13	215.05	230.40	245.01	258.61	108.5	871.6	1982.40
5 1/8	30.35	53.85	89.97	142.32	214.82	313.91	325.75	354.81	383.16	410.51	436.55	460.78	169.1	1554.2	3525.99
6 1/8	48.89	86.74	144.47	228.50	344.70	502.77	521.74	568.28	613.69	657.49	699.20	738.00	243.1	2497.7	5648.67
8 1/8	101.60	179.88	299.39	472.46	710.75	1037.34	1076.62	1172.66	1266.36	1356.75	1442.81	1522.89	424.6	5159.7	11660.71

Table 8.17 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 404A (Single- or High-Stage Applications)
[2014R, Ch 1, Tbl 6] ***(Continued)***

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.55 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.64	0.97	1.41	1.96	2.62	3.44	3.55	3.55	3.55	3.55	3.55	3.55		Δp = 3.6	Δp = 17.4
Steel																
IPS	SCH															
3/8	80	0.04	0.07	0.11	0.18	0.27	0.39	0.40	0.44	0.47	0.51	0.54	0.57	1.3	1.9	4.3
1/2	80	0.08	0.14	0.22	0.35	0.53	0.76	0.79	0.86	0.93	0.99	1.06	1.12	2.1	3.8	8.5
3/4	80	0.18	0.31	0.51	0.79	1.18	1.71	1.78	1.93	2.09	2.24	2.38	2.51	3.9	8.6	19.2
1	80	0.35	0.60	0.99	1.55	2.32	3.36	3.48	3.79	4.09	4.38	4.66	4.92	6.5	16.9	37.5
1 1/4	80	0.75	1.30	2.13	3.33	4.97	7.20	7.45	8.12	8.77	9.39	9.99	10.54	11.6	36.3	80.3
1 1/2	80	1.14	1.98	3.26	5.08	7.57	10.96	11.35	12.37	13.35	14.31	15.21	16.06	16.0	55.3	122.3
2	40	2.65	4.61	7.55	11.78	17.57	25.45	26.36	28.71	31.01	33.22	35.33	37.29	30.4	128.4	283.5
2 1/2	40	4.23	7.34	12.04	18.74	27.94	40.49	41.93	45.67	49.32	52.84	56.19	59.31	43.3	204.7	450.9
3	40	7.48	12.98	21.26	33.11	49.37	71.55	74.10	80.71	87.16	93.38	99.31	104.82	66.9	361.6	796.8
4	40	15.30	26.47	43.34	67.50	100.66	145.57	150.75	164.20	177.32	189.98	202.03	213.24	115.3	735.6	1623.0
5	40	27.58	47.78	78.24	121.87	181.32	262.52	272.21	296.49	320.19	343.04	364.80	385.05	181.1	1328.2	2927.2
6	40	44.58	77.26	126.52	197.09	293.24	424.04	439.72	478.94	517.21	554.13	589.28	621.99	261.7	2148.0	4728.3
8	40	91.40	158.09	258.81	402.66	599.91	867.50	898.42	978.56	1056.75	1132.18	1203.99	1270.82	453.2	4394.4	9674.1
10	40	165.52	286.19	468.14	728.40	1083.73	1569.40	1625.34	1770.31	1911.78	2048.23	2178.15	2299.05	714.4	7938.5	17,477.4
12	ID[b]	264.36	457.37	748.94	1163.62	1733.87	2507.30	2600.54	2832.50	3058.84	3277.16	3485.04	3678.47	1024.6	12,681.8	27,963.7
14	30	342.81	592.13	968.21	1506.59	2244.98	3246.34	3362.07	3661.96	3954.59	4236.83	4505.59	4755.67	1249.2	16,419.6	36,152.5
16	30	493.87	852.84	1395.24	2171.13	3230.27	4678.48	4845.26	5277.44	5699.16	6105.92	6493.24	6853.65	1654.7	23,662.2	52,101.2

Table 8.18 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 507A (Single- or High-Stage Applications)
[2014R, Ch 1, Tbl 7]

Line Size	Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.65 psi)						Liquid Lines		
	Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD	−60	−40	−20	0	20	40	−60	−40	−20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
	Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
	0.67	1.01	1.46	2.02	2.71	3.6	3.65	3.65	3.65	3.65	3.65	3.65		Δp = 3.65	Δp = 17.8
1/2	0.05	0.09	0.15	0.24	0.37	0.55	0.55	0.60	0.65	0.70	0.75	0.79	1.3	2.5	5.96
5/8	0.09	0.17	0.28	0.45	0.69	1.02	1.04	1.13	1.22	1.31	1.40	1.48	2.0	4.7	11.13
3/4	0.16	0.28	0.48	0.77	1.17	1.74	1.76	1.92	2.08	2.24	2.38	2.52	3.0	7.9	18.45
7/8	0.25	0.44	0.74	1.18	1.81	2.68	2.72	2.97	3.22	3.45	3.68	3.89	4.2	12.5	29.14
1 1/8	0.50	0.90	1.51	2.40	3.66	5.41	5.48	5.99	6.49	6.96	7.41	7.84	7.2	25.2	58.74
1 3/8	0.88	1.57	2.63	4.18	6.35	9.41	9.54	10.42	11.28	12.11	12.90	13.63	11.0	44.0	102.09
1 5/8	1.39	2.48	4.17	6.61	10.04	14.84	15.04	16.43	17.79	19.09	20.34	21.50	15.6	69.5	161.04
2 1/8	2.91	5.17	8.65	13.70	20.76	30.66	31.03	33.90	36.70	39.40	41.96	44.36	27.1	144.0	331.97
2 5/8	5.15	9.14	15.27	24.19	36.62	54.04	54.69	59.74	64.68	69.43	73.96	78.18	41.8	254.3	584.28
3 1/8	8.24	14.61	24.40	38.55	58.29	85.90	86.95	94.98	102.84	110.39	117.58	124.29	59.6	405.2	929.27
3 5/8	12.27	21.75	36.22	57.15	86.47	127.52	129.07	140.99	152.66	163.87	174.54	184.50	80.6	601.0	1377.19
4 1/8	17.34	30.66	51.13	80.55	121.93	179.33	181.70	198.48	214.91	230.69	245.71	259.74	104.8	847.0	1935.27
5 1/8	31.09	54.88	91.25	143.93	217.14	319.89	323.48	353.35	382.60	410.70	437.44	462.40	163.3	1513.6	3449.44
6 1/8	49.99	88.20	146.87	230.77	348.36	512.29	518.62	566.52	613.40	658.45	701.32	741.34	234.8	2427.4	5526.55
8 1/8	103.91	182.97	303.62	477.80	720.09	1057.14	1070.49	1169.35	1266.13	1359.11	1447.60	1530.21	410.1	5019.4	11,383.18

Table 8.18 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 507A (Single- or High-Stage Applications) [2014R, Ch 1, Tbl 7] ***(Continued)***

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.65 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
		−60	−40	−20	0	20	40	−60	−40	−20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.67	1.01	1.46	2.02	2.71	3.6	3.65	3.65	3.65	3.65	3.65	3.65		Δp = 3.65	Δp = 17.8
Steel																
IPS	SCH															
3/8	80	0.04	0.07	0.12	0.18	0.27	0.39	0.40	0.43	0.47	0.51	0.54	0.57	1.2	1.9	4.2
1/2	80	0.08	0.14	0.23	0.35	0.53	0.77	0.78	0.86	0.93	0.99	1.06	1.12	2.1	3.7	8.3
3/4	80	0.18	0.31	0.51	0.80	1.20	1.74	1.76	1.93	2.09	2.24	2.39	2.52	3.8	8.4	18.7
1	80	0.35	0.61	1.01	1.57	2.34	3.41	3.45	3.77	4.08	4.38	4.67	4.94	6.3	16.4	36.6
1 1/4	80	0.76	1.32	2.16	3.36	5.02	7.32	7.39	8.08	8.74	9.39	10.00	10.57	11.2	35.2	78.4
1 1/2	80	1.16	2.01	3.29	5.12	7.65	11.15	11.26	12.30	13.32	14.30	15.23	16.10	15.5	53.8	119.4
2	40	2.70	4.68	7.65	11.89	17.76	25.88	26.15	28.56	30.93	33.20	35.36	37.38	29.4	124.8	276.7
2 1/2	40	4.31	7.45	12.18	18.93	28.24	41.17	41.59	45.43	49.19	52.80	56.24	59.45	41.9	198.9	440.6
3	40	7.63	13.19	21.54	33.45	49.90	72.75	73.50	80.29	86.93	93.32	99.39	105.06	64.6	351.5	777.9
4	40	15.57	26.88	43.92	68.12	101.75	148.00	149.53	163.33	176.85	189.84	202.20	213.74	111.4	714.9	1586.3
5	40	28.10	48.52	79.19	122.99	183.27	266.91	270.00	294.93	319.34	342.79	365.11	385.94	174.9	1290.8	2857.5
6	40	45.48	78.45	128.06	198.91	296.40	431.69	436.14	476.41	515.85	553.73	589.78	623.44	252.8	2087.5	4622.0
8	40	93.13	160.66	261.94	406.93	606.38	882.01	891.10	973.39	1053.96	1131.36	1205.02	1273.79	437.7	4270.8	9443.9
10	40	168.64	290.60	473.82	735.12	1095.44	1595.65	1612.10	1760.97	1906.72	2046.75	2180.00	2304.41	690.0	7715.1	17,086.7
12	ID[b]	269.75	464.87	758.01	1174.36	1752.56	2553.03	2579.36	2817.55	3050.75	3274.79	3488.00	3687.06	989.6	12,324.9	27,298.3
14	30	349.22	601.87	979.92	1520.49	2269.19	3300.65	3334.69	3642.64	3944.13	4233.77	4509.42	4766.76	1206.5	15,957.5	35,292.2
16	30	503.20	866.37	1414.32	2191.17	3265.09	4756.74	4805.79	5249.60	5684.09	6101.51	6498.76	6869.63	1598.2	22,996.2	50,861.5

Table 8.19 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 410A (Single- or High-Stage Applications)

[2014R, Ch 1, Tbl 8]

Line Size	Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 4.75 psi)						Liquid Lines		
	Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD	–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
	Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
	0.84	1.27	1.85	2.57	3.46	4.5	4.75	4.75	4.75	4.75	4.75	4.75		Δp = 4.75	Δp = 23.3
1/2	0.10	0.17	0.27	0.42	0.62	0.89	1.13	1.17	1.22	1.26	1.30	1.33	2.0	4.6	10.81
5/8	0.18	0.31	0.51	0.79	1.17	1.67	2.11	2.20	2.29	2.36	2.43	2.49	3.2	8.6	20.24
3/4	0.31	0.53	0.87	1.35	2.00	2.84	3.59	3.74	3.88	4.02	4.14	4.23	4.7	14.3	33.53
7/8	0.48	0.83	1.35	2.08	3.08	4.39	5.53	5.76	5.99	6.19	6.38	6.52	6.7	22.6	52.92
1 1/8	0.98	1.69	2.74	4.22	6.23	8.86	11.16	11.64	12.09	12.50	12.88	13.17	11.4	45.8	106.59
1 3/8	1.72	2.95	4.78	7.34	10.85	15.41	19.39	20.21	21.00	21.72	22.37	22.88	17.4	79.7	185.04
1 5/8	2.73	4.67	7.56	11.61	17.14	24.28	30.63	31.92	33.16	34.30	35.33	36.14	24.6	125.9	291.48
2 1/8	5.69	9.71	15.71	24.05	35.45	50.19	63.20	65.88	68.44	70.78	72.90	74.57	42.8	260.7	601.13
2 5/8	10.09	17.17	27.74	42.45	62.53	88.43	111.20	115.90	120.41	124.53	128.25	131.20	66.0	459.7	1056.39
3 1/8	16.15	27.44	44.24	67.77	99.53	140.83	177.12	184.62	191.80	198.36	204.29	208.98	94.2	733.0	1680.52
3 5/8	24.06	40.84	65.81	100.50	147.66	208.65	262.44	273.54	284.19	293.90	302.70	309.64	127.4	1087.5	2491.00
4 1/8	33.98	57.58	92.66	141.61	208.22	293.70	369.45	385.08	400.07	413.75	426.13	435.90	165.7	1530.2	3500.91
5 1/8	60.95	103.03	165.73	253.05	370.82	523.21	658.32	686.18	712.88	737.26	759.31	776.72	258.2	2729.8	6228.40
6 1/8	98.05	166.00	266.14	405.75	594.85	839.82	1054.47	1099.10	1141.87	1180.91	1216.24	1244.13	371.1	4383.7	9980.43
8 1/8	203.77	344.31	551.73	840.04	1229.69	1733.02	2176.50	2268.62	2356.89	2437.49	2510.41	2567.98	648.3	9049.5	20,561.73

Table 8.19 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 410A (Single- or High-Stage Applications) *(Continued)*
[2014R, Ch 1, Tbl 8] ***(Continued)***

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 4.75 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.84	1.27	1.85	2.57	3.46	4.5	4.75	4.75	4.75	4.75	4.75	4.75		Δp = 4.75	Δp = 23.3
Steel																
IPS	SCH															
3/8	80	0.08	0.13	0.21	0.32	0.46	0.65	0.81	0.84	0.88	0.91	0.93	0.95	1.9	3.4	7.6
1/2	80	0.16	0.26	0.41	0.62	0.91	1.27	1.59	1.66	1.73	1.78	1.84	1.88	3.2	6.7	15.0
3/4	80	0.35	0.59	0.93	1.41	2.04	2.86	3.59	3.74	3.88	4.02	4.14	4.23	6.0	15.1	33.6
1	80	0.69	1.15	1.83	2.75	4.00	5.59	7.02	7.32	7.60	7.86	8.10	8.28	10.0	29.5	65.8
1 1/4	80	1.49	2.48	3.92	5.90	8.58	12.00	15.03	15.67	16.28	16.83	17.34	17.74	17.7	63.3	140.9
1 1/2	80	2.28	3.79	5.98	9.01	13.06	18.27	22.89	23.86	24.79	25.64	26.41	27.01	24.4	96.6	214.7
2	40	5.30	8.80	13.89	20.91	30.32	42.43	53.16	55.41	57.57	59.54	61.32	62.73	46.4	224.2	498.0
2 1/2	40	8.46	14.02	22.13	33.29	48.23	67.48	84.56	88.14	91.57	94.70	97.53	99.77	66.2	356.5	793.0
3	40	14.98	24.81	39.10	58.81	85.22	119.26	149.44	155.76	161.82	167.36	172.37	176.32	102.2	630.0	1398.4
4	40	30.58	50.56	79.68	119.77	173.76	242.63	304.02	316.88	329.21	340.47	350.66	358.70	176.1	1284.6	2851.7
5	40	55.19	91.27	143.84	216.23	312.97	437.56	548.97	572.20	594.46	614.79	633.19	647.71	276.5	2313.7	5137.0
6	40	89.34	147.57	232.61	349.71	506.16	707.69	886.76	924.29	960.25	993.09	1022.80	1046.26	399.6	3741.9	8308.9
8	40	182.90	301.82	475.80	715.45	1035.51	1445.92	1811.80	1888.48	1961.96	2029.05	2089.76	2137.68	692.0	7655.3	16,977.6
10	40	331.22	546.64	860.67	1292.44	1870.67	2615.83	3277.74	3416.46	3549.40	3670.77	3780.59	3867.29	1090.7	13,829.2	30,716.4
12	ID[b]	529.89	873.19	1376.89	2064.68	2992.85	4185.32	5244.38	5466.33	5679.03	5873.23	6048.94	6187.65	1564.3	22,125.4	49,074.9
14	30	685.86	1130.48	1779.99	2673.23	3875.08	5410.92	6780.14	7067.08	7342.06	7593.13	7820.29	7999.63	1907.2	28,647.5	63,445.8
16	30	988.28	1628.96	2569.05	3852.37	5575.79	7797.98	9771.20	10,184.73	10,581.02	10,942.85	11,270.23	11,528.68	2526.4	41,220.5	91,435.1

Table 8.20 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 407C (Single- or High-Stage Applications)

[2014R, Ch 1, Tbl 9]

Line Size	Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.3 psi)						Liquid Lines		
	Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
Type L Copper, OD	–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
	Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
	0.435	0.7	1.06	1.55	2.16	2.92	3.3	3.3	3.3	3.3	3.3	3.3		Δp = 3.5	Δp = 16.9
1/2	0.04	0.08	0.14	0.23	0.36	0.54	0.71	0.75	0.78	0.82	0.86	0.89	2.1	3.8	8.90
5/8	0.08	0.15	0.26	0.43	0.68	1.02	1.33	1.40	1.47	1.54	1.61	1.67	3.4	7.1	16.68
3/4	0.14	0.26	0.45	0.74	1.16	1.74	2.26	2.38	2.50	2.62	2.73	2.84	4.9	11.8	27.66
7/8	0.21	0.40	0.70	1.15	1.79	2.68	3.48	3.67	3.86	4.05	4.22	4.38	6.9	18.7	43.73
1 1/8	0.44	0.82	1.42	2.33	3.63	5.42	7.05	7.43	7.82	8.19	8.53	8.86	11.8	37.9	88.21
1 3/8	0.77	1.43	2.48	4.07	6.33	9.45	12.25	12.92	13.59	14.23	14.83	15.40	18.0	66.2	153.45
1 5/8	1.23	2.27	3.93	6.44	10.00	14.93	19.33	20.39	21.44	22.46	23.40	24.30	25.5	104.7	241.93
2 1/8	2.56	4.74	8.18	13.37	20.72	30.90	39.99	42.17	44.35	46.45	48.40	50.27	44.4	217.1	499.23
2 5/8	4.55	8.42	14.49	23.64	36.62	54.50	70.56	74.41	78.25	81.96	85.40	88.70	68.5	383.7	879.85
3 1/8	7.30	13.47	23.15	37.76	58.34	86.88	112.34	118.47	124.59	130.50	135.97	141.22	97.7	611.3	1401.50
3 5/8	10.90	20.08	34.44	56.15	86.64	128.89	166.39	175.47	184.54	193.29	201.39	209.17	132.2	907.9	2076.59
4 1/8	15.42	28.37	48.62	79.21	122.10	181.34	234.63	247.42	260.22	272.56	283.98	294.95	171.8	1281.5	2923.40
5 1/8	27.70	50.85	86.97	141.60	218.05	323.50	417.91	440.69	463.48	485.46	505.80	525.33	267.8	2288.8	5209.13
6 1/8	44.70	81.91	140.04	227.86	350.42	519.62	670.58	707.15	743.71	778.97	811.62	842.96	385.0	3676.9	8344.10
8 1/8	92.98	170.14	290.93	471.55	725.11	1072.54	1383.29	1458.72	1534.15	1606.88	1674.23	1738.88	672.4	7599.4	17,220.64

Table 8.20 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 407C (Single- or High-Stage Applications) [2014R, Ch 1, Tbl 9] ***(Continued)***

Line Size		Suction Lines (Δt = 2°F)						Discharge Lines (Δt = 1°F, Δp = 3.3 psi)						Liquid Lines		
		Saturated Suction Temperature, °F						Saturated Suction Temperature, °F						See note a		
		–60	–40	–20	0	20	40	–60	–40	–20	0	20	40	Velocity = 100 fpm	Δt = 1°F Drop	Δt = 5°F Drop
		Corresponding Δp, psi/100 ft						Corresponding Δp, psi/100 ft								
		0.435	0.7	1.06	1.55	2.16	2.92	3.3	3.3	3.3	3.3	3.3	3.3		Δp = 3.5	Δp = 16.9
Steel																
IPS	SCH															
3/8	80	0.04	0.07	0.11	0.18	0.27	0.40	0.52	0.55	0.57	0.60	0.63	0.65	2.0	2.9	6.4
1/2	80	0.07	0.13	0.22	0.35	0.54	0.79	1.02	1.07	1.13	1.18	1.23	1.28	3.4	5.7	12.6
3/4	80	0.16	0.30	0.50	0.80	1.22	1.79	2.29	2.42	2.54	2.66	2.78	2.88	6.2	12.8	28.4
1	80	0.32	0.58	0.98	1.57	2.38	3.50	4.50	4.74	4.99	5.22	5.44	5.65	10.3	25.1	55.6
1 1/4	80	0.69	1.25	2.10	3.37	5.12	7.50	9.63	10.15	10.68	11.18	11.65	12.10	18.4	53.7	118.9
1 1/2	80	1.06	1.91	3.21	5.13	7.79	11.44	14.66	15.46	16.26	17.03	17.74	18.43	25.4	82.0	181.1
2	40	2.49	4.46	7.47	11.93	18.13	26.57	34.04	35.89	37.75	39.54	41.20	42.79	48.1	190.3	420.6
2 1/2	40	3.97	7.11	11.90	19.01	28.83	42.25	54.25	57.21	60.16	63.02	65.66	68.19	68.6	303.2	669.0
3	40	7.04	12.59	21.05	33.59	50.94	74.66	95.76	100.99	106.21	111.24	115.90	120.38	106.0	535.7	1182.3
4	40	14.38	25.70	42.97	68.47	103.84	152.24	195.04	205.68	216.31	226.57	236.06	245.18	182.6	1092.0	2405.3
5	40	26.00	46.36	77.55	123.61	187.25	274.21	351.31	370.46	389.62	408.09	425.19	441.61	286.8	1969.0	4343.2
6	40	42.13	75.15	125.49	199.88	302.82	443.47	568.16	599.14	630.12	659.99	687.65	714.21	414.5	3184.3	7015.7
8	40	86.32	153.84	256.66	408.86	619.47	907.26	1162.36	1225.74	1289.12	1350.24	1406.83	1461.15	717.7	6514.5	14,334.3
10	40	156.54	278.57	464.86	739.58	1120.60	1638.95	2102.83	2217.49	2332.15	2442.72	2545.10	2643.38	1131.3	11,784.6	25,932.3
12	ID[b]	250.23	445.65	742.54	1183.19	1790.17	2622.17	3359.45	3542.64	3725.82	3902.46	4066.02	4223.03	1622.5	18,826.0	41,491.5
14	30	324.38	576.93	961.33	1529.58	2317.81	3395.13	4349.77	4586.95	4824.14	5052.85	5264.62	5467.92	1978.2	24,374.8	53,641.7
16	30	468.29	831.27	1385.24	2204.17	3340.17	4885.19	6258.81	6600.09	6941.37	7270.46	7575.17	7867.69	2620.4	35,126.4	77,305.8

Table 8.21 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 22 (Single- or High-Stage Applications)

[2014R, Ch 1, Tbl 3]

Line Size		Suction Lines ($\Delta t = 2°F$) Saturated Suction Temperature, °F					Discharge Lines ($\Delta t = 1°F$, $\Delta p = 3.05$ psi)		Line Size		Liquid Lines See notes a and b	
Type L Copper, OD		–40	–20	0	20	40	Saturated Suction Temperature, °F		Type L Copper, OD		Vel. = 100 fpm	$\Delta t = 1°F$
		Corresponding Δp, psi/100 ft										
		0.79	1.15	1.6	2.22	2.91	–40	40				$\Delta p = 3.05$
1/2		—	—	—	0.40	0.6	0.75	0.85	1/2		2.3	3.6
5/8		—	0.32	0.51	0.76	1.1	1.4	1.6	5/8		3.7	6.7
7/8		0.52	0.86	1.3	2.0	2.9	3.7	4.2	7/8		7.8	18.2
1 1/8		1.1	1.7	2.7	4.0	5.8	7.5	8.5	1 1/8		13.2	37.0
1 3/8		1.9	3.1	4.7	7.0	10.1	13.1	14.8	1 3/8		20.2	64.7
1 5/8		3.0	4.8	7.5	11.1	16.0	20.7	23.4	1 5/8		28.5	102.5
2 1/8		6.2	10.0	15.6	23.1	33.1	42.8	48.5	2 1/8		49.6	213.0
2 5/8		10.9	17.8	27.5	40.8	58.3	75.4	85.4	2 5/8		76.5	376.9
3 1/8		17.5	28.4	44.0	65.0	92.9	120.2	136.2	3 1/8		109.2	601.5
3 5/8		26.0	42.3	65.4	96.6	137.8	178.4	202.1	3 5/8		147.8	895.7
4 1/8		36.8	59.6	92.2	136.3	194.3	251.1	284.4	4 1/8		192.1	1263.2
Steel									**Steel**			
IPS	**SCH**								**IPS**	**SCH**		
1/2	40	—	0.38	0.58	0.85	1.2	1.5	1.7	1/2	80	3.8	5.7
3/4	40	0.50	0.8	1.2	1.8	2.5	3.3	3.7	3/4	80	6.9	12.8
1	40	0.95	1.5	2.3	3.4	4.8	6.1	6.9	1	80	11.5	25.2
1 1/4	40	2.0	3.2	4.8	7.0	9.9	12.6	14.3	1 1/4	80	20.6	54.1
1 1/2	40	3.0	4.7	7.2	10.5	14.8	19.0	21.5	1 1/2	80	28.3	82.6
2	40	5.7	9.1	13.9	20.2	28.5	36.6	41.4	2	40	53.8	192.0

Table 8.21 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 22 (Single- or High-Stage Applications)
[2014R, Ch 1, Tbl 3] ***(Continued)***

Line Size		Suction Lines (Δt = 2°F) Saturated Suction Temperature, °F					Discharge Lines (Δt = 1°F, Δp = 3.05 psi) Saturated Suction Temperature, °F		Line Size		Liquid Lines See notes a and b	
		–40	–20	0	20	40					Vel. = 100 fpm	Δt = 1°F
		Corresponding Δp, psi/100 ft										
		0.79	1.15	1.6	2.22	2.91	–40	40				Δp = 3.05
Steel									Steel			
IPS	SCH								IPS	SCH		
2 1/2	40	9.2	14.6	22.1	32.2	45.4	58.1	65.9	2 1/2	40	76.7	305.8
3	40	16.2	25.7	39.0	56.8	80.1	102.8	116.4	3	40	118.5	540.3
4	40	33.1	52.5	79.5	115.9	163.2	209.5	237.3	4	40	204.2	1101.2

Notes:

1. Table capacities are in tons of refrigeration.

   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length

   Δt = corresponding change in saturation temperature, °F per 100 ft

2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

   $$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.55}$$

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

   $$\Delta t = \text{Table } \Delta t \left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.8}$$

4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing Temperature, °F	Suction Line	Discharge Line
80	1.11	0.79
90	1.07	0.88
100	1.03	0.95
110	0.97	1.04
120	0.90	1.10
130	0.86	1.18
140	0.80	1.26

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Line pressure drop Δp is conservative; if subcooling is substantial or line is short, a smaller size line may be used. Applications with very little subcooling or very long lines may require a larger line.

Table 8.22 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 134a (Single- or High-Stage Applications)

[2014R, Ch 1, Tbl 5]

Line Size		Suction Lines (Δt = 2°F) Saturated Suction Temperature, °F					Discharge Lines (Δt = 1°F, Δp = 2.2 psi/100 ft) Saturated Suction Temperature, °F			Line Size		Liquid Lines (See notes a and b)	
Type L Copper, OD		0	10	20	30	40				Type L Copper, OD		Velocity = 100 fpm	Δt = 1°F
		Corresponding Δp, psi/100 ft											
		1.00	1.19	1.41	1.66	1.93	0	20	40				Δp = 2.2
1/2		0.14	0.18	0.23	0.29	0.35	0.54	0.57	0.59	1/2		2.13	2.79
5/8		0.27	0.34	0.43	0.54	0.66	1.01	1.07	1.12	5/8		3.42	5.27
7/8		0.71	0.91	1.14	1.42	1.75	2.67	2.81	2.94	7/8		7.09	14.00
1 1/8		1.45	1.84	2.32	2.88	3.54	5.40	5.68	5.95	1 1/8		12.10	28.40
1 3/8		2.53	3.22	4.04	5.02	6.17	9.42	9.91	10.40	1 3/8		18.40	50.00
1 5/8		4.02	5.10	6.39	7.94	9.77	14.90	15.70	16.40	1 5/8		26.10	78.60
2 1/8		8.34	10.60	13.30	16.50	20.20	30.80	32.40	34.00	2 1/8		45.30	163.00
2 5/8		14.80	18.80	23.50	29.10	35.80	54.40	57.20	59.90	2 5/8		69.90	290.00
3 1/8		23.70	30.00	37.50	46.40	57.10	86.70	91.20	95.50	3 1/8		100.00	462.00
3 5/8		35.10	44.60	55.80	69.10	84.80	129.00	135.00	142.00	3 5/8		135.00	688.00
4 1/8		49.60	62.90	78.70	97.40	119.43	181.00	191.00	200.00	4 1/8		175.00	971.00
5 1/8		88.90	113.00	141.00	174.00	213.00	323.00	340.00	356.00	—		—	—
6 1/8		143.00	181.00	226.00	280.00	342.00	518.00	545.00	571.00	—		—	—
Steel										Steel			
IPS	SCH									IPS	SCH		
1/2	80	0.22	0.28	0.35	0.43	0.53	0.79	0.84	0.88	1/2	80	3.43	4.38
3/4	80	0.51	0.64	0.79	0.98	1.19	1.79	1.88	1.97	3/4	80	6.34	9.91
1	80	1.00	1.25	1.56	1.92	2.33	3.51	3.69	3.86	1	80	10.50	19.50
1 1/4	40	2.62	3.30	4.09	5.03	6.12	9.20	9.68	10.10	1 1/4	80	18.80	41.80

Table 8.22 Suction, Discharge, and Liquid Line Capacities in Tons for Refrigerant 134a (Single- or High-Stage Applications) [2014R, Ch 1, Tbl 5] ***(Continued)***

Line Size		Suction Lines ($\Delta t = 2°F$) Saturated Suction Temperature, °F					Discharge Lines ($\Delta t = 1°F$, $\Delta p = 2.2$ psi/100 ft)			Line Size		Liquid Lines See notes a and b	
		0	10	20	30	40	Saturated Suction Temperature, °F					Velocity = 100 fpm	$\Delta t = 1°F$
		Corresponding Δp, psi/100 ft											
		1.00	1.19	1.41	1.66	1.93	0	20	40				$\Delta p = 2.2$
Steel										Steel			
IPS	SCH									IPS	SCH		
1 1/2	40	3.94	4.95	6.14	7.54	9.18	13.80	14.50	15.20	1 1/2	80	25.90	63.70
2	40	7.60	9.56	11.90	14.60	17.70	26.60	28.00	29.30	2	40	49.20	148.00
2 1/2	40	12.10	15.20	18.90	23.10	28.20	42.40	44.60	46.70	2 1/2	40	70.10	236.00
3	40	21.40	26.90	33.40	41.00	49.80	75.00	78.80	82.50	3	40	108.00	419.00
4	40	43.80	54.90	68.00	83.50	101.60	153.00	160.00	168.00	4	40	187.00	853.00

Notes:

1. Table capacities are in tons of refrigeration.

   Δp = pressure drop from line friction, psi per 100 ft of equivalent line length

   Δt = corresponding change in saturation temperature, °F per 100 ft

2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

   $$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.55}$$

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

   $$\Delta t = \text{Table } \Delta t\left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.8}$$

4. Values based on 105°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures.

Condensing Temperature, °F	Suction Line	Discharge Line
80	1.158	0.804
90	1.095	0.882
100	1.032	0.961
110	0.968	1.026
120	0.902	1.078
130	0.834	1.156

[a] Sizing shown is recommended where any gas generated in receiver must return up condensate line to the condenser without restricting condensate flow. Water-cooled condensers, where receiver ambient temperature may be higher than refrigerant condensing temperature, fall into this category.

[b] Line pressure drop Δp is conservative; if subcooling is substantial or line is short, a smaller size line may be used. Applications with very little subcooling or very long lines may require a larger line.

Table 8.23 Minimum Refrigeration Capacity in Tons for Oil Entrainment up Suction Risers (Type L Copper Tubing)
[2014R, Ch 1, Tbl 20]

Refrig-erant	Saturated Suction Temp., °F	Suction Gas Temp., °F	Pipe OD, in.											
			1/2	**5/8**	**3/4**	**7/8**	**1 1/8**	**1 3/8**	**1 5/8**	**2 1/8**	**2 5/8**	**3 1/8**	**3 5/8**	**4 1/8**
			Area, in^2											
			0.146	**0.233**	**0.348**	**0.484**	**0.825**	**1.256**	**1.780**	**3.094**	**4.770**	**6.812**	**9.213**	**11.970**
22	–40.0	–30.0	0.067	0.119	0.197	0.298	0.580	0.981	1.52	3.03	5.20	8.12	11.8	16.4
		–10.0	0.065	0.117	0.194	0.292	0.570	0.963	1.49	2.97	5.11	7.97	11.6	16.1
		10.0	0.066	0.118	0.195	0.295	0.575	0.972	1.50	3.00	5.15	8.04	11.7	16.3
	–20.0	–10.0	0.087	0.156	0.258	0.389	0.758	1.28	1.98	3.96	6.80	10.6	15.5	21.5
		10.0	0.085	0.153	0.253	0.362	0.744	1.26	1.95	3.88	6.67	10.4	15.2	21.1
		30.0	0.086	0.154	0.254	0.383	0.747	1.26	1.95	3.90	6.69	10.4	15.2	21.1
	0.0	10.0	0.111	0.199	0.328	0.496	0.986	1.63	2.53	5.04	8.66	13.5	19.7	27.4
		30.0	0.108	0.194	0.320	0.484	0.942	1.59	2.46	4.92	8.45	13.2	19.2	26.7
		50.0	0.109	0.195	0.322	0.486	0.946	1.60	2.47	4.94	8.48	13.2	19.3	26.8
	20.0	30.0	0.136	0.244	0.403	0.608	1.18	2.00	3.10	6.18	10.6	16.6	24.2	33.5
		50.0	0.135	0.242	0.399	0.603	1.17	1.99	3.07	6.13	10.5	16.4	24.0	33.3
		70.0	0.135	0.242	0.400	0.605	1.18	1.99	3.08	6.15	10.6	16.5	24.0	33.3
	40.0	50.0	0.167	0.300	0.495	0.748	1.46	2.46	3.81	7.60	13.1	20.4	29.7	41.3
		70.0	0.165	0.296	0.488	0.737	1.44	2.43	3.75	7.49	12.9	20.1	29.3	40.7
		90.0	0.165	0.296	0.488	0.738	1.44	2.43	3.76	7.50	12.9	20.1	29.3	40.7

Table 8.23 Minimum Refrigeration Capacity in Tons for Oil Entrainment up Suction Risers (Type L Copper Tubing)
[2014R, Ch 1, Tbl 20] *(Continued)*

Refrig-erant	Saturated Suction Temp., °F	Suction Gas Temp., °F	Pipe OD, in.											
			1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
			Area, in^2											
			0.146	0.233	0.348	0.484	0.825	1.256	1.780	3.094	4.770	6.812	9.213	11.970
134a	0.0	10.0	0.089	0.161	0.259	0.400	0.78	1.32	2.03	4.06	7.0	10.9	15.9	22.1
		30.0	0.075	0.135	0.218	0.336	0.66	1.11	1.71	3.42	5.9	9.2	13.4	18.5
		50.0	0.072	0.130	0.209	0.323	0.63	1.07	1.64	3.28	5.6	8.8	12.8	17.8
	10.0	20.0	0.101	0.182	0.294	0.453	0.88	1.49	2.31	4.61	7.9	12.4	18.0	25.0
		40.0	0.084	0.152	0.246	0.379	0.74	1.25	1.93	3.86	6.6	10.3	15.1	20.9
		60.0	0.081	0.147	0.237	0.366	0.71	1.21	1.87	3.73	6.4	10.0	14.6	20.2
	20.0	30.0	0.113	0.205	0.331	0.510	0.99	1.68	2.60	5.19	8.9	13.9	20.3	28.2
		50.0	0.095	0.172	0.277	0.427	0.83	1.41	2.17	4.34	7.5	11.6	17.0	23.6
		70.0	0.092	0.166	0.268	0.413	0.81	1.36	2.10	4.20	7.2	11.3	16.4	22.8
	30.0	40.0	0.115	0.207	0.335	0.517	1.01	1.70	2.63	5.25	9.0	14.1	20.5	28.5
		60.0	0.107	0.193	0.311	0.480	0.94	1.58	2.44	4.88	8.4	13.1	19.1	26.5
		80.0	0.103	0.187	0.301	0.465	0.91	1.53	2.37	4.72	8.1	12.7	18.5	25.6
	40.0	50.0	0.128	0.232	0.374	0.577	1.12	1.90	2.94	5.87	10.1	15.7	22.9	31.8
		70.0	0.117	0.212	0.342	0.528	1.03	1.74	2.69	5.37	9.2	14.4	21.0	29.1
		90.0	0.114	0.206	0.332	0.512	1.00	1.69	2.61	5.21	8.9	14.0	20.4	28.3

Notes:
1. Refrigeration capacity in tons is based on 90°F liquid temperature and superheat as indicated by listed temperature. For other liquid line temperatures, use correction factors in table at right.
2. Values computed using ISO 32 mineral oil for R-22. R-134a computed using ISO 32 ester-based oil.

Refrig-erant	Liquid Temperature, °F								
	50	60	70	80	100	110	120	130	140
22	1.17	1.14	1.10	1.06	0.98	0.94	0.89	0.85	0.80
134a	1.26	1.20	1.13	1.07	0.94	0.87	0.80	0.74	0.67

Table 8.24 Minimum Refrigeration Capacity in Tons for Oil Entrainment up Hot-Gas Risers (Type L Copper Tubing)

[2014R, Ch 1, Tbl 19]

Refrig-erant	Saturated Temp., °F	Discharge Gas Temp., °F	Pipe OD, in.											
			1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
			Area, in^2											
			0.146	**0.233**	**0.348**	**0.484**	**0.825**	**1.256**	**1.780**	**3.094**	**4.770**	**6.812**	**9.213**	**11.970**
22	80.0	110.0	0.235	0.421	0.695	1.05	2.03	3.46	5.35	10.7	18.3	28.6	41.8	57.9
		140.0	0.223	0.399	0.659	0.996	1.94	3.28	5.07	10.1	17.4	27.1	39.6	54.9
		170.0	0.215	0.385	0.635	0.960	1.87	3.16	4.89	9.76	16.8	26.2	38.2	52.9
	90.0	120.0	0.242	0.433	0.716	1.06	2.11	3.56	5.50	11.0	18.9	29.5	43.0	59.6
		150.0	0.226	0.406	0.671	1.01	1.97	3.34	5.16	10.3	17.7	27.6	40.3	55.9
		180.0	0.216	0.387	0.540	0.956	1.88	3.18	4.92	9.82	16.9	26.3	38.4	53.3
	100.0	130.0	0.247	0.442	0.730	1.10	2.15	3.83	5.62	11.2	19.3	30.1	43.9	60.8
		160.0	0.231	0.414	0.884	1.03	2.01	3.40	5.26	10.5	18.0	28.2	41.1	57.0
		190.0	0.220	0.394	0.650	0.982	1.91	3.24	3.00	9.96	17.2	26.8	39.1	54.2
	110.0	140.0	0.251	0.451	0.744	1.12	2.19	3.70	5.73	11.4	19.6	30.6	44.7	62.0
		170.0	0.235	0.421	0.693	1.05	2.05	3.46	3.35	10.7	18.3	28.6	41.8	57.9
		200.0	0.222	0.399	0.658	0.994	1.94	3.28	5.06	10.1	17.4	27.1	39.5	54.8
	120.0	150.0	0.257	0.460	0.760	1.15	2.24	3.78	5.85	11.7	20.0	31.3	45.7	63.3
		180.0	0.239	0.428	0.707	1.07	2.08	3.51	5.44	10.8	18.6	29.1	42.4	58.9
		210.0	0.225	0.404	0.666	1.01	1.96	3.31	5.12	10.2	17.6	27.4	40.0	55.5

Table 8.24 Minimum Refrigeration Capacity in Tons for Oil Entrainment up Hot-Gas Risers (Type L Copper Tubing) [2014R, Ch 1, Tbl 19] *(Continued)*

Refrig-erant	Saturated Temp., °F	Discharge Gas Temp., °F	Pipe OD, in.											
			1/2	5/8	3/4	7/8	1 1/8	1 3/8	1 5/8	2 1/8	2 5/8	3 1/8	3 5/8	4 1/8
			Area, in^2											
			0.146	**0.233**	**0.348**	**0.484**	**0.825**	**1.256**	**1.780**	**3.094**	**4.770**	**6.812**	**9.213**	**11.970**
134a	80.0	110.0	0.199	0.360	0.581	0.897	1.75	2.96	4.56	9.12	15.7	24.4	35.7	49.5
		140.0	0.183	0.331	0.535	0.825	1.61	2.72	4.20	8.39	14.4	22.5	32.8	45.6
		170.0	0.176	0.318	0.512	0.791	1.54	2.61	4.02	8.04	13.8	21.6	31.4	43.6
	90.0	120.0	0.201	0.364	0.587	0.906	1.76	2.99	4.61	9.21	15.8	24.7	36.0	50.0
		150.0	0.184	0.333	0.538	0.830	1.62	2.74	4.22	8.44	14.5	22.6	33.0	45.8
		180.0	0.177	0.320	0.516	0.796	1.55	2.62	4.05	8.09	13.9	21.7	31.6	43.9
	100.0	130.0	0.206	0.372	0.600	0.926	1.80	3.05	4.71	9.42	16.2	25.2	36.8	51.1
		160.0	0.188	0.340	0.549	0.848	1.65	2.79	4.31	8.62	14.8	23.1	33.7	46.8
		190.0	0.180	0.326	0.526	0.811	1.58	2.67	4.13	8.25	14.2	22.1	32.2	44.8
	110.0	140.0	0.209	0.378	0.610	0.942	1.83	3.10	4.79	9.57	16.5	25.7	37.4	52.0
		170.0	0.191	0.346	0.558	0.861	1.68	2.84	4.38	8.76	15.0	23.5	34.2	47.5
		200.0	0.183	0.331	0.534	0.824	1.61	2.72	4.19	8.38	14.4	22.5	32.8	45.5
	120.0	150.0	0.212	0.383	0.618	0.953	1.86	3.14	4.85	9.69	16.7	26.0	37.9	52.6
		180.0	0.194	0.351	0.566	0.873	1.70	2.88	4.44	8.88	15.3	23.8	34.7	48.2
		210.0	0.184	0.334	0.538	0.830	1.62	2.74	4.23	8.44	14.5	22.6	33.0	45.8

Notes:
1. Refrigeration capacity in tons based on saturated suction temperature of 20°F with 15°F superheat at indicated saturated condensing temperature with 15°F subcooling. For other saturated suction temperatures with 15°F superheat, use correction factors in the table at right.
2. Table computed using ISO 32 mineral oil for R-22, and ISO 32 ester-based oil for R-134a.

Refrigerant	Saturated Suction Temperature, °F			
	–40	–20	0	+40
22	0.92	0.95	0.97	1.02
134a	—	—	0.96	1.04

Table 8.25 Suction, Discharge, and Liquid Line Capacities in Tons for Ammonia (Single- or High-Stage Applications) [2014R, Ch 2, Tbl 2]

Steel Line Size		Suction Lines ($\Delta t = 1°F$) Saturated Suction Temperature, °F					Discharge Lines	Steel Line Size		Liquid Lines	
IPS	SCH	–40 $\Delta p = 0.31$	–20 $\Delta p = 0.49$	0 $\Delta p = 0.73$	20 $\Delta p = 1.06$	40 $\Delta p = 1.46$	$\Delta t = 1°F$ $\Delta p = 2.95$	IPS	SCH	Velocity = 100 fpm	$\Delta p = 2.0$ psi $\Delta t = 0.7°F$
3/8	80	—	—	—	—	—	—	3/8	80	8.6	12.1
1/2	80	—	—	—	—	—	3.1	1/2	80	14.2	24.0
3/4	80	—	—	—	2.6	3.8	7.1	3/4	80	26.3	54.2
1	80	—	2.1	3.4	5.2	7.6	13.9	1	80	43.8	106.4
1 1/4	40	3.2	5.6	8.9	13.6	19.9	36.5	1 1/4	80	78.1	228.6
1 1/2	40	4.9	8.4	13.4	20.5	29.9	54.8	1 1/2	80	107.5	349.2
2	40	9.5	16.2	26.0	39.6	57.8	105.7	2	40	204.2	811.4
2 1/2	40	15.3	25.9	41.5	63.2	92.1	168.5	2 1/2	40	291.1	1292.6
3	40	27.1	46.1	73.5	111.9	163.0	297.6	3	40	449.6	2287.8
4	40	55.7	94.2	150.1	228.7	333.0	606.2	4	40	774.7	4662.1
5	40	101.1	170.4	271.1	412.4	600.9	1095.2	5	40	—	—
6	40	164.0	276.4	439.2	667.5	971.6	1771.2	6	40	—	—
8	40	337.2	566.8	901.1	1366.6	1989.4	3623.0	8	40	—	—
10	40	611.6	1027.2	1634.3	2474.5	3598.0	—	10	40	—	—
12	ID*	981.6	1644.5	2612.4	3963.5	5764.6	—	12	ID*	—	—

Notes:

1. Table capacities are in tons of refrigeration.

   Δp = pressure drop due to line friction, psi per 100 ft of equivalent line length

   Δt = corresponding change in saturation temperature, °F per 100 ft

2. Line capacity for other saturation temperatures Δt and equivalent lengths L_e

$$\text{Line capacity} = \text{Table capacity}\left(\frac{\text{Table } L_e}{\text{Actual } L_e} \times \frac{\text{Actual } \Delta t}{\text{Table } \Delta t}\right)^{0.55}$$

3. Saturation temperature Δt for other capacities and equivalent lengths L_e

$$\Delta t = \text{Table } \Delta t\left(\frac{\text{Actual } L_e}{\text{Table } L_e}\right)\left(\frac{\text{Actual capacity}}{\text{Table capacity}}\right)^{1.8}$$

4. Values based on 90°F condensing temperature. Multiply table capacities by the following factors for other condensing temperatures:

Condensing Temperature, °F	Suction Lines	Discharge Lines
70	1.05	0.78
80	1.02	0.89
90	1.00	1.00
100	0.98	1.11

5. Discharge and liquid line capacities based on 20°F suction. Evaporator temperature is 0°F. The capacity is affected less than 3% when applied from –40 to +40°F extremes.

Table 8.26 Liquid Ammonia Line Capacities in Kilowatts [2014R, Ch 2, Tbl 3]

Nominal Size, mm	Pumped Liquid Overfeed Ratio			High-Pressure Liquid at 21 kPa[a]	Hot-Gas Defrost[a]	Equalizer High Side[b]	Thermosiphon Lubricant Cooling Lines Gravity Flow[c]		
	3:1	4:1	5:1				Supply	Return	Vent
40	513	387	308	1544	106	791	59	35	60
50	1175	879	703	3573	176	1055	138	88	106
65	1875	1407	1125	5683	324	1759	249	155	187
80	2700	2026	1620	10 150	570	3517	385	255	323
100	4800	3600	2880	—	1154	7034	663	413	586
125	—	—	—	—	2089	—	1041	649	1062
150	—	—	—	—	3411	—	1504	938	1869
200	—	—	—	—	—	—	2600	1622	3400

Source: Wile (1977).
[a]Rating for hot-gas branch lines under 30 m with minimum inlet pressure of 724 kPa (gage), defrost pressure of 483 kPa (gage), and –29°C evaporators designed for a 5.6 K temperature differential.
[b]Line sizes based on experience using total system evaporator kilowatts.
[c]From Frick Co. (1995). Values for line sizes above 100 mm are extrapolated.

Lubricants In Refrigerant Systems

Oil in refrigerant compressors lubricates, acts as coolant, and seals the suction from the discharge side. Oil mixes well with hydrocarbon refrigerants at higher temperatures; miscibility is reduced as temperature lowers. Oil leaves the compressor and dissolves into the refrigerant in the condenser, and passes through the liquid line to the evaporator where it separates. In higher temperature systems, it returns by gravity or is dragged by the returning vapor. Low temperature halocarbon systems need an oil separator at the compressor discharge. Oil return up vertical piping requires significant refrigerant velocity. Lubricants are generally not miscible with ammonia and separate easily out of the liquid. Oil separators at the discharge of compressors are essential. Oil must be periodically or continuously removed and returned to the compressor.

There is no ideal lubricant. For halocarbon refrigerants, there are mineral lubricants, both naphthenic and paraffinic, and synthetic lubricants, ester and glycol. Viscosity grades required vary with the temperature and the solubility of the refrigerant in the lubricant. Additives are used to enhance lubricant properties or impact new characteristics. They may be polar compounds, polymers, or compounds containing active elements such as sulfur or phosphorus. Lubricants should be dry; normally almost all hydrocarbon lubricants have a moisture content of about 30 ppm. Synthetic lubricants polyalkylene glycols (PAGs) are used commonly in automobile R-134a systems; polyalphaolefins (PAOs) are mainly used an immiscible oil in ammonia systems; polyol esters are used with HFC refrigerants in all types of compressors. Low pour point is essential for oils in ammonia systems.

When retrofitting to a new refrigerant, follow the recommendations of the equipment manufacturer on the lubricants that are suitable for use.

Mixing lubricants can cause serious problems; to extend equipment life, it is important to use lubricants approved or specified by the system or compressor manufacturer.

Table 8.27 Secondary Coolant Performance Comparisons [2014R, Ch 13, Tbl 1]

Secondary Coolant	Concentration (by Weight), %	Freeze Point, °F	gpm/ton[a]	Pressure Drop,[b] psi	Heat Transfer Coefficient[c] h_i, Btu/h·ft^2·°F
Propylene glycol	39	–5.1	2.56	2.91	205
Ethylene glycol	38	–6.9	2.76	2.38	406
Methanol	26	–5.3	2.61	2.05	473
Sodium chloride	23	–5.1	2.56	2.30	558
Calcium chloride	22	–7.8	2.79	2.42	566
Aqua ammonia	14	–7.0	2.48	2.44	541
Trichloroethylene	100	–123	7.44	2.11	432
d-Limonene	100	–142	6.47	1.48	321
Methylene chloride	100	–142	6.39	1.86	58
R-11	100	–168	7.61	2.08	428

[a]Based on inlet secondary coolant temperature at pump of 25°F.

[b]Based on one length of 16 ft tube with 1.06 in. ID and use of Moody Chart (1944) for an average velocity of 7 fps. Input/output losses equal one Vel. $H_D(V^2\rho/2g)$ for 7 fps velocity. Evaluations are at a bulk temperature of 20°F and a temperature range of 10°F.

[c]Based on curve fit equation for Kern's (1950) adaptation of Sieder and Tate's (1936) heat transfer equation using 16 ft tube for L/D = 181 and film temperature of 5°F lower than average bulk temperature with 7 fps velocity.

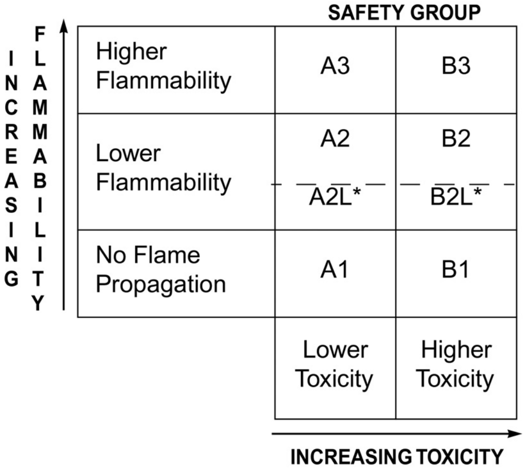
Table 8.28 Relative Pumping Energy Required* [2014R, Ch 13, Tbl 3]

Secondary Coolant	Energy Factor
Aqua ammonia	1.000
Methanol	1.078
Propylene glycol	1.142
Ethylene glycol	1.250
Sodium chloride	1.295
Calcium chloride	1.447
d-Limonene	2.406
Methylene chloride	3.735
Trichloroethylene	4.787
Aqua ammonia	1.000
Methanol	1.078
R-11	5.022

*Based on same pump pressure, refrigeration load, 20°F average temperature, 10°F range, and freezing point (for water-based secondary coolants) 20 to 23°F below lowest secondary coolant temperature.

9. REFRIGERANT SAFETY

(For more guidance, see ANSI/ASHRAE Standard 15, *Safety Standard for Refrigeration Systems*, and ANSI/ASHRAE Standard 34, *Designation and Safety Classification of Refrigerants.*)



* A2L and B2L are lower flammability refrigerants with a maximum burning velocity of ≤3.9 in./s (10 cm/s).

Figure 9.1 Refrigerant Safety Group Classification [Std 34-2016, Fig 1]

Table 9.1 Refrigerant Data and Safety Classifications [Std 34-2016, Tbl 4-1]

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	OEL[f], ppm v/v	Safety Group	RCL[c] (ppm v/v)	RCL[c] (lb/Mcf)	RCL[c] (g/m^3)	Highly Toxic or Toxic[d] Under Code Classification
Methane Series								
11	trichlorofluoromethane	CCl_3F	C1000	A1	1100	0.39	6.2	Neither
12	dichlorodifluoromethane	CCl_2F_2	1000	A1	18,000	5.6	90	Neither
12B1	bromochlorodifluoromethane	$CBrClF_2$						Neither
13	chlorotrifluoromethane	$CClF_3$	1000	Al				Neither
13B1	bromotrifluoromethane	$CBrF_3$	1000	Al				Neither
14[e]	tetrafluoromethane (carbon tetrafluoride)	CF_4	1000	Al	110,000	25	400	Neither
21	dichlorofluoromethane	$CHCl_2F$		B1				Toxic
22	chlorodifluoromethane	$CHClF_2$	1000	Al	59,000	13	210	Neither
23	trifluoromethane	CHF_3	1000	Al	41,000	7.3	120	Neither
30	dichloromethane (methylene chloride)	CH_2Cl_2		B1				Neither
31	chlorofluoromethane	CH_2ClF						Neither
32	difluoromethane (methylene fluoride)	CH_2F_2	1000	A2L	36,000	4.8	77	Neither
40	chloromethane (methyl chloride)	CH_3Cl		B2				Toxic
41	fluoromethane (methyl fluoride)	CH_3F						Neither
50	methane	CH_4	1000	A3				Neither
Ethane Series								
113	1,1,2-trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2	1000	A1	2600	1.2	20	Neither
114	1,2-dichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$	1000	A1	20,000	8.7	140	Neither
115[g]	chloropentafluoroethane	$CClF_2CF_3$	1000	A1	120,000	47	760	Neither
116[e]	hexafluoroethane	CF_3CF_3	1000	A1	97,000	34	550	Neither
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	50	B1	9100	3.5	57	Neither
124	2-chloro-1,1,1,2-tetrafluoroethane	$CHClFCF_3$	1000	A1	10,000	3.5	56	Neither
125[e]	pentafluoroethane	CHF_2CF_3	1000	A1	75,000	23	370	Neither
134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3	1000	A1	50,000	13	210	Neither
141b	1,1-dichloro-1-fluoroethane	CH_3CCl_2F	500		2600	0.78	12	Neither
142b	1-chloro-1,1-difluoroethane	CH_3CClF_2	1000	A2	20,000	5.1	83	Neither
143a	1,1,1-trifluoroethane	CH_3CF_3	1000	A2L	21,000	4.5	70	Neither
152a	1,1-difluoroethane	CH_3CHF_2	1000	A2	12,000	2.0	32	Neither
170	ethane	CH_3CH_3	1000	A3	7000	0.54	8.7	Neither
Ethers								
E170	methoxymethane (dimethyl ether)	CH_3OCH_3	1000	A3	8500	1.0	16	Neither
Propane								
218[e]	octafluoropropane	$CF_3CF_2CF_3$	1000	A1	90,000	43	690	Neither
227ea[e]	1,1,1,2,3,3,3-heptafluoropropane	CF_3CHFCF_3	1000	A1	84,000	36	580	Neither
236fa	1,1,1,3,3,3-hexafluoropropane	$CF_3CH_2CF_3$	1000	A1	55,000	21	340	Neither
245fa	1,1,1,3,3-pentafluoropropane	$CHF_2CH_2CF_3$	300	B1	34,000	12	190	Neither
290	propane	$CH_3CH_2CH_3$	1000	A3	5300	0.56	9.5	Neither
Cyclic Organic Compounds								
C318	octafluorocyclobutane	$-(CF_2)_4-$	1000	A1	80,000	41	660	Neither

Table 9.1 Refrigerant Data and Safety Classifications [Std 34-2016, Tbl 4-1] ***(Continued)***

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	OEL[f], ppm v/v	Safety Group	RCL[c] (ppm v/v)	(lb/Mcf)	(g/m^3)	Highly Toxic or Toxic[d] Under Code Classification
Miscellaneous Organic Compounds								
Hydrocarbons								
600	butane	$CH_3CH_2CH_2CH_3$	1000	A3	1000	0.15	2.4	Neither
600a	2-methylpropane (isobutane)	$CH(CH_3)_2CH_3$	1000	A3	4000	0.59	9.6	Neither
601	pentane	$CH_3CH_2CH_2CH_2CH_3$	600	A3	1000	0.18	2.9	Neither
601a	2-methylbutane (isopentane)	$(CH_3)_2CHCH_2CH_3$	600	A3	1000	0.18	2.9	Neither
Oxygen compounds								
610	ethoxyethane (ethyl ether)	$CH_3CH_2OCH_2CH_3$	400					Neither
611	methyl formate	$HCOOCH_3$	100	B2				Neither
Sulfur compounds								
620	(Reserved for future assignment)							
Nitrogen Compounds								
630	methanamine (methyl amine)	CH_3NH_2						Toxic
631	ethanamine (ethyl amine)	$CH_3CH_2(NH_2)$						Neither
Inorganic Compounds								
702	hydrogen	H_2		A3				Neither
704	helium	He		A1				Neither
717	ammonia	NH_3	25	B2L	320	0.014	0.22	Neither
718	water	H_2O		A1				Neither
720	neon	Ne		A1				Neither
728	nitrogen	N_2		A1				Neither
732	oxygen	O_2						Neither
740	argon	Ar		A1				Neither
744	carbon dioxide	CO_2	5000	A1	30,000	3.4	54	Neither
744A	nitrous oxide	N_2O						Neither
764	sulfur dioxide	SO_2		B1				Neither
Unsaturated Organic Compounds								
1130(E)	trans-1,2-dichloroethene	CHCl=CHCl	200	B1	100	0.25	4	Neither
1150	ethene (ethylene)	$CH_2=CH_2$	200	A3				Neither
1233zd(E)	trans-1-chloro-3,3,3-trifluoro-1-propene	$CF_3CH=CHCl$	800	A1	16,000	5.3	85	Neither
1234yf	2,3,3,3-tetrafluoro-1-propene	$CF_3CF=CH_2$	500	A2L	16,000	4.7	75	Neither
1234ze(E)	trans-1,3,3,3-tetrafluoro-1-propene	$CF_3CH=CFH$	800	A2L	16,000	4.7	75	Neither
1270	propene (propylene)	$CH_3CH=CH_2$	500	A3	1000	0.11	1.7	Neither
1336mzz(Z)	cis-1,1,1,4,4,4-hexaflouro-2-butene	$CF_3CHCHCF_3$	500	A1	13,000	5.4	87	Neither

a. The chemical name and chemical formula are not part of this standard. Chemical names conform to IUPAC nomenclature[14,15] except where shortened, unambiguous names are used following ASHRAE Standard 34 convention.
b. The preferred chemical name is followed by the popular name in parentheses.
c. Data taken from J.M. Calm, "ARTI Refrigerant Database," Air- Conditioning and Refrigeration Technology Institute (ARTI), Arlington, VA, July 2001; J.M. Calm, "Toxicity Data to Determine Refrigerant Concentration Limits," Report DE/CE 23810-110, Air- Conditioning and Refrigeration Technology Institute (ARTI), Arlington, VA, September 2000; J.M. Calm, "The Toxicity of Refrigerants," *Proceedings of the 1996 International Refrigeration Conference*, Purdue University, West Lafayette, IN, pp. 157–62, 1996; D.P. Wilson and R.G. Richard, "Determination of Refrigerant Lower Flammability Limits (LFLs) in Compliance with Proposed Addendum p to ANSI/ASHRAE Standard 34-1992 (1073-RP)," *ASHRAE Transactions* 2002, 108(2); D.W. Coombs, "HFC-32 Assessment of Anesthetic Potency in Mice by Inhalation," Huntingdon Life Sciences Ltd., Huntingdon, Cambridgeshire, England, February 2004 and amendment February 2006; D.W. Coombs, "HFC-22 An Inhalation Study to Investigate the Cardiac Sensitization Potential in the Beagle Dog," Huntingdon Life Sciences Ltd., Huntingdon, Cambridgeshire, England, August 2005; and other toxicity studies.
d. *Highly toxic*, *toxic*, or *neither*, where *highly toxic* and *toxic* are as defined in the *International Fire Code*, *Uniform Fire Code*, and OSHA regulations, and *neither* identifies those refrigerants having lesser toxicity than either of those groups[1,2,3].
e. At locations with altitudes higher than 4920 ft (1500 m), the ODL and RCL shall be 69,100 ppm.
f. The OELs are eight-hour TWAs, as defined in Section 3, unless otherwise noted; a "C" designation denotes a ceiling limit.
g. At locations with altitudes higher than 3300 ft (1000 m) but below or equal to 4920 ft (1500 m), the ODL and RCL shall be 112,000 ppm, and at altitudes higher than 4920 ft (1500 m), the ODL and RCL shall be 69,100 ppm.

Table 9.2 Data and Safety Classifications for Refrigerant Blends

[Std 34-2016, Tbl 4-2]

Refrigerant Number	Composition (Mass%)	Composition Tolerances	OEL[h], ppm v/v	Safety Group	RCL[a] (ppm v/v)	RCL[a] (lb/ Mcf)	RCL[a] (g/m^3)	Highly Toxic or Toxic[f] Under Code Classification
Zeotropes								
400	R-12/114 (must be specified)			A1				Neither
	(50.0/50.0)		1000	A1	28,000	10	160	
	(60.0/40.0)		1000	A1	30,000	11	170	
401A	R-22/152a/124 (53.0/13.0/34.0)	(±2.0/+0.5, –1.5/±1.0)	1000	A1	27,000	6.6	110	Neither
401B	R-22/152a/124 (61.0/11.0/28.0)	(±2.0/+0.5, –1.5/±1.0)	1000	A1	30,000	7.2	120	Neither
401C	R-22/152a/124 (33.0/15.0/52.0)	(±2.0/+0.5, –1.5/±1.0)	1000	A1	20,000	5.2	84	Neither
402A	R-125/290/22 (60.0/2.0/38.0)	(±2.0/+0.1, –1.0/±2.0)	1000	A1	66,000	17	270	Neither
402B	R-125/290/22 (38.0/2.0/60.0)	(±2.0/+0.1, –1.0/±2.0)	1000	A1	63,000	15	240	Neither
403A	R-290/22/218 (5.0/75.0/20.0)	(+0.2, –2.0/±2.0/±2.0)	1000	A2	33,000	7.6	120	Neither
403B[g]	R-290/22/218 (5.0/56.0/39.0)	(+0.2, –2.0/±2.0/±2.0)	1000	A1	70,000	18	290	Neither
404A[i]	R-125/143a/134a (44.0/52.0/4.0)	(±2.0/±1.0/±2.0)	1000	A1	130,000	31	500	Neither
405A	R-22/152a/142b/C318 (45.0/7.0/5.5/42.5)	Individual components = (±2.0/±1.0/±1.0/±2.0); sum of R-152a and R-142b = (+0.0, –2.0)	1000		57,000	16	260	Neither
406A	R-22/600a/142b (55.0/4.0/41.0)	(±2.0/±1.0/±1.0)	1000	A2	21,000	4.7	25	Neither
407A[g]	R-32/125/134a (20.0/40.0/40.0)	(±2.0/±2.0/±2.0)	1000	A1	83,000	19	300	Neither
407B[g]	R-32/125/134a (10.0/70.0/20.0)	(±2.0/±2.0/±2.0)	1000	A1	79,000	21	330	Neither
407C[g]	R-32/125/134a (23.0/25.0/52.0)	(±2.0/±2.0/±2.0)	1000	A1	81,000	18	290	Neither
407D	R-32/125/134a (15.0/15.0/70.0)	(±2.0/±2.0/±2.0)	1000	A1	68,000	16	250	Neither
407E[g]	R-32/125/134a (25.0/15.0/60.0)	(±2.0/±2.0/±2.0)	1000	A1	80,000	17	280	Neither
407F	R-32/125/134a (30.0/30.0/40.0)	(±2.0/±2.0/±2.0)	1000	A1	95,000	20	320	Neither
407G	R-32/125/134a (2.5/2.5/95.0)	(±0.5/±0.5/±1.0)	1000	A1	52,000	13	210	Neither
408A[g]	R-125/143a/22 (7.0/46.0/47.0)	(±2.0/±1.0/±2.0)	1000	A1	95,000	21	340	Neither
409A	R-22/124/142b (60.0/25.0/15.0)	(±2.0/±2.0/±1.0)	1000	A1	29,000	7.1	110	Neither
409B	R-22/124/142b (65.0/25.0/10.0)	(±2.0/±2.0/±1.0)	1000	A1	30,000	7.3	120	Neither
410A[i]	R-32/125 (50.0/50.0)	(+0.5, –1.5/+1.5, –0.5)	1000	A1	140,000	26	420	Neither
410B[i]	R-32/125 (45.0/55.0)	(±1.0/±1.0)		A1	140,000	27	430	Neither
411A[e]	R-1270/22/152a (1.5/87.5/11.0)	(+0.0, –1.0/+2.0, –0.0/+0.0, –1.0)	990	A2	14,000	2.9	46	Neither
411B[e]	R-1270/22/152a (3.0/94.0/3.0)	(+0.0, –1.0/+2.0, –0.0/+0.0, –1.0)	980	A2	13,000	2.8	45	Neither
412A	R-22/218/142b (70.0/5.0/25.0)	(±2.0/±2.0/±1.0)	1000	A2	22,000	5.1	82	Neither
413A	R-218/134a/600a (9.0/88.0/3.0)	(±1.0/±2.0/+0.0, –1.0)	1000	A2	22,000	5.8	94	Neither
414A	R-22/124/600a/142b (51.0/28.5/4.0/16.5)	(±2.0/±2.0/±0.5/+0.5, –1.0)	1000	A1	26,000	6.4	100	Neither
414B	R-22/124/600a/142b (50.0/39.0/1.5/9.5)	(±2.0/±2.0/±0.5/+0.5, –1.0)	1000	A1	23,000	6.0	95	Neither
415A	R-22/152a (82.0/18.0)	(±1.0/±1.0)	1000	A2	14,000	2.9	47	Neither
415B	R-22/152a (25.0/75.0)	(±1.0/±1.0)	1000	A2	12,000	2.1	34	Neither
416A[e]	R-134a/124/600 (59.0/39.5/1.5)	(+0.5, –1.0/+1.0, –0.5/+1.0, –0.2)	1000	A1	14,000	3.9	62	Neither
417A[e]	R-125/134a/600 (46.6/50.0/3.4)	(±1.1/±1.0/+0.1, –0.4)	1000	A1	13,000	3.5	56	Neither

Refrigerant Number	Composition (Mass%)	Composition Tolerances	OEL[h], ppm v/v	Safety Group	RCL[a] (ppm v/v)	(lb/Mcf)	(g/m³)	Highly Toxic or Toxic[f] Under Code Classification
Zeotropes (*continued*)								
417B	R-125/134a/600 (79.0/18.3/2.7)	(±1.0/±1.0/+0.1, –0.5)	1000	A1	15,000	4.3	70	Neither
417C	R-125/134a/600 (19.5/78.8/1.7)	(±1.0/±1.0/+0.1, –0.5)	1000	A1	21,000	5.4	87	Neither
418A	R-290/22/152a (1.5/96.0/2.5)	(±0.5/±1.0/±0.5)	1000	A2	22,000	4.8	77	Neither
419A[g]	R-125/134a/E170 (77.0/19.0/4.0)	(±1.0/±1.0/±1.0)	1000	A2	15,000	4.2	67	Neither
419B	R-125/134a/E170 (48.5/48.0/3.5)	(±1.0/±1.0/±0.5)	1000	A2	17,000	4.6	74	Neither
420A	R-134a/142b (88.0/12.0)	(+1.0, –0.0/+0.0, –1.0)	1000	A1	45,000	12	190	Neither
421A	R-125/134a (58.0/42.0)	(±1.0/±1.0)	1000	A1	61,000	17	280	Neither
421B	R-125/134a (85.0/15.0)	(±1.0/±1.0)	1000	A1	69,000	21	330	Neither
422A	R-125/134a/600a (85.1/11.5/3.4)	(±1.0/±1.0/+0.1, –0.4)	1000	A1	63,000	18	290	Neither
422B	R-125/134a/600a (55.0/42.0/3.0)	(±1.0/±1.0/+0.1, –0.5)	1000	A1	56,000	16	250	Neither
422C	R-125/134a/600a (82.0/15.0/3.0)	(±1.0/±1.0/+0.1, –0.5)	1000	A1	62,000	18	290	Neither
422D	R-125/134a/600a (65.1/31.5/3.4)	+0.9, –1.1/±1.0/+0.1, –0.4)	1000	A1	58,000	16	260	Neither
422E	R-125/134a/600a (58.0/39.3/2.7)	(±1.0/+1.7, –1.3/+0.3, –0.2)	1000	A1	57,000	16	260	Neither
423A	R-134a/227ea (52.5/47.5)	(±1.0/±1.0)	1000	A1	59,000	19	310	Neither
424A[e]	R-125/134a/600a/600/601a (50.5/47.0/0.9/1.0/0.6)	(±1.0/±1.0/+0.1, –0.2/+0.1, +0.2/+0.1, –0.2)	970	A1	23,000	6.2	100	Neither
425A	R-32/134a/227ea (18.5/69.5/12.0)	(±0.5/±0.5/±0.5)	1000	A1	72,000	16	260	Neither
426A[e]	R-125/134a/600/601a (5.1/93.0/1.3/0.6)	(±1.0/±1.0/+0.1, –0.2/+0.1, –0.2)	990	A1	20,000	5.2	83	Neither
427A	R-32/125/143a/134a (15.0/25.0/10.0/50.0)	(±2.0/±2.0/±2.0/±2.0)	1000	A1	79,000	18	290	Neither
428A	R-125/143a/290/600a (77.5/20.0/0.6/1.9)	(±1.0/±1.0/+0.1, –0.2/+0.1, –0.2)	1000	A1	83,000	23	370	Neither
429A	R-E170/152a/600a (60.0/10.0/30.0)	(±1.0/±1.0/±1.0)	1000	A3	6300	0.81	13	Neither
430A	R-152a/600a (76.0/24.0)	(±1.0/±1.0)	1000	A3	8000	1.3	21	Neither
431A	R-290/152a (71.0/29.0)	(±1.0/±1.0)	1000	A3	5500	0.69	11	Neither
432A	R-1270/E170 (80.0/20.0)	(±1.0/±1.0)	700	A3	1200	0.13	2.1	Neither
433A	R-1270/290 (30.0/70.0)	(±1.0/±1.0)	880	A3	3100	0.34	5.5	Neither
433B	R-1270/290 (5.0/95.0)	(±1.0/±1.0)	950	A3	4500	0.51	8.1	Neither
433C	R-1270/290 (25.0/75.0)	(±1.0/±1.0)	790	A3	3600	0.41	6.6	Neither
434A[g]	R-125/143a/134a/600 (63.2/18.0/16.0/2.8)	(±1.0/±1.0/±1.0/+0.1, –0.2)	1000	A1	73,000	20	320	Neither
435A	R-E170/152a (80.0/20.0)	(±1.0/±1.0)	1000	A3	8500	1.1	17	Neither
436A	R-290/600a (56.0/44.0)	(±1.0/±1.0)	1000	A3	4000	0.50	8.1	Neither
436B	R-290/600a (52.0/48.0)	(±1.0/±1.0)	1000	A3	4000	0.51	8.2	Neither
437A	R-125/134a/600/601 (19.5/78.5/1.4/0.6)	(+0.5, –1.8/+1.5, –0.7/+0.1, –0.2/+0.1, –0.2)	990	A1	19,000	5.0	82	Neither
438A	R-32/125/134a/600/601a (8.5/45.0/44.2/1.7/0.6)	(+0.5, –1.5/±1.5/±1.5/+0.1, –0.2/+0.1, –0.2)	990	A1	20,000	4.9	79	Neither
439A	R-32/125/600a (50.0/47.0/3.0)	(±1.0/±1.0/±0.5)	990	A2	26,000	4.7	76	Neither

Table 9.2 Data and Safety Classifications for Refrigerant Blends
[Std 34-2016, Tbl 4-2] ***(Continued)***

Refrigerant Number	Composition (Mass%)	Composition Tolerances	OEL[h], ppm v/v	Safety Group	RCL[a] (ppm v/v)	RCL[a] (lb/Mcf)	RCL[a] (g/m^3)	Highly Toxic or Toxic[f] Under Code Classification
Zeotropes (*continued*)								
440A	R-290/134a/152a (0.6/1.6/97.8)	(±0.1/±0.6/±0.5)	1000	A2	12,000	1.9	31	Neither
441a	R-170/290/600a/600 (3.1/54.8/6.0/36.1)	(±0.3/±2.0/±0.6/±2.0)	1000	A3	3200	0.39	6.3	Neither
442A	R-32/125/134a/152a/227ea (31.0/31.0/30.0/3.0/5.0)	(±1.0/±1.0±/1.0/±0.5/±1.0)	1000	A1	100,000	21	330	Neither
443A	R-1270/290/600a (55.0/40.0/5.0)	(±2.0/±2.0/±1.2)	580	A3	1700	0.19	3.1	Neither
444A	R-32/152a/1234ze(E) (12.0/5.0/83.0)	(±1.0/±1.0/±2.0)	850	A2L	21,000	5.1	81	Neither
444B	R-32/152a/1234ze(E) (41.5/10.0/48.5)	(±1.0/±1.0/±1.0)	890	A2L	23,000	4.3	69	Neither
445A	R-744/134a/1234ze(E) (6.0/9.0/85.0)	(±1.0/±1.0/±2.0)	930	A2L	16,000	4.2	67	Neither
446A	R-32/1234ze(E)/600 (68.0/29.0/3.0)	(+0.5, –1.0/+2.0, –0.6/+0.1, –1.0)	960	A2L	16,000	2.5	39	Neither
447A	R-32/125/1234ze(E) (68.0/3.5/28.5)	(+1.5, –0.5/+1.5, –0.5/+1.0, –1.0)	900	A2L	16,000	2.6	42	Neither
447B	R-32/125/1234ze(E) (68.0/8.0/24.0)	(+1.0, –2.0/+2.0, –1.0/+1.0, –2.0)	970	A2L	30,000	23	360	Neither
448A	R-32/125/1234yf/134a/1234ze(E) (26.0/26.0/20.0/21.0/7.0)	(+0.5, –2.0/+2.0, –0.5/+0.5, –2.0/+2.0, –1.0/+0.5, –2.0)	890	A1	110,000	24	390	Neither
449A	R-32 /125 /1234yf /134a (24.3/24.7/25.3/25.7)	(+0.2, –1.0/+1.0, –0.2/+0.2, –1.0/+1.0, –0.2)	830	A1	100,000	23	370	Neither
449B	R-32/125/1234yf/134a (25.2/24.3/23.2/27.3)	(+0.3, –1.5/+1.5, –0.3/+0.3, –1.5/+1.5, –0.3)	850	A1	100,000	23	370	Neither
449C	R-32/125/1234yf/134a (20.0/20.0/31.0/29.0)	(+0.5, –1.5/+1.5, –0.5/+0.5, –1.5/+1.5, –0.5)	800	A1	98,000	23	360	Neither
450A	R-134a/1234ze(E) (42.0/58.0)	(±2.0/±2.0)	880	A1	72,000	20	320	Neither
451A	R-1234yf/134a (89.8/10.2)	(±0.2/±0.2)	520	A2L	18,000	5.3	81	Neither
451B	R-1234yf/134a (88.8/11.2)	(±0.2/±0.2)	530	A2L	18,000	5.3	81	Neither
452A	R-32/125/1234yf (11.0/59.0/30.0)	(±1.7/±1.8/+0.1, –1.0)	780	A1	10,000	27	440	Neither
452B	R-32/125/1234yf (67.0/7.0/26.0)	(±2.0/±1.5/±2.0)	870	A2L	30,000	23	360	Neither
452C	R-32/125/1234yf (12.5/61.0/26.5)	(+0.5, –1.5/±1.0/+0.5, –1.5)	800	A1	100,000	27	430	Neither
453A	R-32/125/134a/227ea/600/601a (20.0/20.0/53.8/5.0/0.6/0.6)	(±1.0/±1.0/±1.0/±0.5/+0.1, –0.2/+0.1, –0.2)	1000	A1	34,000	7.8	120	Neither
454A	R-32/1234yf (35.0/65.0)	(+2.0/–2.0, +2.0/–2.0)	690	A2L	16,000	28	450	Neither
454B	R-32/1234yf (68.9/31.1)	(+1.0/–1.0, +1.0/–1.0)	850	A2L	19,000	22	360	Neither
454C	R-32/1234yf (21.5/78.5)	(±2.0/±2.0)	620	A2L	19,000	29	460	Neither
455A	R-744/32/1234yf (3.0/21.5/75.5)	(+2.0, –1.0/+1.0, –2.0/±2.0)	650	A2L	30,000	23	380	Neither
456A	R-32/134a/1234ze(E) (6.0/45.0/49.0)	(±1.0/±1.0/±1.0)	900	A1	77,000	20	320	Neither
457A	R-32/1234yf/152a (18.0/70.0/12.0)	(+0.5, –1.5/+0.5, –1.5/+0.1, –1.9)	650	A2L	15,000	25	400	Neither
458A	R-32/125/134a/227ea/236fa (20.5/4.0/61.4/13.5/0.6)	(±0.5/±0.5/±0.5/±0.5/±0.1)	1000	A1	76,000	18	280	Neither

Table 9.2 Data and Safety Classifications for Refrigerant Blends
[Std 34-2016, Tbl 4-2] ***(Continued)***

Refrigerant Number	Composition (Mass%)	Composition Tolerances	OEL[h], ppm v/v	Safety Group	RCL[a] (ppm v/v)	RCL[a] (lb/Mcf)	RCL[a] (g/m^3)	Highly Toxic or Toxic[f] Under Code Classification
Azeotropes[b]								
500	R-12/152a (73.8/26.2)		1000	A1	30,000	7.6	120	Neither
501	R-22/12 (75.0/25.0)[c]		1000	A1	54,000	13	210	Neither
502[g]	R-22/115 (48.8/51.2)		1000	A1	73,000	21	330	Neither
503	R-23/13 (40.1/59.9)		1000					Neither
504[i]	R-32/115 (48.2/51.8)		1000		140,000	28	450	Neither
505	R-12/31 (78.0/22.0)[c]							Neither
506	R-31/114 (55.1/44.9)							Neither
507A[d,i]	R-125/143a (50.0/50.0)		1000	A1	130,000	32	520	Neither
508A[d]	R-23/116 (39.0/61.0)		1000	A1	55,000	14	220	Neither
508B	R-23/116 (46.0/54.0)		1000	A1	52,000	13	200	Neither
509A[d,g]	R-22/218 (44.0/56.0)		1000	A1	75,000	24	390	Neither
510A	R-E170/600a (88.0/12.0)	(±0.5/±0.5)	1000	A3	7300	0.87	14	Neither
511A	R-290/E170 (95.0/5.0)	(±1.0/±1.0)	1000	A3	5300	0.59	9.5	Neither
512A	R-134a/152a (5.0/95.0)	(±1.0/±1.0)	1000	A2	11,000	1.9	31	Neither
513A	R-1234yf/134a (56.0/44.0)	(±1.0/±1.0)	650	A1	72,000	20	320	Neither
513B	R-1234yf/134a (58.5/41.5)	(±0.5/±0.5)	640	A1	74,000	21	330	Neither
514A	R-1336mzz(Z)/1130 (E) (74.7/25.3)	(+1.5, –0.5/+0.5, –1.5)	320	B1	2400	0.86	14	Neither
515A	R-1234ze(E)/227ea (88.0/12.0)	(+1.0, –2.0/+2.0, –1.0)	810	A1	62,000	19	300	Neither

a. Data taken from J.M. Calm, “ARTI Refrigerant Database,” Air- Conditioning and Refrigeration Technology Institute (ARTI), Arlington, VA, July 2001; J.M. Calm, “Toxicity Data to Determine Refrigerant Concentration Limits,” Report DE/CE 23810-110, Air- Conditioning and Refrigeration Technology Institute (ARTI), Arlington, VA, September 2000; J.M. Calm, “The Toxicity of Refrigerants,” *Proceedings of the 1996 International Refrigeration Conference*, Purdue University, West Lafayette, IN, pp. 157–62, 1996; D.P. Wilson and R.G. Richard, “Determination of Refrigerant Lower Flammability Limits (LFLs) in Compliance with Proposed Addendum p to ANSI/ASHRAE Standard 34-1992 (1073-RP),” *ASHRAE Transactions* 2002, 108(2); D.W. Coombs, “HFC-32 Assessment of Anesthetic Potency in Mice by Inhalation,” Huntingdon Life Sciences Ltd., Huntingdon, Cambridgeshire, England, February 2004 and amendment February 2006; D.W. Coombs, “HFC-22 An Inhalation Study to Investigate the Cardiac Sensitization Potential in the Beagle Dog,” Huntingdon Life Sciences Ltd., Huntingdon, Cambridgeshire, England, August 2005; and other toxicity studies.
b. Azeotropic refrigerants exhibit some segregation of components at conditions of temperature and pressure other than those at which they were formulated. The extent of segregation depends on the particular azeotrope and hardware system configuration.
c. The exact composition of this azeotrope is in question, and additional experimental studies are needed.
d. R-507, R-508, and R-509 are allowed alternative designations for R-507A, R-508A, and R-509A due to a change in designations after assignment of R-500 through R-509. Corresponding changes were not made for R-500 through R-506.
e. The RCL values for these refrigerant blends are approximated in the absence of adequate data for a component comprising less than 4% m/m of the blend and expected to have only a small influence in an acute, accidental release.
f. *Highly toxic*, *toxic*, or *neither*, where *highly toxic* and *toxic* are as defined in the *International Fire Code*, *Uniform Fire Code*, and OSHA regulations, and *neither* identifies those refrigerants having lesser toxicity than either of those groups [1,2,3].
g. At locations with altitudes higher than 4920 ft (1500 m), the ODL and RCL shall be 69,100 ppm.
h. The OELs are eight-hour TWAs as defined in Section 3 unless otherwise noted; a “C” designation denotes a ceiling limit.
i. At locations with altitudes higher than 3300 ft (1000 m) but below or equal to 4920 ft (1500 m), the ODL and RCL shall be 112, 000 ppm, and at altitudes higher than 4920 ft (1500 m), the ODL and RCL shall be 69,100 ppm.

Refrigerating Machinery Rooms

When required, the machinery room shall

- Be dimensioned so parts are accessible with space for service and maintenance.
- Have tight-fitting doors opening outward, self-closing if they open into the building, with no openings permitting passage of escaping refrigerant into the building except gasketed access panels of ductwork and air-handling equipment.
- Contain a leak detector located where refrigerant from a leak will concentrate that actuates visual and audible alarms inside the room and outside each entrance, and activates the mechanical ventilation.
- On alarm, exhaust of $Q_{cfm} = 100 \times G^{0.5}$ where G is pounds of refrigerant in the largest system, with openings for inlet air to avoid recirculation; multiple fans or multispeed fans to operate to reduce airflow for normal operation to at least 0.5 cfm per ft^2 or 20 cfm per person, and operable when occupied to limit temperature rise to 18°F above inlet air or a maximum of 122°F.
- Combustion air can be used for equipment in the machinery room only if ducted from outside the room and sealed from refrigerant entry, and a refrigerant detector is employed to shut off combustion on refrigerant leaks. (Exceptions: CO_2 or water refrigerant; or ammonia only driven by internal combustion engine.)
- Be no airflow to or from an occupied space through a machinery room unless ducted and sealed against refrigerant leaks.
- Restrict access to authorized personnel with clear signage at each entrance. If system is in an open enclosure outdoors more than 20 ft from building openings, mechanical or natural ventilation may be used; free-opening area for natural ventilation shall be $F_{sqft} = G^{0.5}$.

The total of Group A2, B2, A3, or B3 refrigerants except R-717 (ammonia) shall not exceed 1100 lb without approval of the authority having jurisdiction. Special requirements in 7.5 of Standard 15-2007 apply relative to recovered, reclaimed, or recycled refrigerants, or mixing of refrigerants, refrigerant, or lubricant conversion. Group A2, A3, B1, B2, or B3 refrigerants shall not be used in high-probability systems (where a refrigerant leak can enter occupied space) for human comfort.

Refrigerant Piping

Piping shall not be installed in elevators or other shafts that have moving objects or open into living quarters. It shall not penetrate floors except the top floor to the roof, or the basement to the first floor, unless enclosed in a gastight, fire-resistive shaft. The piping shall be enclosed in a pipe duct if inside floors.

Pressure Relief Protection

Refrigerating systems shall be protected by pressure relief devices. ANSI/ASHRAE Standard 15 covers required location and sizing. All pressure relief valves shall be marked "UV" or "VR," and all rupture members marked with data required by paragraph UG 127 of Section VIII, Division 1, of the *ASME Boiler and Pressure Vessel Code*; and fusible plugs shall be marked with the melting temperature. Generally, pressure relief devices and fusible plugs shall discharge to the atmosphere not less than 15 ft above ground and 20 ft from a window, ventilation opening, pedestrian walkway, or exit in any building.

10. REFRIGERATION

Transmission Load

The overall coefficient of heat transfer U of the wall, floor, or ceiling of a refrigerated space can be derived from:

$$U = \frac{1}{1/f_i + x_1/k_1 + x_2/k_2 + 1/f_o} \tag{10.1}$$

where

U	=	overall heat transfer coefficient, Btu/h· ft^2· °F
x	=	wall thickness, in.
k	=	thermal conductivity of wall material, Btu· in/h· ft^2· °F
f_i	=	inside film or surface conductance, Btu/h· ft^2· °F
f_o	=	outside film or surface conductance, Btu/h· ft^2· °F

1.65 Btu/h· ft^2· °F for f_i and f_o is frequently used for still air. If the outer surface is exposed to 15 mph wind, f_o is increased to 6 Btu/h· ft^2· °F.

With thick walls and low conductivity, the resistance x/k makes U so small that l/f has little effect and can be omitted from the calculation.

After establishing U, the heat gain is given by Equation 10.2:

$$q = UA\Delta t \tag{10.2}$$

where

q	=	heat leakage, Btu/h
A	=	outside area of section, ft^2
Δt	=	difference between outside air temperature and air temperature of the refrigerated space, °F

Latent heat gain due to moisture transmission through walls, floors, and ceilings of modern-construction refrigerated facilities is negligible.

Table 10.1 Thermal Conductivity of Cold Storage Insulation [2014R, Ch 24, Tbl 1]

Insulation	Thermal Conductivity[a] k, Btu·in/h·ft^2·°F
Polyurethane board (R-11 expanded)	0.16 to 0.18
Polyisocyanurate, cellular (R-141b expanded)	0.19
Polystyrene, extruded (R-142b)	0.24
Polystyrene, expanded (R-142b)	0.26
Corkboard[b]	0.30
Foam glass[c]	0.31

[a]Values are for a mean temperature of 75°F, and insulation is aged 180 days.
[b]Seldom used. Data are only for reference.
[c]Virtually no effects from aging.

Table 10.2 Minimum Insulation Thickness [2014R, Ch 24, Tbl 2]

Storage Temperature, °F	Expanded Polyisocyanurate Thickness	
	Northern U.S., in.	Southern U.S., in.
50 to 60	2	2
40 to 50	2	2
25 to 40	2	3
15 to 25	3	3
0 to 15	3	4
–15 to 0	4	4
–40 to –15	5	5

Table 10.3 Allowance for Sun Effect [2014R, Ch 24, Tbl 3]

Typical Surface Types	East Wall, °F	South Wall, °F	West Wall, °F	Flat Roof, °F
Dark-colored surfaces				
Slate roofing	8	5	8	20
Tar roofing				
Black paint				
Medium-colored surfaces				
Unpainted wood	6	4	6	15
Brick				
Red tile				
Dark cement				
Red, gray, or green paint				
Light-colored surfaces				
White stone	4	2	4	9
Light-colored cement				
White paint				

Note: Add to the normal temperature difference for heat leakage calculations to compensate for sun effect. Do not use for air-conditioning design.

Product Load

1. Heat removed to cool from initial temperature to some lower temperature above freezing:

$$Q_1 = mc_1(t_1 - t_2) \tag{10.3}$$

2. Heat removed in cooling from the initial temperature to a freezing point of the product:

$$Q_2 = mc_1(t_1 - t_f) \tag{10.4}$$

3. Heat removed to freeze the product:

$$Q_3 = m h_{if} \tag{10.5}$$

4. Heat removed in cooling from the freezing point to the final temperature below the freezing point:

$$Q_4 = mc_2\,(t_f - t_3) \tag{10.6}$$

where

Q_1, Q_2, Q_3, Q_4	=	heat removed, Btu
m	=	weight of the product, lb
c_1	=	specific heat of the product above freezing, Btu/lb· °F
t_1	=	initial temperature of the product above freezing, °F
t_2	=	lower temperature of the product above freezing, °F
t_f	=	freezing temperature of the product, °F
h_{if}	=	latent heat of fusion of the product, Btu/lb
c_2	=	specific heat of the product below freezing, Btu/lb· °F
t_3	=	final temperature of the product below freezing, °F

Specific heats above and below freezing for many products are given in 2014R, Ch 19, Tbl 3.

Refrigeration system capacity for products brought into refrigerated spaces is determined from the time allotted for heat removal and assumes that the product is properly exposed to remove the heat in that time. The calculation is:

$$q = \frac{Q_2 + Q_3 + Q_4}{n} \tag{10.7}$$

where

q	=	product cooling load, Btu/h
n	=	allotted time period, h

A product's latent heat of fusion is related to its water content and can be estimated by multiplying the product's percent of water (expressed as a decimal) by the water's latent heat of fusion, 144 Btu/lb. Most food products freeze in the range of 26 to 31°F. When the exact freezing temperature is not known, assume that it is 28°F.

Table 10.4 Heat Gain from Typical Electric Motors [2014R, Ch 24, Tbl 6]

Motor Name-plate or Rated Horse-power	Motor Type	Nominal rpm	Full Load Motor Efficiency, %	Location of Motor and Driven Equipment with Respect to Conditioned Space or Airstream		
				A	B	C
				Motor in, Driven Equipment in, Btu/h	Motor out, Driven Equipment in, Btu/h	Motor in, Driven Equipment out, Btu/h
0.05	Shaded pole	1500	35	360	130	240
0.08				580	200	380
0.125				900	320	590
0.16				1160	400	760
0.25	Split phase	1750	54	1180	640	540
0.33			56	1500	840	660
0.50			60	2120	1270	850
0.75	3-Phase	1750	72	2650	1900	740

Table 10.5 Heat Equivalent of Occupancy [2014R, Ch 24, Tbl 7]

Refrigerated Space Temperature, °F	Heat Equivalent/Person, Btu/h
50	720
40	840
30	950
20	1050
10	1200
0	1300
–10	1400

Note: Heat equivalent may be estimated by the following equation:

$$q_p = 1295 - 11.5t$$

where t = temperature of refrigerated space, °F

Packaging Related Load

Cardboard and wood used as part of product packaging adsorb or desorb moisture, depending on air temperature and relative humidity. This moisture sorption represents a conversion between sensible and latent heat: the latent heat of sorption is countered by sensible heat transfer between the packaging and the air by convection. The heat load q_t from to the i packaging components is

$$q_t = \frac{\sum m_i c_i (t_1 - t_3)}{3600n} \tag{10.8}$$

$$q_l = \frac{\left[\frac{m(X_1 - X_3)L}{1 + X_1}\right]_{wood} + \left[\frac{m(X_1 - X_3)L}{1 + X_1}\right]_{cardboard}}{3600n} \tag{10.9}$$

where

q_l	=	total heat load, Btu
m_i	=	mass of ith packaging component, lb
c_i	=	specific heat of ith packaging component, Btu/lb·°F
X_1	=	entering packaging component moisture content, lb/lb dry basis
X_3	=	packaging component moisture content after time n, lb/lb dry basis
L	=	latent heat of sorption, Btu/lb (1075 Btu/lb)

Typical values of c_i are given in Table 10.6. X_1 can be measured or estimated using moisture sorption isotherms based on the air temperature and relative humidity the packaging experienced before entering the refrigerated space. The moisture sorption isotherms for wood and cardboard are given in Figures 10.1 and 10.2. X_3 can be estimated using Figures 10.3 or 10.4 to get Y and

$$X_3 = X^* + (X_1 - X^*)Y \tag{10.10}$$

where

Y	=	fractional unaccomplished moisture change (Figure 10.4)
X^*	=	equilibrium packaging component moisture content, lb/lb dry basis

X^* can be estimated using the moisture sorption isotherms based on the air temperature and relative humidity in the refrigerated space using Figures 10.1 and 10.2 for wood and cardboard, respectively, where equilibrium moisture content (EMC) is plotted against equilibrium relative humidity (ERH).

Table 10.6 Typical Specific Heat Capacities of Common Packaging Materials
[2014R, Ch 24, Tbl 8]

Material	Specific Heat Capacity c, Btu/lb·°F
Wood	0.41
Cardboard	0.33
Plastic	0.38
Aluminum	0.20
Steel	0.12

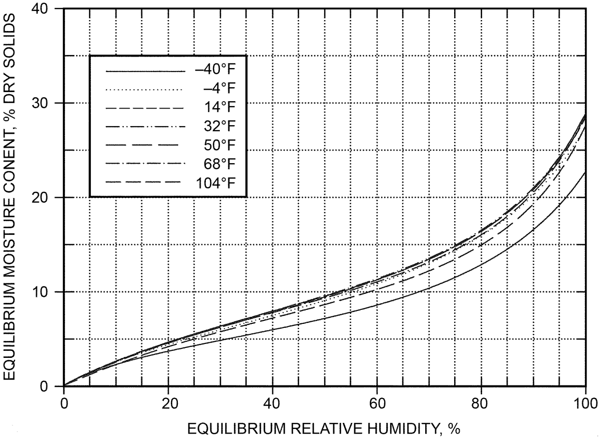


Figure 10.1 Moisture Sorption Isotherms for Wood as Function of Air Temperature and Relative Humidity [2014R, Ch 24, Fig 3]

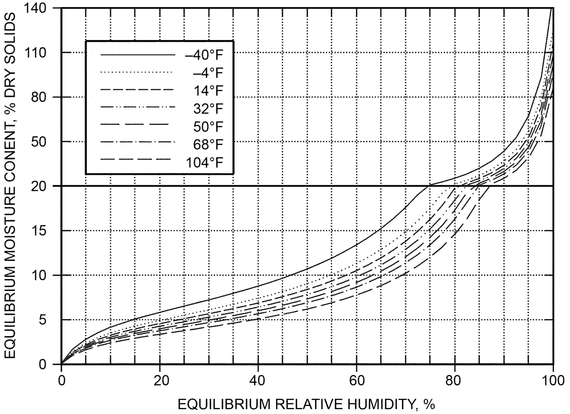
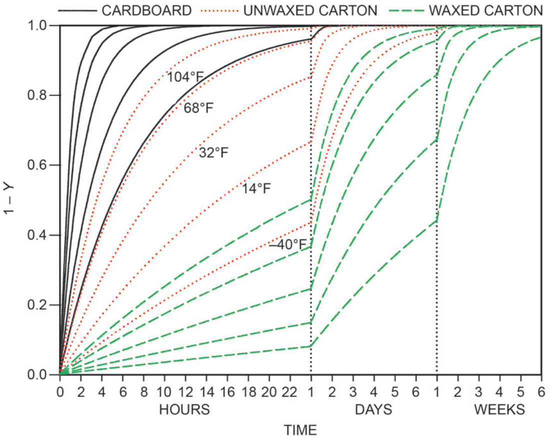
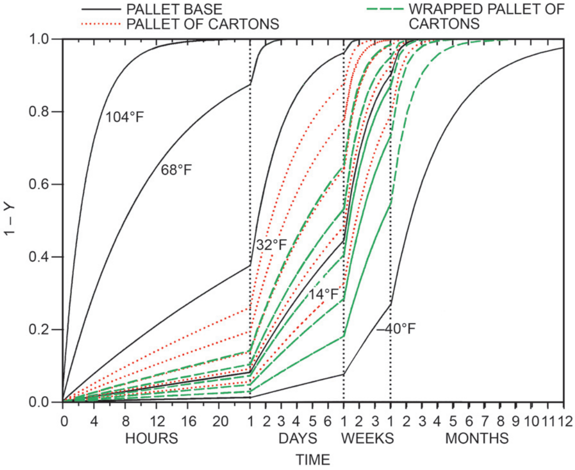


Figure 10.2 Moisture Sorption Isotherms for Cardboard as Function of Air Temperature and Relative Humidity [2014R, Ch 24, Fig 4]



(All plots given in the same temperature order as for unwaxed carton)

Figure 10.3 Fractional Unaccomplished Moisture Change as Function of Time and Temperature for Sheets of Cardboard, Unwaxed Cartons, and Waxed Cartons [2014R, Ch 24, Fig 5]



(All plots given in same temperature order as for pallet of cartons)

Figure 10.4 Fractional Unaccomplished Moisture Change as Function of Time and Temperature for Wooden Pallet Bases, Unwrapped Pallets of Cartons, and Wrapped Pallets of Cartons [2014R, Ch 24, Fig 6]

Refrigeration Load

Infiltration Air Load

Heat gain through doorways from air exchange is:

$$q_t = qD_tD_f(1 - E) \qquad (10.11)$$

where

q_t	=	average heat gain for the 24 h or other period, Btu/h
q	=	sensible and latent refrigeration load for fully established flow, Btu/h
D_t	=	doorway open-time factor
D_f	=	doorway flow factor
E	=	effectiveness of doorway protective device

$$q = 3790\ WH^{1.5}\ (Q_s/A)(1/R_s) \qquad (10.12)$$

where

Q_s/A	=	sensible heat load of infiltration air per square foot of doorway opening as read from Figure 10.3, ton/ft^2
W	=	doorway width, ft
H	=	doorway height, ft
R_s	=	sensible heat ratio of the infiltration air heat gain, from a psychrometric chart

Doorway open-time factor D_t can be calculated as follows:

$$D_t = \frac{(P\theta_p + 60\theta_o)}{3600\theta_d} \qquad (10.13)$$

where

D_t	=	decimal portion of time doorway is open
P	=	number of doorway passages
θ_p	=	door open-close time, seconds per passage
θ_o	=	time door simply stands open, min
θ_d	=	the daily (or other) time period, h

Equipment-Related Load

Equipment-related load consists essentially of fan heat where forced air circulation is used, reheat where humidity control is provided, defrosting heat gain where defrosting occurs, and moisture evaporation where the defrosting process is exposed to refrigerated air. To accurately select heat-extracting equipment, a distinction should be made between those equipment heat loads that are felt within the refrigerated space and those that are introduced directly to the refrigerating fluid.

Equipment heat gain is usually minor at space temperatures above approximately 30°F, but may be up to 15%.

Safety Factor

Generally, a 10% safety factor is applied to the calculated load to allow for possible discrepancies between the design criteria and actual operation. Refrigeration system capacity should be sufficient to handle the load with the actual running time, allowing for defrost cycles.

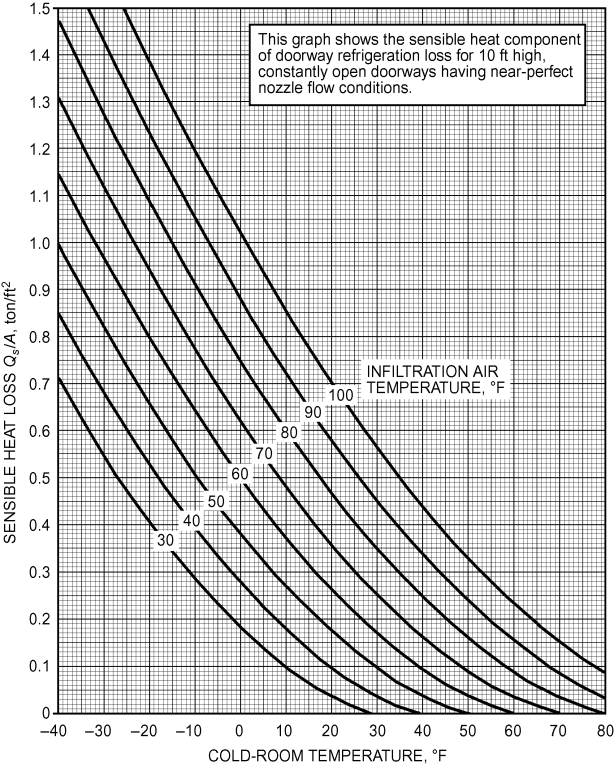


Figure 10.5 Sensible Heat Gain by Air Exchange for Continuously Open Door with Fully Established Flow [2014R, Ch 24, Fig 9]

Refrigeration Equipment

Liquid Coolers

A liquid cooler (hereafter called a cooler) is a heat exchanger A in which refrigerant is evaporated, thereby cooling a fluid (usually water or brine) circulating through the cooler.

Various types of liquid coolers and their characteristics are listed in Table 10.7 and Figures 10.6 through 10.12.

Heat transfer for liquid coolers can be expressed by the following steady-state heat transfer equation:

$$q = UA\Delta t_m \tag{10.14}$$

where

q = total heat transfer rate, Btu/h

Δt_m = mean temperature difference, °F

A = heat transfer surface area associated with U, ft^2

U = overall heat transfer coefficient, $Btu/h \cdot ft^2 \cdot °F$

The area A can be calculated if the geometry of the cooler is known. The mean temperature difference is

$$\Delta t_m = (\Delta t_1 - \Delta t_2)/\ln(\Delta t_1/\Delta t_2) \tag{10.15}$$

where Δt_1 and Δt_2 are temperature differences between the fluids at each end of the heat exchanger.

Table 10.7 Types of Coolers

Type of Cooler	Subtype	Usual Refrigerant Feed Device	Usual Capacity Range, tons	Commonly Used Refrigerants
Direct-expansion	Shell-and-tube	Thermal expansion valve Electronic modulation valve	2 to 500 2 to 500	12, 22, 134a, 404A, 407C, 410A, 500, 502, 507A, 717
	Tube-in-tube	Thermal expansion valve	5 to 25	12, 22, 134a, 717
	Brazed-plate	Thermal expansion valve	0.6 to 200	12, 22, 134a, 404A 407C, 410A, 500, 502, 507A, 508B, 717, 744
	Semiwelded plate	Thermal expansion valve	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Flooded	Shell-and-tube	Low-pressure float	25 to 2000	11, 12, 22, 113, 114
		High-pressure float	25 to 6000	123, 134a, 500, 502, 507A, 717
		Fixed orifice(s) Weir	25 to 6000 25 to 6000	
	Spray shell-and-tube	Low-pressure float	50 to 10,000	11, 12, 13B1, 22
		High-pressure float	50 to 10,000	113, 114, 123, 134a
	Brazed-plate	Low-pressure float	0.6 to 200	12, 22, 134a, 500, 502, 507A, 717, 744
	Semiwelded plate	Low-pressure float	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Baudelot	Flooded	Low-pressure float	10 to 100	22, 717
	Direct-expansion	Thermal expansion valve	5 to 25	12, 22, 134a, 717
Shell-and-coil	—	Thermal expansion valve	2 to 10	12, 22, 134a, 717

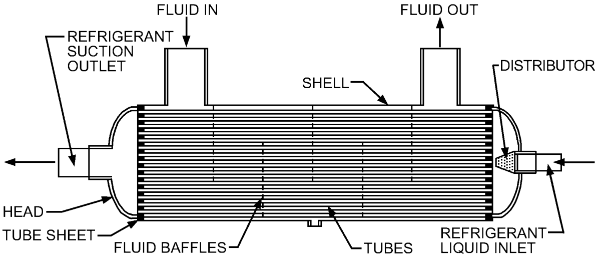


Figure 10.6 Direct-Expansion Shell-and-Tube Cooler [2016S, Ch 42, Fig 1]]

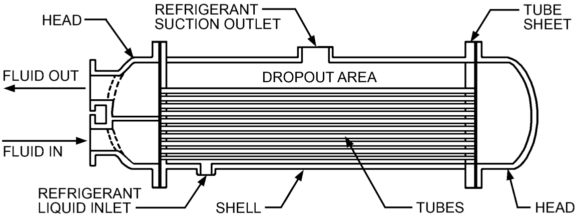


Figure 10.7 Flooded Shell-and-Tube Cooler [2016S, Ch 42, Fig 2]]

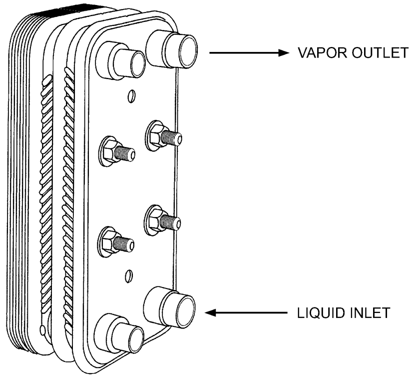


Figure 10.8 Flooded Plate Cooler [2016S, Ch 42, Fig 3]

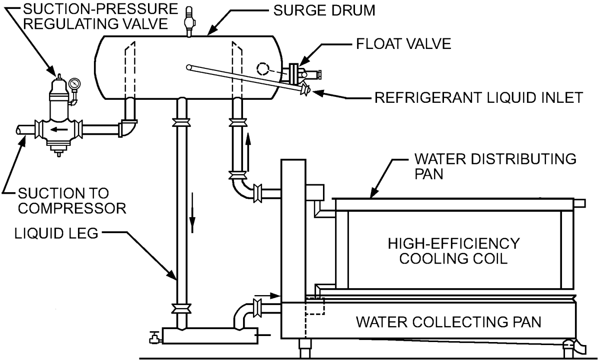


Figure 10.9 Baudelot Cooler [2016S, Ch 42, Fig 4]

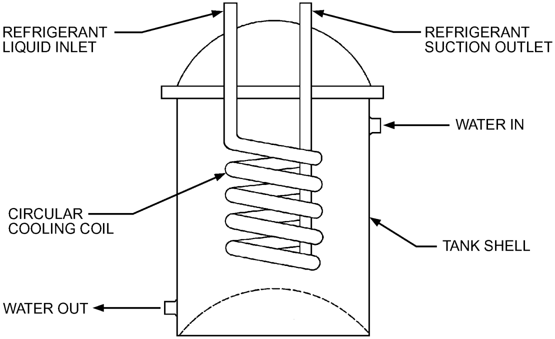


Figure 10.10 Shell-and-Coil Cooler [2016S, Ch 42, Fig 5]

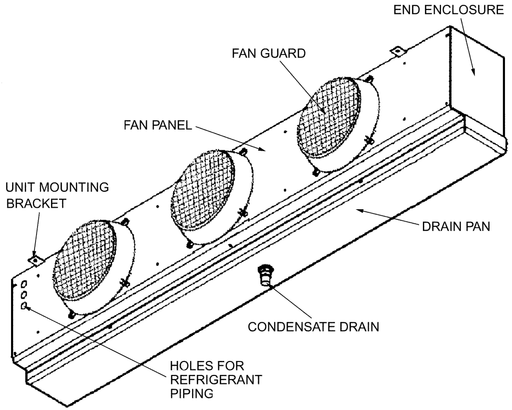


Figure 10.11 Low-Profile Cooler [2014R, Ch 14, Fig 3]

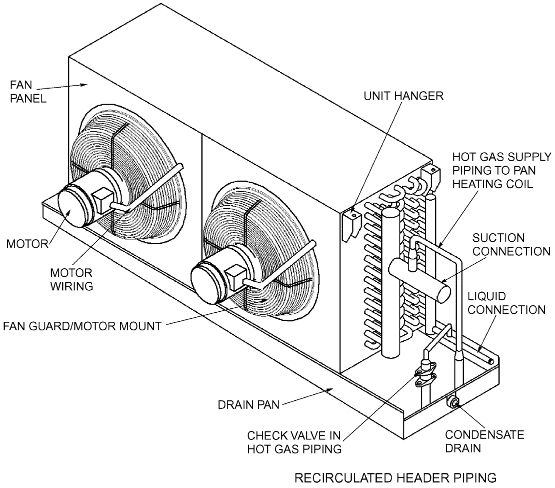


Figure 10.12 Liquid Overfeed Type Unit Cooler [2014R, Ch 14, Fig 4]

U may be calculated by one of the following equations.

Based on inside surface area:

$$U = \frac{1}{1/h_i + [A_i/(A_o h_o)] + (t/k)(A_i/A_m) + r_{fi}} \tag{10.16}$$

Based on outside surface area:

$$U = \frac{1}{[A_o/(A_i h_i)] + 1/h_o + (t/k)(A_o/A_m) + r_{fo}} \tag{10.17}$$

where

h_i = inside heat transfer coefficient based on inside surface area, Btu/h·ft^2·°F

h_o = outside heat transfer coefficient based on outside surface area, Btu/h·ft^2·°F

A_o = outside heat transfer surface area, ft^2

A_i = inside heat transfer surface area, ft^2

A_m = mean heat transfer area of metal wall, ft^2

k = thermal conductivity of heat transfer material, Btu/h·ft·°F

t = thickness of heat transfer surface (tube wall thickness), ft

r_{fi} = fouling factor of fluid side based on inside surface area, ft^2·h·°F/Btu

r_{fo} = fouling factor of fluid side based on outside surface area, ft^2·h·°F/Btu

Note: If fluid is on inside, multiply r_{fi} by A_o/A_i to find r_{fo}.
If fluid is on outside, multiply r_{fo} by A_i/A_o to find r_{fi}.

These equations can be applied to incremental sections of the heat exchanger to include local effects on the value of U, and then the increments summed to obtain a more accurate design.

Over time, most fluids foul the fluid-side heat transfer surface, reducing the cooler's overall heat transfer coefficient. If fouling is expected to be a problem, a mechanically cleanable cooler should be used, such as a flooded, Baudelot, or cleanable direct-expansion tube-in-tube cooler. Direct-expansion shell-and-tube, shell-and-coil, and brazed-plate coolers can be cleaned chemically. Flooded coolers and direct-expansion tube-in-tube coolers with enhanced fluid-side heat transfer surfaces tend to be self-cleaning because of high fluid turbulence, so a smaller fouling factor can probably be used for these coolers. Research shows that negligible fouling occurs in closed-loop evaporator tubes at 3 to 5 fps and 7 fps water velocities. AHRI Standard 480 discusses fouling calculations.

The refrigerant side of the cooler is not subject to fouling, and a fouling factor need not be included for that side.

Typically, the t/k term in Equations 10.16 and 10.17 may be negligible for material with high thermal conductivity. However, with low-thermal-conductivity material or thick-walled tubing, it may become significant. Refer to Chapter 4 of the 2017 *ASHRAE Handbook—Fundamentals* and to Chapter 39 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for further details.

Pressure drop is usually minimal in Baudelot and shell-and-coil coolers but must be considered in direct-expansion and flooded coolers. Both direct-expansion and flooded coolers rely on turbulent fluid flow to improve heat transfer. This turbulence is obtained at the expense of pressure drop.

For air-conditioning, pressure drop is commonly limited to 10 psi to keep pump size and energy cost reasonable. For flooded coolers, compare also Chapter 39 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* for a discussion of pressure drop for flow in tubes. Pressure drop for fluid flow in shell-and-tube direct-expansion coolers depends greatly on tube and baffle geometry. The following equation projects the change in pressure drop caused by a change in flow:

$$\text{New pressure drop} = \text{Original pressure drop}\left[\frac{\text{New rate}}{\text{Original rate}}\right]^{1.8} \tag{10.18}$$

The refrigerant-side pressure drop must be considered for direct-expansion, shell-and-coil, brazed-plate, and (sometimes) Baudelot coolers. When there is a pressure drop on the refrigerant side, the refrigerant inlet and outlet pressures and corresponding saturated temperature are different. This difference changes the mean temperature difference, which affects the total heat transfer rate. If pressure drop is high, expansion valve operation may be affected because of reduced pressure drop across the valve. This pressure drop varies, depending on the refrigerant used, operating temperature, and type of tubing.

When the fluid being cooled is electrically conductive, the system must be grounded to prevent electrochemical corrosion.

The constant superheat thermal expansion valve is the most common control used, located directly upstream of the cooler.

In flooded coolers, an orifice is often used as the throttling device between condenser and cooler.

Freeze prevention must be considered for coolers operating near the fluid's freezing point. In some coolers, freezing causes extensive damage. Two methods can be used for freeze protection: (1) hold saturated suction pressure above the fluid freezing point or (2) shut the system off if fluid temperature approaches the freezing point.

If the cooler is used only when ambient temperature is above freezing, drain the fluid from the cooler for cold weather. Alternatively, if the cooler is used year-round, the following methods can be used to prevent freezing:

- Heat tape or other heating device to keep cooler above freezing
- For water, adding an appropriate amount of ethylene glycol
- Continuous pump operation

Most compressors discharge a small percentage of oil in the discharge gas. This oil mixes with condensed refrigerant in the condenser and flows to the cooler. Because the oil is nonvolatile, it does not evaporate and may collect in the cooler.

In direct-expansion coolers, gas velocity in the tubes and suction gas header is usually sufficient to carry oil from the cooler into the suction line. From there, with proper piping design, it can be carried back to the compressor. At light load and low temperature, oil may gather in the superheat section of the cooler, detracting from performance. For this reason, operating refrigerant circuits at light load for long periods should be avoided, especially under low-temperature conditions.

In flooded coolers, vapor velocity above the tube bundle is usually insufficient to return oil up the suction line, and oil tends to accumulate in the cooler. With time, depending on the compressor oil loss rate, oil concentration in the cooler may become large. When concentration exceeds about 1%, heat transfer performance may be adversely affected if enhanced tubing is used.

It is common in flooded coolers to take some oil-rich liquid and return it to the compressor on a continuing basis, to establish a rate of return equal to the compressor oil loss rate.

Cooler maintenance centers around (1) safety and (2) cleaning the fluid side. Periodically inspect the cooler for any weakening of its pressure boundaries. The manufacturer or a service organization experienced in cooler maintenance should have details for cleaning.

A cooler operating at a saturated suction temperature lower than the ambient-air dew point should be insulated to prevent condensation.

Forced-Circulation Air Coolers

A cooling coil and a motor-driven fan are the basic components, and coil defrosting means are added for low-temperature operations where coil frosting might impede performance. Blow-through direct-drive propeller fans are most common, but for long throws, draw-through configuration is preferred. For loads above 32°F, coil spacing is usually 6 to 8 fins per inch; below 32°F a maximum of 4 fins per inch is preferred. Even distribution of halocarbon refrigerant is usually attained in direct-expansion coils by refrigerant distributors. Units in larger refrigeration systems are often liquid-pumped recirculating types with orifice disks.

Defrost for coils and drain pans of low-temperature units may be hot-gas, electric, or water. Usually defrosting is done with the fan off. Control of defrost is usually by microprocessor, with a thermostat mounted within the coil. Usually a rise to 45°F returns the unit to the operating cycle. Drain lines should be well-pitched, insulated, and trapped outside the refrigerated space.

Capacities of air coolers are usually based on the temperature difference between inlet air and refrigerant in the coil. The higher the TD, the lower the space relative humidity. Between 8°F and 16°F TD is usual, except for packaged products and workrooms where TD of 25°F is common. Low-temperature units generally have TD below 15°F for system economics and limiting defrost frequency.

Most frequent control of refrigerant flow is an expansion valve, most frequently thermostatic type. Electric expansion valves, requiring a valve, controller, and control sensor, are also available.

Large refrigerating systems more frequently have flooded evaporators, most often low-side float valves. Refrigerant valves opening or closing flow are usually solenoid valves. Larger flows may require pilot-operated solenoid valves. When it is desired to limit compressor motor load during pulldown, an evaporator pressure regulating valve may be used to limit compressor suction pressure.

11. AIR-CONDITIONING LOAD DATA

Cooling Loads

Obtain appropriate weather data and select design conditions. In addition to the conventional dry-bulb with mean coincident wet-bulb, also consider dew-point with mean coincident dry-bulb, particularly with spaces requiring large amounts of outdoor air or close control of moisture. Select indoor dry-bulb, wet-bulb, and ventilation rate, including permissible variations and control limits. Consider proposed schedules of occupancy, lighting, and processes that contribute to the internal load. Several different times of day and months must frequently be analyzed to determine the peak load time. See the appendix for climatic design conditions for selected locations. Chapter 14, Climatic Design Information, of the 2017 *ASHRAE Handbook—Fundamentals* has more extensive design data specific to 818 worldwide locations.

ANSI/ASHRAE/ACCA Standard 183-2007 sets the minimum standards for nonresidential load calculations.

Currently there are two ASHRAE cooling load calculation methods. The first is the Heat Balance (HB) method, whose equations are coded in a generic computer program linked to a user interface program. The source code for these programs is in the ASHRAE Load Calculation Toolkit.

The second method is the Radiant Time Series (RTS) method, a simplification of the heat balance method, still requiring a complex computer program for a multiroom building.

Due to the variation in heat transfer coefficients, precision of construction, and manner of actual building operation, a cooling load calculation can never be more than a good estimate of the actual load.

To design and size components of central air-conditioning systems, more than the cooling load is needed. Type of system, fan energy and location, direct heat loss and gain, duct leakage, heat extracted from lights, and type of return system must all be considered.

Heating Loads

Similar calculations to cooling load are made, but temperatures outside conditioned spaces are usually lower than space temperatures maintained. Solar heat gains, and internal heat gains are not included and thermal storage of building structure or content is usually ignored. This is usually sufficient to cope with a worst-case situation. There is very often need for cooling in cold months, for perimeter spaces with high solar heat gain and interior spaces with significant heat gain.

Previous Cooling Load Calculation Methods

Procedures described in Chapters 17 and 18 of the 2017 *ASHRAE Handbook–Fundamentals* are the most current and scientifically derived means for estimating cooling load for a defined building space, but methods in earlier editions of the ASHRAE Handbook are valid for many applications. These earlier procedures are simplifications of the Heat Balance principles, and their use requires experience to deal with atypical or unusual circumstances. In fact, any cooling or heating load estimate is no better than the assumptions used to define conditions and parameters such as physical makeup of the various envelope surfaces, conditions of occupancy and use, and ambient weather conditions. Experience of the practitioner can never be ignored.

The primary difference between the HB and RTS methods and the older methods is the newer methods' direct approach, compared to the simplifications necessitated by the limited computer capability available previously.

The **transfer function method (TFM)**, for example, required many calculation steps. It was originally designed for energy analysis with emphasis on daily, monthly, and annual energy use, and thus was more oriented to average hourly cooling loads than peak design loads.

The **total equivalent temperature differential method with time averaging (TETD/TA)** has been a highly reliable (if subjective) method of load estimating since its initial presentation in the 1967 *Handbook of Fundamentals*. Originally intended as a manual method of calculation, it proved suitable only as a computer application because of the need to calculate an extended profile of hourly heat gain values, from which radiant components had to be averaged over a time representative of the general mass of the building involved. Because perception of thermal storage characteristics of a given building is almost entirely subjective, with little specific information for the user to judge variations, the TETD/TA method's primary usefulness has always been to the experienced engineer.

The **cooling load temperature differential method with solar cooling load factors (CLTD/CLF)** attempted to simplify the two-step TFM and TETD/TA methods into a single-step technique that proceeded directly from raw data to cooling load without intermediate conversion of radiant heat gain to cooling load. A series of factors were taken from cooling load calculation results (produced by more sophisticated methods) as "cooling load temperature differences" and "cooling load factors" for use in traditional conduction ($q = UA\Delta t$) equations. The results are approximate cooling load values rather than simple heat gain values. The simplifications and assumptions used in the original work to derive those factors limit this method's applicability to those building types and conditions for which the CLTD/CLF factors were derived; the method should not be used beyond the range of applicability.

The TFM, TETD/TA, and CLTD/CLF procedures have not been invalidated or discredited. Experienced engineers have successfully used them in millions of buildings around the world. The accuracy of cooling load calculations in practice depends primarily on the availability of accurate information and the design engineer's judgment in the assumptions made in interpreting the available data. Those factors have much greater influence on a project's success than does the choice of a particular cooling load calculation method.

The primary benefit of HB and RTS calculations is their somewhat reduced dependency on purely subjective input (e.g., determining a proper time-averaging period for TETD/TA; ascertaining appropriate safety factors to add to the rounded-off TFM results; determining whether CLTD/CLF factors are applicable to a specific unique application). However, using the most up-to-date techniques in real-world design still requires judgment on the part of the design engineer and care in choosing appropriate assumptions, just as in applying older calculation methods.

Table 11.1 Summary of Load Sources and Equations for Estimating Space Design Cooling Load

Load Source	Equation	Reference, Table, Description
External		
Roof	$q = UA(\text{CLTD})$	Design heat transmission coefficients, Table 11.6 Areas calculated from plans CLTD, Tables 11.7–11.10
Walls	$q = UA(\text{CLTD})$	Design heat transmission coefficients, Table 11.6 Areas calculated from plans CLTD, Tables 11.11–11.13
Glass Conduction	$q = UA(\text{CLTD})$	Glass area calculated from plans U-factors, pg. 202 CLTD for conduction load through glass, pg. 202
Glass Solar	$q = A(\text{SC})\text{SCL}$	Solar cooling load (SCL) factors, Table 11.14 Net glass area from plans Shading coefficients for combination of glass and internal shading, Table 11.15 Compute shaded area from building projections Externally shaded glass: use north orientation data
Partitions, Ceilings, Floors	$q = UA(\text{TD})$	Design heat transmission coefficients, Table 11.6 Area calculated from plans
Internal		
Lights	$q = \text{INPUT}$	Input rating from electrical plans or lighting fixture data, Table 11.17
People		
Sensible	$q_s = \text{No. (Sens. H.G.)}$	Number of people in space Sensible heat gain from occupants, Table 11.16
Latent	$q_l = \text{No. (Lat. H.G.)}$	Latent heat gain from occupants
Equipment and Appliances	$q_s = \text{HEAT GAIN}$	Recommended rate of heat gain, Tables 11.18–11.35
Power	$q = \text{HEAT GAIN}$	p. 224
Infiltration Air		
Sensible	$q_s = 1.10\ (\text{CFM})\ \Delta t$	Inside-outside air temperature difference, °F
Latent	$q_l = 4840\ (\text{CFM})\ \Delta W$	Inside-outside air humidity ratio difference, grains/lbda
Total	$q = 4.5\ (\text{CFM})\ \Delta h$	Inside-outside air enthalpy difference, Btu/lb_{da}

CAUTION: Approximate data—Use for preliminary computations only. See *Load Calculation Applications Manual* for more details.

Heat Flow *Q* Through Building Materials

(In addition to heat flow through building materials the resistance of surfaces and air spaces must be included in calculating U-factors.)

$$Q \text{ (Btu/h)} = U \times \text{Area (ft}^2) \times \text{temperature difference (°F)} \tag{11.1}$$

where U = overall coefficient of heat transmission, Btu/h· ft^2· °F, of materials + interior and exterior resistances:

$$1/U = \Sigma R \text{ (resistance of components)} \tag{11.2}$$

For multiple layers of homogeneous materials, R values are added in series:

$$1/U = R_{cold\ surface} + R_1 + R_2 + R_n ... + R_{warm\ surface} \tag{11.3}$$

For wood stud walls, studs 16 in. on center (series and parallel):

$$1/U = R_{cold\ surface} + \left\{ \frac{+\ 0.25\ R_{stud}}{+\ 0.75\ R_{stud\ space}} \right\} + R_{warm\ surface} \tag{11.4}$$

(Plus, in series, $R_{insulation}$, R_{siding}, $R_{wallboard}$, etc.)

For metal framed construction, heat flow through the metal causes thermal bridging, increasing the U-factor significantly.

Conductive Heat Flow Through Glazing

Solar radiation gain through glazing is usually more significant in cooling load calculations than conductive heat gain. Solar heat gain is neglected in heating load calculations.

Conductive heat flow through glazing including surface resistance (approximate data)

Single glazing	$U = 1.1$
Double glazing	$U = 0.55$
Triple glazing	$U = 0.33$

Table 11.2 Effective Thermal Resistance of Plane Air Spaces,[a,b,c] h·ft²·°F/Btu
[2017F, Ch 26, Tbl 3]

Position of Air Space	Direction of Heat Flow	Air Space		Effective Emittance ε_{eff}[d,e]									
		Mean Temp.[d], °F	Temp. Diff.,[d] °F	0.5 in. Air Space[c]					0.75 in. Air Space[c]				
				0.03	0.05	0.2	0.5	0.82	0.03	0.05	0.2	0.5	0.82
Horiz.	Up ↑	90	10	2.13	2.03	1.51	0.99	0.73	2.34	2.22	1.61	1.04	0.75
		50	30	1.62	1.57	1.29	0.96	0.75	1.71	1.66	1.35	0.99	0.77
		50	10	2.13	2.05	1.60	1.11	0.84	2.30	2.21	1.70	1.16	0.87
		0	20	1.73	1.70	1.45	1.12	0.91	1.83	1.79	1.52	1.16	0.93
		0	10	2.10	2.04	1.70	1.27	1.00	2.23	2.16	1.78	1.31	1.02
		–50	20	1.69	1.66	1.49	1.23	1.04	1.77	1.74	1.55	1.27	1.07
		–50	10	2.04	2.00	1.75	1.40	1.16	2.16	2.11	1.84	1.46	1.20
Vertical	Horiz. →	90	10	2.47	2.34	1.67	1.06	0.77	3.50	3.24	2.08	1.22	0.84
		50	30	2.57	2.46	1.84	1.23	0.90	2.91	2.77	2.01	1.30	0.94
		50	10	2.66	2.54	1.88	1.24	0.91	3.70	3.46	2.35	1.43	1.01
		0	20	2.82	2.72	2.14	1.50	1.13	3.14	3.02	2.32	1.58	1.18
		0	10	2.93	2.82	2.20	1.53	1.15	3.77	3.59	2.64	1.73	1.26
		–50	20	2.90	2.82	2.35	1.76	1.39	2.90	2.83	2.36	1.77	1.39
		–50	10	3.20	3.10	2.54	1.87	1.46	3.72	3.60	2.87	2.04	1.56
Horiz.	Down ↓	90	10	2.48	2.34	1.67	1.06	0.77	3.55	3.29	2.10	1.22	0.85
		50	30	2.66	2.54	1.88	1.24	0.91	3.77	3.52	2.38	1.44	1.02
		50	10	2.67	2.55	1.89	1.25	0.92	3.84	3.59	2.41	1.45	1.02
		0	20	2.94	2.83	2.20	1.53	1.15	4.18	3.96	2.83	1.81	1.30
		0	10	2.96	2.85	2.22	1.53	1.16	4.25	4.02	2.87	1.82	1.31
		–50	20	3.25	3.15	2.58	1.89	1.47	4.60	4.41	3.36	2.28	1.69
		–50	10	3.28	3.18	2.60	1.90	1.47	4.71	4.51	3.42	2.30	1.71

[a] See Chapter 25 of the 2017 *ASHRAE Handbook—Fundamentals*. Thermal resistance values were determined from $R = 1/C$, where $C = h_c + \varepsilon_{eff} h_r$, h_c is conduction/convection coefficient, $\varepsilon_{eff} h_r$ is radiation coefficient $\approx 0.0068\varepsilon_{eff}[(t_m + 460)/100]^3$, and t_m is mean temperature of air space.

[b] Values apply for ideal conditions (i.e., air spaces of uniform thickness bounded by plane, smooth, parallel surfaces with no air leakage to or from the space). **This table should not be used for hollow siding or profiled cladding.**

[c] A single resistance value cannot account for multiple air spaces; each air space requires a separate resistance calculation that applies only for established boundary conditions. Resistances of horizontal spaces with heat flow downward are substantially independent of temperature difference.

[d] Interpolation is permissible for other values of mean temperature, temperature difference, and effective emittance ε_{eff}. Interpolation and moderate extrapolation for air spaces greater than 3.5 in. are also permissible.

[e] Effective emittance ε_{eff} of air space is given by $1/\varepsilon_{eff} = 1/\varepsilon_1 + 1/\varepsilon_2 - 1$, where ε_1 and ε_2 are emittances of surfaces of air space (see 2017F, Ch 26, Tbl 2). **Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where the highly reflective surface can be maintained over the service life of the assembly.**

Table 11.3 Surface Film Coefficients/Resistances [2017F, Ch 26, Tbl 10]

Position of Surface	Direction of Heat Flow	Surface Emittance, ε					
		Nonreflective ε = 0.90		Reflective			
				ε = 0.20		ε = 0.05	
Indoor		h_i	R_i	h_i	R_i	h_i	R_i
Horizontal	Upward	1.63	0.61	0.91	1.10	0.76	1.32
Sloping at 45°	Upward	1.60	0.62	0.88	1.14	0.73	1.37
Vertical	Horizontal	1.46	0.68	0.74	1.35	0.59	1.70
Sloping at 45°	Downward	1.32	0.76	0.60	1.67	0.45	2.22
Horizontal	Downward	1.08	0.92	0.37	2.70	0.22	4.55
Outdoor (any position)		h_o	R_o				
15 mph wind (for winter)	Any	6.00	0.17	—	—	—	—
7.5 mph wind (for summer)	Any	4.00	0.25	—	—	—	—

Notes:
1. Surface conductance h_i and h_o measured in Btu/h·ft^2·°F; resistance R_i and R_o in h·ft^2·°F/Btu.
2. No surface has both an air space resistance value and a surface resistance value.
3. Conductances are for surfaces of the stated emittance facing virtual blackbody surroundings at same temperature as ambient air. Values based on surface/air temperature difference of 10°F and surface temperatures of 70°F.
4. See Chapter 4 of the 2017 *ASHRAE Handbook—Fundamentals* for more detailed information.
5. Condensate can have significant effect on surface emittance (see 2017F, Ch 26, Tbl 2). Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where highly reflective surface can be maintained over the service life of the assembly.

Table 11.4 Emissivity of Various Surfaces and Effective Emittances of Facing Air Spaces[a] [2017F, Ch 26, Tbl 2]

Surface	Average Emissivity ε	Effective Emittance ε_{eff} of Air Space	
		One Surface's Emittance ε; Other, 0.9	Both Surfaces' Emittance ε
Aluminum foil, bright	0.05	0.05	0.03
Aluminum foil, with condensate just visible (>0.7 g/ft^2)	0.30[b]	0.29	—
Aluminum foil, with condensate clearly visible (>2.9 g/ft^2)	0.70[b]	0.65	—
Aluminum sheet	0.12	0.12	0.06
Aluminum-coated paper, polished	0.20	0.20	0.11
Brass, nonoxidized	0.04	0.038	0.02
Copper, black oxidized	0.74	0.41	0.59
Copper, polished	0.04	0.038	0.02
Iron and steel, polished	0.2	0.16	0.11
Iron and steel, oxidized	0.58	0.35	0.41
Lead, oxidized	0.27	0.21	0.16
Nickel, nonoxidized	0.06	0.056	0.03
Silver, polished	0.03	0.029	0.015
Steel, galvanized, bright	0.25	0.24	0.15
Tin, nonoxidized	0.05	0.047	0.026
Aluminum paint	0.50	0.47	0.35
Building materials: wood, paper, masonry, nonmetallic paints	0.90	0.82	0.82
Regular glass	0.84	0.77	0.72

[a]Values apply in 4 to 40 μm range of electromagnetic spectrum. Also, oxidation, corrosion, and accumulation of dust and dirt can dramatically increase surface emittance. Emittance values of 0.05 should only be used where the highly reflective surface can be maintained over the service life of the assembly. Except as noted, data from VDI (1999).

[b]Values based on data in Bassett and Trethowen (1984).

Table 11.5 Effective Thermal Resistance of Ventilated Attics[a] (Summer Condition)

Ventilation Air Temp., °F	Sol-Air[e] Temp., °F	Not Ventilation[b]		Natural Ventilation		Power Ventilation[c]					
		Ventilation Rate, cfm/ft²									
		0		0.1		0.5		1.0		1.5	
		Ceiling Resistance R[d], °F·ft²·h/Btu									
		10	20	10	20	10	20	10	20	10	20
Part A. Nonreflective Surfaces											
80	120	1.9	1.9	2.8	3.4	6.3	9.3	9.6	16	11	20
	140	1.9	1.9	2.8	3.5	6.5	10	9.8	17	12	21
	160	1.9	1.9	2.8	3.6	6.7	11	10	18	13	22
100	120	1.9	1.9	2.2	2.3	3.3	4.4	4.0	6.0	4.1	6.9
	140	1.9	1.9	2.4	2.7	4.2	6.1	5.8	8.7	6.5	10
	160	1.9	1.9	2.6	3.2	5.0	7.6	7.2	11	8.3	13
Part B. Reflective Surfaces[f]											
80	120	6.5	6.5	8.1	8.8	13	17	17	25	19	30
	140	6.5	6.5	8.2	9.0	14	18	18	26	20	31
	160	6.5	6.5	8.3	9.2	15	18	19	27	21	32
100	120	6.5	6.5	7.0	7.4	8.0	10	8.5	12	8.8	12
	140	6.5	6.5	7.3	7.8	10	12	11	15	12	16
	160	6.5	6.5	7.6	8.2	11	14	13	18	15	20

[a]Although the term effective resistance is commonly used when there is attic ventilation, this table includes values for situations with no ventilation. The effective resistance of the attic added to the resistance (1/U) of the ceiling yields the effective resistance of this combination based on sol-air and room temperatures. These values apply to wood frame construction with a roof deck and roofing that has a conductance of 1.0 Btu/h·ft²·°F.

[b]This condition cannot be achieved in the field unless extreme measures are taken to tightly seal the attic.

[c]Based on air discharging outward from attic.

[d]When determining ceiling resistance, do not add the effect of a reflective surface facing the attic, as it is accounted for in part B of this table.

[e]Roof surface temperature rather than sol-air temperature can be used if 0.25 is subtracted from the attic resistance shown.

[f]Surfaces with effective emittance $\varepsilon_{eff} = 0.05$ between ceiling joists facing attic space.

Table 11.6 Building and Insulating Materials: Design Values[a]
[2017F, Ch 26, Tbl 1]

Description	Density, lb/ft³	Conductivity[b] k, Btu·in/h·ft²·°F	Resistance R, h·ft²·°F/Btu	Specific Heat c_p, Btu/lb·°F
Insulating Materials				
Blanket and batt[c,d]				
Glass-fiber batts				0.2
	0.47 to 0.51	0.32 to 0.33	—	—
	0.61 to 0.75	0.28 to 0.30	—	—
	0.79 to 0.85	0.26 to 0.27	—	—
	1.4	0.23	—	—
Rock and slag wool batts	—	—	—	0.2
	2 to 2.3	0.25 to 0.26	—	—
	2.8	0.23 to 0.24	—	—
Mineral wool, felted	1 to 3	0.28	—	—
	1 to 8	0.24	—	—
Board and slabs				
Cellular glass	7.5	0.29	—	0.20
Cement fiber slabs, shredded wood with				
Portland cement binder	25 to 27	0.50 to 0.53	—	—
with magnesia oxysulfide binder	22	0.57	—	0.31
Glass fiber board	—	—	—	0.2
	1.5 to 6.0	0.23 to 0.24	—	—
Expanded rubber (rigid)	4	0.2	—	0.4
Extruded polystyrene, smooth skin	—	—	—	0.35
aged per Can/ULC Standard S770-2003	1.4 to 3.6	0.18 to 0.20	—	—
aged 180 days	1.4 to 3.6	0.20		
European product	1.9	0.21		
aged 5 years at 75°F	2 to 2.2	0.21	—	—
blown with low global warming potential (GWP) (<5) blowing agent		0.24 to 0.25	—	—
Expanded polystyrene, molded beads	—	—	—	0.35
	1.0 to 1.5	0.24 to 0.26	—	—
	1.8	0.23	—	—
Mineral fiberboard, wet felted	10	0.26	—	0.2
Rock wool board	—	—	—	0.2
floors and walls	4.0 to 8.0	0.23 to 0.25	—	—
roofing	10. to 11.	0.27 to 0.29	—	0.2
Acoustical tile[e]	21 to 23	0.36 to 0.37	—	0.14 to 0.19
Perlite board	9	0.36	—	—
Polyisocyanurate	—	—	—	0.35
unfaced, aged per Can/ULC Standard S770-2003	1.6 to 2.3	0.16 to 0.17	—	—
with foil facers, aged 180 days	—	0.15 to 0.16	—	—
Phenolic foam board with facers, aged	—	0.14 to 0.16	—	—
Loose fill				
Cellulose fiber, loose fill	—	—	—	0.33
attic application up to 4 in.	1.0 to 1.2	0.31 to 0.32	—	—
attic application > 4 in.	1.2 to 1.6	0.27 to 0.28	—	—
wall application, densely packed	3.5	0.27 – 0.28	—	—
Perlite, expanded	2 to 4	0.27 to 0.31	—	0.26
	4 to 7.5	0.31 to 0.36	—	—
	7.5 to 11	0.36 to 0.42	—	—
Glass fiber[d]				
attics, ~4 to 12 in.	0.4 to 0.5	0.36 to 0.38	—	—
attics, ~12 to 22 in.	0.5 to 0.6	0.34 to 0.36	—	—
closed attic or wall cavities	1.8 to 2.3	0.24 to 0.25	—	—
Rock and slag wool[d]				
attics, ~3.5 to 4.5 in.	1.5 to 1.6	0.34	—	—
attics, ~5 to 17 in.	1.5 to 1.8	0.32 to 0.33	—	—
closed attic or wall cavities	4.0	0.27 to 0.29	—	—
Vermiculite, exfoliated	7.0 to 8.2	0.47	—	0.32
	4.0 to 6.0	0.44	—	—
Spray applied				
Cellulose, sprayed into open wall cavities	1.6 to 2.6	0.27 to 0.28	—	—
Glass fiber, sprayed into open wall or attic cavities	1.0	0.27 to 0.29	—	—
	1.8 to 2.3	0.23 to 0.26	—	—
Polyurethane foam	—	—	—	0.35
low density, open cell	0.45 to 0.65	0.26 to 0.29	—	—
medium density, closed cell, aged 180 days	1.9 to 3.2	0.14 to 0.20	—	—

Table 11.6 Building and Insulating Materials: Design Values[a]
[2017F, Ch 26, Tbl 1] ***(Continued)***

Description	Density, lb/ft³	Conductivity[b] *k*, Btu·in/h·ft²·°F	Resistance *R*, h·ft²·°F/Btu	Specific Heat c_p, Btu/lb·°F
Building Board and Siding				
Board				
Asbestos/cement board	120	4	—	0.24
Cement board	71	1.7	—	0.2
Fiber/cement board	88	1.7	—	0.2
	61	1.3	—	0.2
	26	0.5	—	0.45
	20	0.4	—	0.45
Gypsum or plaster board	40	1.1	—	0.21
Oriented strand board (OSB) 7/16 in.	41	—	0.62	0.45
1/2 in.	41	—	0.68	0.45
Plywood (douglas fir) 1/2 in.	29	—	0.79	0.45
5/8 in.	34	—	0.85	0.45
Plywood/wood panels 3/4 in.	28	—	1.08	0.45
Vegetable fiber board				
sheathing, regular density 1/2 in.	18	—	1.32	0.31
intermediate density 1/2 in.	22	—	1.09	0.31
nail-based sheathing 1/2 in.	25	—	1.06	0.31
shingle backer 3/8 in.	18	—	0.94	0.3
sound-deadening board1/2 in.	15	—	1.35	0.3
tile and lay-in panels, plain or acoustic	18	0.4	—	0.14
laminated paperboard	30	0.5	—	0.33
homogeneous board from repulped paper	30	0.5	—	0.28
Hardboard				
medium density	50	0.73	—	0.31
high density, service-tempered and service grades	55	0.82	—	0.32
high density, standard-tempered grade	63	1.0	—	0.32
Particleboard				
low density	37	0.71	—	0.31
medium density	50	0.94	—	0.31
high density	62	1.18	0.85	—
underlayment 5/8 in.	44	0.73	0.82	0.29
Waferboard	37	0.63	0.21	0.45
Shingles				
Asbestos/cement	120	—	0.21	—
Wood, 16 in., 7 1/2 in. exposure	—	—	0.87	0.31
Wood, double, 16 in., 12 in. exposure	—	—	1.19	0.28
Wood, plus ins. backer board 5/16 in.	—	—	1.4	0.31
Siding				
Asbestos/cement, lapped 1/4 in.	—	—	0.21	0.24
Asphalt roll siding	—	—	0.15	0.35
Asphalt insulating siding (1/2 in. bed)	—	—	0.21	0.24
Hardboard siding 7/16 in.	—	—	0.15	0.35
Wood, drop, 8 in. 1 in.	—	—	0.79	0.28
Wood, bevel				
8 in., lapped1/2 in.	—	—	0.81	0.28
10 in., lapped3/4 in.	—	—	1.05	0.28
Wood, plywood, 3/8 in., lapped	—	—	0.59	0.29
Aluminum, steel, or vinyl,[h, i] over sheathing				—
hollow-backed	—	—	0.62	0.29[i]
insulating-board-backed 3/8 in.	—	—	1.82	0.32
foil-backed 3/8 in.	—	—	2.96	—
Architectural (soda-lime float) glass	158	6.9	—	0.21
Building Membrane				
Vapor-permeable felt	—	—	0.06	—
Vapor: seal, 2 layers of mopped 15 lb felt	—	—	0.12	—
Vapor: seal, plastic film	—	—	Negligible	—
Finish Flooring Materials				
Carpet and rebounded urethane pad 3/4 in.	7	—	2.38	—
Carpet and rubber pad (one-piece) 3/8 in.	20	—	0.68	—
Pile carpet with rubber pad 3/8 to 1/2 in.	18	—	1.59	—
Linoleum/cork tile 1/4 in.	29	—	0.51	—
PVC/rubber floor covering	—	2.8	—	—
rubber tile 1.0 in.	119	—	0.34	—
terrazzo 1.0 in.	—	—	0.08	0.19
Metals (See 2013F, Ch 33, Tbl 3)				

Table 11.6 Building and Insulating Materials: Design Values[a]
[2017F, Ch 26, Tbl 1] ***(Continued)***

Description	Density, lb/ft³	Conductivity[b] *k*, Btu·in/h·ft²·°F	Resistance *R*, h·ft²·°F/Btu	Specific Heat c_p, Btu/lb·°F
Roofing				
Asbestos/cement shingles	120	—	0.21	0.24
Asphalt (bitumen with inert fill)	100	2.98	—	—
	119	4.0	—	—
	144	7.97	—	—
Asphalt roll roofing	70	—	0.15	0.36
Asphalt shingles	70	—	0.44	0.3
Built-up roofing 3/8 in.	70	—	0.33	0.35
Mastic asphalt (heavy, 20% grit)	59	1.32	—	—
Reed thatch	17	0.62	—	—
Roofing felt	141	8.32	—	—
Slate 1/2 in.	—	—	0.05	0.3
Straw thatch	15	0.49	—	—
Wood shingles, plain and plastic-film-faced	—	—	0.94	0.31
Plastering Materials				
Cement plaster, sand aggregate	116	5.0	—	0.2
Sand aggregate 3/8 in.	—	—	0.08	0.2
3/4 in.	—	—	0.15	0.2
Gypsum plaster	70	2.63	—	—
	80	3.19	—	—
Lightweight aggregate 1/2 in.	45	—	0.32	—
5/8 in.	45	—	0.39	—
on metal lath 3/4 in.	—	—	0.47	—
Perlite aggregate	45	1.5	—	0.32
Sand aggregate	105	5.6	—	0.2
on metal lath 3/4 in.	—	—	0.13	—
Vermiculite aggregate	30	1.0	—	—
	40	1.39	—	—
	45	1.7	—	—
	50	1.8	—	—
	60	2.08	—	—
Perlite plaster	25	0.55	—	—
	38	1.32	—	—
Pulpboard or paper plaster	38	0.48	—	—
Sand/cement plaster, conditioned	98	4.4	—	—
Sand/cement/lime plaster, conditioned	90	3.33	—	—
Sand/gypsum (3:1) plaster, conditioned	97	4.5	—	—
Masonry Materials				
Masonry units				
Brick, fired clay	150	8.4 to 10.2	—	—
	140	7.4 to 9.0	—	—
	130	6.4 to 7.8	—	—
	120	5.6 to 6.8	—	0.19
	110	4.9 to 5.9	—	—
	100	4.2 to 5.1	—	—
	90	3.6 to 4.3	—	—
	80	3.0 to 3.7	—	—
	70	2.5 to 3.1	—	—
Clay tile, hollow				
1 cell deep 3 in.	—	—	0.80	0.21
4 in.	—	—	1.11	—
2 cells deep 6 in.	—	—	1.52	—
8 in.	—	—	1.85	—
10 in.	—	—	2.22	—
3 cells deep 12 in.	—	—	2.50	—
Lightweight brick	50	1.39	—	—
	48	1.51	—	—
Concrete blocks[f, g]				
Limestone aggregate				
8 in., 36 lb, 138 lb/ft³ concrete, 2 cores	—	—	—	—
with perlite-filled cores	—	—	2.1	—
12 in., 55 lb, 138 lb/ft³ concrete, 2 cores	—		—	—
with perlite-filled cores	—	—	3.7	—

Table 11.6 Building and Insulating Materials: Design Values[a]
[2017F, Ch 26, Tbl 1] ***(Continued)***

Description	Density, lb/ft^3	Conductivity[b] k, $Btu \cdot in/h \cdot ft^2 \cdot °F$	Resistance R, $h \cdot ft^2 \cdot °F/Btu$	Specific Heat c_p, $Btu/lb \cdot °F$
Normal-weight aggregate (sand and gravel)				
8 in., 33 to 36 lb, 126 to 136 lb/ft^3 concrete, 2 or 3 cores	—	—	1.11 to 0.97	0.22
with perlite-filled cores	—	—	2.0	—
with vermiculite-filled cores	—	—	1.92 to 1.37	—
12 in., 50 lb, 125 lb/ft^3 concrete, 2 cores	—	—	1.23	0.22
Medium-weight aggregate (combinations of normal and lightweight aggregate)				
8 in., 26 to 29 lb, 97 to 112 lb/ft^3 concrete, 2 or 3 cores	—	—	1.71 to 1.28	—
with perlite-filled cores	—	—	3.7 to 2.3	—
with vermiculite-filled cores	—	—	3.3	—
with molded-EPS-filled (beads) cores	—	—	3.2	—
with molded EPS inserts in cores	—	—	2.7	—
Lightweight aggregate (expanded shale, clay, slate or slag, pumice)				
6 in., 16 to 17 lb, 85 to 87 lb/ft^3 concrete, 2 or 3 cores	—	—	1.93 to 1.65	—
with perlite-filled cores	—	—	4.2	—
with vermiculite-filled cores	—	—	3.0	—
8 in., 19 to 22 lb, 72 to 86 lb/ft^3 concrete	—	—	3.2 to 1.90	0.21
with perlite-filled cores	—	—	6.8 to 4.4	—
with vermiculite-filled cores	—	—	5.3 to 3.9	—
with molded-EPS-filled (beads) cores	—	—	4.8	—
with UF foam-filled cores	—	—	4.5	—
with molded EPS inserts in cores	—	—	3.5	—
12 in., 32 to 36 lb, 80 to 90 lb/ft^3, concrete, 2 or 3 cores	—	—	2.6 to 2.3	—
with perlite-filled cores	—	—	9.2 to 6.3	—
with vermiculite-filled cores	—	—	5.8	—
Stone, lime, or sand	180	72	—	—
Quartzitic and sandstone	160	43	—	—
	140	24	—	—
	120	13	—	0.19
Calcitic, dolomitic, limestone, marble, and granite	180	30	—	—
	160	22	—	—
	140	16	—	—
	120	11	—	0.19
	100	8	—	—
Gypsum partition tile				
3 by 12 by 30 in., solid	—	—	1.26	0.19
4 cells	—	—	1.35	—
4 by 12 by 30 in., 3 cells	—	—	1.67	—
Limestone	150	3.95	—	0.2
	163	6.45	—	0.2
Concretes[i]				
Sand and gravel or stone aggregate concretes	150	10.0 to 20.0	—	—
(concretes with >50% quartz or quartzite sand have conductivities in higher end of	140	9.0 to 18.0	—	0.19 to 0.24
range)	130	7.0 to 13.0	—	—
Lightweight aggregate or limestone concretes	120	6.4 to 9.1	—	—
expanded shale, clay, or slate; expanded slags; cinders;	100	4.7 to 6.2	—	0.2
pumice (with density up to 100 lb/ft^3); scoria (sanded	80	3.3 to 4.1	—	0.2
concretes have conductivities in higher	60	2.1 to 2.5	—	—
end of range)	40	1.3	—	—
Gypsum/fiber concrete (87.5% gypsum, 12.5% wood chips)	51	1.66	—	0.2
Cement/lime, mortar, and stucco	120	9.7	—	—
	100	6.7	—	—
	80	4.5	—	—

Table 11.6 Building and Insulating Materials: Design Values[a]
[2017F, Ch 26, Tbl 1] ***(Continued)***

Description	Density, lb/ft³	Conductivity[b] k, Btu·in/h·ft²·°F	Resistance R, h·ft²·°F/Btu	Specific Heat c_p, Btu/lb·°F
Perlite, vermiculite, and polystyrene beads	50	1.8 to 1.9	—	—
	40	1.4 to 1.5	—	0.15 to 0.23
	30	1.1	—	—
	20	0.8	—	—
Foam concretes	120	5.4	—	—
	100	4.1	—	—
	80	3.0	—	—
	70	2.5	—	—
Foam concretes and cellular concretes	60	2.1	—	—
	40	1.4	—	—
	20	0.8	—	—
Aerated concrete (oven-dried)	27 to 50	1.4	—	0.2
Polystyrene concrete (oven-dried)	16 to 50	2.54	—	0.2
Polymer concrete	122	11.4	—	—
	138	7.14	—	—
Polymer cement	117	5.39	—	—
Slag concrete	60	1.5	—	—
	80	2.25	—	—
	100	3	—	—
	125	8.53	—	—
Woods (12% moisture content)[j]				
Hardwoods	—	—	—	0.39[k]
Oak	41 to 47	1.12 to 1.25	—	—
Birch	43 to 45	1.16 to 1.22	—	—
Maple	40 to 44	1.09 to 1.19	—	—
Ash	38 to 42	1.06 to 1.14	—	—
Softwoods	—	—	—	0.39[k]
Southern pine	36 to 41	1.00 to 1.12	—	—
Southern yellow pine	31	1.06 to 1.16	—	—
Eastern white pine	25	0.85 to 0.94	—	—
Douglas fir/larch	34 to 36	0.95 to 1.01	—	—
Southern cypress	31 to 32	0.90 to 0.92	—	—
Hem/fir, spruce/pine/fir	24 to 31	0.74 to 0.90	—	—
Spruce	25	0.74 to 0.85	—	—
Western red cedar	22	0.83 to 0.86	—	—
West coast woods, cedars	22 to 31	0.68 to 0.90	—	—
Eastern white cedar	23	0.82 to 0.89	—	—
California redwood	24 to 28	0.74 to 0.82	—	—
Pine (oven-dried)	23	0.64	—	0.45
Spruce (oven-dried)	25	0.69	—	0.45

Notes for Table 11.6

[a]Values are for mean temperature of 75°F. Representative values for dry materials are intended as design (not specification) values for materials in normal use. Thermal values of insulating materials may differ from design values depending on in-situ properties (e.g., density and moisture content, orientation, etc.) and manufacturing variability. For properties of specific product, use values supplied by manufacturer or unbiased tests.

[b]Symbol λ also used to represent thermal conductivity.

[c]Does not include paper backing and facing, if any. Where insulation forms boundary (reflective or otherwise) of airspace, see 2017F, Ch 26, Tbls 2 and 3 for insulating value of airspace with appropriate effective emittance and temperature conditions of space.

[d]Conductivity varies with fiber diameter (see Chapter 25 of the 2017 *ASHRAE Handbook—Fundamentals*). Batt, blanket, and loose-fill mineral fiber insulations are manufactured to achieve specified R-values, the most common of which are listed in the table. Because of differences in manufacturing processes and materials, the product thicknesses, densities, and thermal conductivities vary over considerable ranges for a specified R-value.

[e]Insulating values of acoustical tile vary, depending on density of board and on type, size, and depth of perforations.

[f]Values for fully grouted block may be approximated using values for concrete with similar unit density.

[g]Values for concrete block and concrete are at moisture contents representative of normal use.

[h]Values for metal or vinyl siding applied over flat surfaces vary widely, depending on ventilation of the airspace beneath the siding; whether airspace is reflective or nonreflective; and on thickness, type, and application of insulating backing-board used. Values are averages for use as design guides, and were obtained from several guarded hot box tests (ASTM Standard C1363) on hollow-backed types and types made using backing of wood fiber, foamed plastic, and glass fiber. Departures of ±50% or more from these values may occur.

[i]Vinyl specific heat = 0.25 Btu/lb·°F.

[j]See Adams (1971), MacLean (1941), and Wilkes (1979). Conductivity values listed are for heat transfer across the grain. Thermal conductivity of wood varies linearly with density, and density ranges listed are those normally found for wood species given. If density of wood species is not known, use mean conductivity value. For extrapolation to other moisture contents, the following empirical equation developed by Wilkes (1979) may be used:

$$k = 0.1791 + \frac{(1.874 \times 10^{-2} + 5.753 \times 10^{-4} M)\rho}{1 + 0.01 M}$$

where ρ is density of moist wood in lb/ft³, and M is moisture content in percent.

[k]From Wilkes (1979), an empirical equation for specific heat of moist wood at 75°F is as follows:

$$c_p = \frac{(0.299 + 0.01 M)}{(1 + 0.01 M)} + \Delta c_p$$

where Δc_p accounts for heat of sorption and is denoted by

$$\Delta c_p = M(1.921 \times 10^{-3} - 3.168 \times 10^{-5} M)$$

where M is moisture content in percent by mass.

Cooling Load Temperature Differences (CLTDs)

Table 11.7 CLTDs for Flat Roofs—24°N Latitude, July

Roof No	Solar time, h											
	2	4	6	8	10	12	14	16	18	20	22	24
1	–2	–5	–6	9	44	76	92	86	58	23	8	2
2	0	–4	–6	1	30	64	86	89	70	36	14	5
3	8	2	–2	3	22	47	68	77	68	47	29	16
4	11	3	–2	–4	5	27	55	75	80	67	43	23
5	16	8	3	1	10	30	52	68	70	59	41	27
8	24	17	11	9	14	27	43	54	58	52	42	32
9	25	16	9	4	5	17	36	54	65	63	51	37
10	31	22	15	9	8	16	30	45	56	59	52	41
13	31	25	20	16	16	23	33	43	49	49	43	37
14	32	27	23	19	19	24	32	40	45	45	42	37

Table 11.8 CLTDs for Flat Roofs—36°N Latitude, July

Roof No.	Solar time, h											
	2	4	6	8	10	12	14	16	18	20	22	24
1	–2	–5	–6	12	45	75	90	84	60	26	9	2
2	0	–4	–6	4	32	63	84	87	70	39	15	5
3	8	2	–2	4	24	47	67	75	68	48	30	17
4	11	3	–1	–3	7	29	55	74	79	67	45	24
5	16	8	3	2	12	31	52	67	70	59	42	27
8	25	17	12	9	15	28	42	54	58	53	43	33
9	26	16	9	4	7	19	37	54	64	63	52	38
10	32	23	15	10	9	17	30	45	56	58	52	42
13	31	25	20	16	17	24	33	43	49	49	44	37
14	32	28	23	20	20	25	32	40	45	46	42	37

Table 11.9 CLTDs for Flat Roofs—48°N Latitude, July

Roof No.	Solar time, h											
	2	4	6	8	10	12	14	16	18	20	22	24
1	–2	–5	–5	15	44	69	83	79	59	29	9	2
2	0	–4	–5	6	32	60	78	81	68	41	16	5
3	8	2	–1	6	24	45	63	71	65	48	30	17
4	12	3	–1	–2	8	29	52	69	74	65	45	25
5	16	8	3	3	13	31	49	63	66	58	42	27
8	24	17	11	10	16	27	40	51	55	51	42	32
9	26	16	9	5	8	19	35	51	60	61	51	38
10	31	22	15	10	10	17	29	43	53	56	51	41
13	30	25	20	16	18	24	32	41	47	47	43	37
14	32	27	23	20	20	24	31	38	43	44	41	36

CAUTION: Approximate data—Use for preliminary computations only. Also, see notes on next page.

Notes for CLTD Data for Flat Roofs

1. Data apply directly to (1) dark surface, (2) indoor temperature is 78°F, (3) outdoor maximum temperature of 95°F with mean temperature of 85°F and daily range of 21°F, (4) solar radiation typical of clear day on 21st day of month, (5) outside surface film resistance of 0.333 h·ft^2·°F/Btu, and (6) inside surface resistance of 0.685 h·ft^2·°F/Btu.
2. Adjustments to design temperatures

$$\text{Corr. CLTD} = \text{CLTD} + (78 - t_r) + (t_m - 85) \tag{11.5}$$

where t_r = inside temperature and t_m = mean outdoor temperature, or t_m = maximum outdoor temperature – (daily range)/2.

No adjustment recommended for color or for ventilation of air space above a ceiling.

For design purposes, the data suffice for plus or minus 2 weeks from the 21st day of given month.

Table 11.10 Roof Classifications for Use with CLTD Tables for Flat Roofs

Mass Location	Suspended Ceiling	*R*, h·ft^2·°F/Btu	Wood 1 in.	2 in. (Heavyweight) Concrete	Steel Deck	Attic Ceiling Comb.
Mass inside insul.	Without	0 to 10	*	2	*	*
		10 to 20	*	4	*	*
		20 to 25	*	5	*	*
	With	0 to 5	*	5	*	*
		5 to 10	*	8	*	*
		10 to 20	*	13	*	*
		20 to 25	*	14	*	*
Mass evenly placed	Without	0 to 5	1	2	1	1
		5 to 15	2	*	1	2
		15 to 25	4	*	2	2
	With	0 to 5	*	3	1	*
		5 to 10	4	*	1	*
		10 to 15	5	*	2	
		15 to 20	9	*	2	*
		20 to 25	10	*	4	*
Mass outside insul.	Without	0 to 5	*	2	*	*
		5 to 10	*	3	*	*
		10 to 15	*	4	*	*
		15 to 25	*	5	*	*
	With	0 to 10	*	3	*	*
		10 to 15	*	4	*	*
		15 to 20	*	5	*	*

*Denotes roof that is not possible with the chosen parameters

Table 11.11 Approximate CLTDs for Sunlit Walls—24°N Latitude, July

Wall Facing	Solar time, h								Solar time, h								Solar time, h							
	6	8	10	12	14	16	18	20	6	8	10	12	14	16	18	20	6	8	10	12	14	16	18	20
	Low Mass, Low R-Value Wall								Low Mass, Medium R-Value Wall								Low Mass, High R-Value Wall							
N	–2	13	18	22	28	32	34	17	1	0	6	13	18	23	28	30	–2	2	12	18	23	28	32	29
NE	0	39	53	39	30	30	24	13	0	3	20	36	39	35	32	27	–2	9	36	46	38	32	29	22
E	0	44	63	48	32	30	24	13	1	3	22	43	46	40	34	28	–2	10	42	55	44	35	30	23
SE	–2	25	44	42	32	30	24	13	0	1	13	28	35	35	32	27	–2	4	26	40	38	33	29	22
S	–3	3	12	24	31	30	23	13	0	–1	1	7	16	24	27	25	–2	–1	4	13	24	29	28	22
SW	–3	3	13	22	40	58	52	20	1	–1	1	7	15	29	43	47	–2	–1	5	13	24	42	54	44
W	–3	3	13	22	42	73	75	27	2	0	2	7	15	30	52	61	–1	–1	5	13	23	46	69	61
NW	–3	3	13	22	37	62	67	25	1	0	2	7	15	27	45	54	–1	–1	5	13	22	40	60	55
	High Mass, Low R-Value Wall								High Mass, Medium R-Value Wall								High Mass, High R-Value Wall							
N	3	3	7	12	16	21	25	27	10	8	8	10	12	15	18	21	12	9	8	8	10	13	16	19
NE	3	6	20	31	33	32	31	27	11	9	14	21	25	26	27	26	13	10	10	15	21	24	27	27
E	4	6	22	36	39	36	33	29	12	10	15	24	29	30	30	29	14	11	11	17	24	28	30	31
SE	3	4	14	25	30	30	30	26	10	8	11	17	21	24	25	25	13	10	9	12	17	21	24	25
S	3	1	3	7	14	20	23	22	8	6	5	6	10	14	17	18	10	8	6	5	7	10	14	17
SW	5	3	4	8	14	26	38	40	13	10	9	9	11	17	24	30	17	13	10	8	9	12	18	25
W	7	4	4	8	15	28	45	51	17	13	11	11	13	18	28	36	21	16	12	10	11	13	20	30
NW	6	3	4	8	14	25	40	46	15	12	10	10	12	17	25	32	19	14	11	9	10	12	18	26

CAUTION: Approximate data—Use for preliminary computations only.

Table 11.12 Approximate CLTDs for Sunlit Walls—36°N Latitude, July

Wall Facing	Solar time, h								Solar time, h								Solar time, h							
	6	8	10	12	14	16	18	20	6	8	10	12	14	16	18	20	6	8	10	12	14	16	18	20
	Low Mass, Low R-Value Wall								Low Mass, Medium R-Value Wall								Low Mass, High R-Value Wall							
N	–1	12	14	21	28	29	30	17	0	0	5	10	16	22	26	27	–2	3	9	15	21	27	28	27
NE	1	41	46	30	29	29	24	14	0	4	21	33	33	31	30	27	–2	12	36	39	32	30	28	23
E	1	49	64	48	31	30	24	14	1	4	26	45	47	40	34	29	–2	14	46	56	45	34	30	23
SE	–1	31	52	52	36	30	24	14	1	2	16	34	44	41	35	29	–2	7	31	48	47	37	31	23
S	–3	4	18	39	47	40	25	14	0	–1	2	11	25	36	38	32	–2	–1	6	21	37	44	37	25
SW	–2	4	13	23	50	67	59	23	1	0	2	8	17	34	51	54	–1	–1	5	13	28	50	62	51
W	–2	4	13	21	42	73	78	31	2	0	2	8	15	30	52	63	–1	–1	5	13	23	46	69	65
NW	–2	4	13	21	29	53	65	28	1	0	2	8	15	24	39	51	–2	–1	5	13	21	33	53	55
	High Mass, Low R-Value Wall								High Mass, Medium R-Value Wall								High Mass, High R-Value Wall							
N	3	3	6	10	15	20	23	25	9	7	8	9	11	14	17	19	11	9	7	7	9	11	14	17
NE	3	7	20	28	29	29	29	26	10	9	14	20	23	24	25	25	13	10	10	15	19	22	24	25
E	4	8	25	38	40	37	34	29	12	11	17	25	30	31	31	30	15	11	12	18	25	30	31	31
SE	4	5	17	30	37	36	33	29	12	10	13	20	26	29	29	28	14	11	10	14	20	26	29	30
S	3	2	4	11	22	31	33	29	10	8	7	9	14	20	24	25	13	10	7	7	10	15	21	24
SW	6	3	4	8	16	31	44	46	15	12	10	10	13	19	28	34	19	15	11	10	10	14	21	29
W	7	4	5	9	15	28	46	54	17	14	12	11	13	18	28	37	22	17	13	11	11	14	20	30
NW	6	3	4	8	14	22	35	43	14	11	10	10	12	15	22	30	18	14	11	9	10	12	17	24

CAUTION: Approximate data—Use for preliminary computations only.

Table 11.13 Approximate CLTDs for Sunlit Walls—48°N Latitude, July

CAUTION: Approximate data—Use for preliminary computations only.

Wall Facing	Solar time, h								Solar time, h								Solar time, h							
	6	8	10	12	14	16	18	20	6	8	10	12	14	16	18	20	6	8	10	12	14	16	18	20
	Low Mass, Low R-Value Wall								**Low Mass, Medium R-Value Wall**								**Low Mass, High R-Value Wall**							
N	3	10	13	21	27	28	27	21	1	2	6	10	16	21	25	26	–1	5	9	14	21	26	27	27
NE	10	42	38	26	28	29	24	15	1	7	23	31	30	29	28	26	0	18	36	34	28	28	28	23
E	10	54	64	47	31	29	25	15	1	8	30	47	48	40	34	29	0	20	49	57	44	34	29	23
SE	4	36	59	61	45	31	25	15	1	4	20	40	51	49	40	32	–1	11	36	55	56	43	33	24
S	–2	5	28	52	62	51	29	15	1	0	3	16	34	48	50	40	–1	0	9	30	50	57	47	30
SW	–1	5	12	29	59	75	65	29	2	0	3	8	20	40	58	61	–1	0	6	14	33	58	69	57
W	–1	5	13	21	41	72	80	41	2	0	3	8	15	29	51	64	–1	0	6	13	22	45	69	69
NW	–2	5	12	21	27	45	62	37	2	0	2	8	14	22	34	47	–1	0	5	13	20	29	46	54
	High Mass, Low R-Value Wall								**High Mass, Medium R-Value Wall**								**High Mass, High R-Value Wall**							
N	3	4	6	10	14	19	22	24	9	8	8	9	11	14	17	19	12	9	8	8	9	11	14	17
NE	4	10	22	26	26	27	27	25	10	10	15	20	22	23	24	24	13	10	12	16	19	22	23	24
E	4	11	28	40	40	37	34	29	12	12	19	27	32	32	32	30	15	12	14	20	27	31	32	32
SE	4	7	20	35	43	42	38	32	13	12	15	23	30	34	34	32	16	12	12	17	24	30	34	34
S	5	3	6	16	31	41	43	37	13	10	9	12	19	27	32	33	16	12	10	10	14	21	28	32
SW	7	4	5	9	19	36	50	52	18	14	12	12	15	23	32	39	22	17	13	11	12	16	24	33
W	8	5	6	9	15	27	45	55	19	15	12	12	14	19	28	38	23	18	14	12	12	14	20	30
NW	6	4	5	8	14	20	31	41	14	11	10	10	12	15	20	28	18	14	11	9	10	12	16	22

Note 1. Apply data directly to (1) dark surface, (2) indoor temperature of 78°F, (3) outdoor maximum temperature of 95°F with mean temperature of 85°F and daily range of 21°F, (4) outside surface film resistance of 0.333 (h·ft^2·°F)/Btu, and (5) inside surface resistance of 0.685 (h·ft^2·°F)/Btu.

Note 2. Adjustments to design temperatures:

$$\text{Corr. CLTD} = \text{CLTD} + (78 - t_r) + (t_m - 85)$$

where t_r = inside temperature and t_m = mean outdoor temperature, or t_m = maximum outdoor temperature – (daily range)/2

Note 3. Adjustments to months other than July: For design purposes, the data suffice for plus or minus 2 weeks from the 21st day of given month.

Table 11.14 Solar Cooling Load for Sunlit Glass (SCL)

Tables do not consider zone type and are conservative. Use for preliminary computations only.

Glass Facing	Solar time, h																	
	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
24°N Latitude, July																		
N	0	19	35	36	36	38	40	42	42	40	38	39	43	32	11	6	3	1
NE	0	54	124	150	144	115	78	58	49	44	38	32	25	14	6	3	1	1
E	0	57	139	177	180	154	107	68	54	46	40	33	25	14	6	3	1	1
SE	0	26	74	104	114	106	83	59	50	44	38	32	25	14	6	3	1	1
S	0	5	15	23	30	35	40	43	43	40	37	32	24	14	6	3	1	1
SW	0	5	15	23	30	35	39	42	61	88	110	118	105	62	24	12	6	3
W	0	5	15	23	30	35	39	41	67	116	160	186	184	118	44	21	11	5
NW	0	5	15	23	30	35	39	41	51	83	122	151	158	106	39	19	9	5
Hor	0	10	55	113	170	218	253	271	273	258	225	176	115	54	24	12	6	3
36°N Latitude, July																		
N	0	25	29	28	32	36	39	40	41	39	36	32	33	36	12	6	3	1
NE	0	79	129	139	120	84	58	50	45	41	37	32	26	17	7	3	2	1
E	0	86	153	184	182	155	107	67	54	45	39	33	26	17	7	3	2	1
SE	0	42	90	125	142	140	119	86	58	48	40	34	27	17	7	3	2	1
S	0	8	17	24	36	53	70	80	79	68	52	38	29	18	7	3	2	1
SW	0	8	17	24	30	35	38	57	90	122	141	144	127	85	32	15	8	4
W	0	8	17	24	30	35	38	40	66	115	159	188	191	149	53	25	12	6
NW	0	8	17	24	30	35	38	40	40	56	93	129	148	127	43	21	10	5
Hor	0	20	66	120	171	215	246	263	265	251	221	178	124	66	28	13	7	3
48°N Latitude, July																		
N	14	28	24	27	31	34	37	38	38	37	35	31	27	34	25	9	4	2
NE	32	101	130	126	95	61	49	44	41	38	35	31	26	19	10	4	2	1
E	31	112	165	188	182	153	104	65	51	43	38	32	27	19	10	4	2	1
SE	11	58	106	143	164	168	152	119	77	54	43	35	28	20	10	4	2	1
S	3	11	18	30	58	90	116	130	130	116	88	56	37	24	12	5	3	1
SW	3	11	18	24	30	34	46	82	122	152	168	166	146	106	50	22	11	5
W	3	11	18	24	30	34	36	38	64	112	156	186	193	167	89	36	17	9
NW	3	11	18	24	30	34	36	38	38	40	67	106	134	134	76	30	14	7
Hor	5	32	73	120	163	200	226	241	242	230	205	170	125	76	35	16	8	4

Data apply directly to: (1) standard double strength glass with no inside shade, and (2) clear sky, 21st day of month.

Adjustments to table data:

- Latitudes other than 24, 36 and 48°N
  Linear interpolation is acceptable.
- Months other than July
  For design purposes, data will suffice for plus or minus 2 weeks from the 21st day of given month.
- Other types of glass and internal shade
  Use shading coefficients as multiplier.
- Externally shaded glass
  Use north orientation.

Table 11.15 Shading Coefficients* for Single Glass with Indoor Shading by Venetian Blinds or Roller Shades

Type of Glass	Nominal Thickness,[a] in.	Solar Transmit-tance[b]	Type of Shading: Venetian Blinds		Roller Shade: Opaque		Roller Shade: Translucent
			Medium	Light	Dark	White	Light
Clear	3/32[c]	0.87 to 0.80	0.74[d]	0.67[d]	0.81	0.39	0.44
			(0.63)[e]	(0.58)[e]			
Clear	1/4 to 1/2	0.80 to 0.71					
Clear pattern	1/8 to 1/2	0.87 to 0.79					
Heat-absorbing pattern	1/8	—					
Tinted	3/16, 7/32	0.74, 0.71					
Heat-absorbing[f]	3/16, 1/4	0.46					
Heat-absorbing pattern	3/16, 1/4	—	0.57	0.53	0.45	0.30	0.36
Tinted	1/8, 7/32	0.59, 0.45					
Heat-absorbing or pattern	—	0.44 to 0.30	0.54	0.52	0.40	0.28	0.32
Heat-absorbing[f]	3/8	0.34					
Heat-absorbing or pattern		0.29 to 0.15					
	—	0.24	0.42	0.40	0.36	0.28	0.31
Reflective-coated	S.C. = 0.30[g]		0.25	0.23			
	= 0.40		0.33	0.29			
	= 0.50		0.42	0.38			
	= 0.60		0.50	0.44			

[a]Refer to manufacturers' literature for values.
[b]For vertical blinds with opaque white and beige louvers in the tightly closed position, SC is 0.25 and 0.29 when used with glass of 0.71 to 0.80 transmittance.
[c]Typical residential glass thickness.
[d]From Van Dyck and Konen (1982), for 45° open venetian blinds, 35° solar incidence, and 35° profile angle.
[e]Values for closed venetian blinds. Use these values only when operation is automated for solar gain reduction (as opposed to daylight use).
[f]Refers to gray, bronze, and green tinted heat-absorbing glass.
[g]SC for glass with no shading device.

* Note: Shading Coefficient (SC) has been superseded by solar heat gain coefficient (SHGC) including the effect of incident angle of solar radiation on the glass, and the effect of type of framing. This shading coefficient table is sufficiently accurate for the approximate cooling load calculations of this publication. For the glazing portion of single-pane clear and tinted fenestration, SC = SHGC/0.87. This does not include frame effects.

Table 11.16 Representative Rates at Which Heat and Moisture are Given Off by Human Beings in Different States of Activity [2017F, Ch 18, Tbl 1]

Degree of Activity	Location	Total Heat, Btu/h		Sensible Heat, Btu/h	Latent Heat, Btu/h	% Sensible Heat that is Radiant[b]	
		Adult Male	Adjusted, M/F[a]			Low V	High V
Seated at theater	Theater, matinee	390	330	225	105		
Seated at theater, night	Theater, night	390	350	245	105	60	27
Seated, very light work	Offices, hotels, apartments	450	400	245	155		
Moderately active office work	Offices, hotels, apartments	475	450	250	200		
Standing, light work; walking	Department store; retail store	550	450	250	200	58	38
Walking, standing	Drug store, bank	550	500	250	250		
Sedentary work	Restaurant[c]	490	550	275	275		
Light bench work	Factory	800	750	275	475		
Moderate dancing	Dance hall	900	850	305	545	49	35
Walking 3 mph; light machine work	Factory	1000	1000	375	625		
Bowling[d]	Bowling alley	1500	1450	580	870		
Heavy work	Factory	1500	1450	580	870	54	19
Heavy machine work; lifting	Factory	1600	1600	635	965		
Athletics	Gymnasium	2000	1800	710	1090		

Notes:
1. Tabulated values are based on 75°F room dry-bulb temperature. For 80°F room dry bulb, total heat remains the same, but sensible heat values should be decreased by approximately 20%, and latent heat values increased accordingly.
2. Also see 2017F, Ch 9, Tbl 4 for additional rates of metabolic heat generation.
3. All values are rounded to nearest 5 Btu/h.

[a] Adjusted heat gain is based on normal percentage of men, women, and children for the application listed, and assumes that gain from an adult female is 85% of that for an adult male, and gain from a child is 75% of that for an adult male.
[b] Values approximated from data in 2017F, Ch 9, Tbl 6, where V is air velocity with limits shown in that table.
[c] Adjusted heat gain includes 60 Btu/h for food per individual (30 Btu/h sensible and 30 Btu/h latent).
[d] Figure one person per alley actually bowling, and all others as sitting (400 Btu/h) or standing or walking slowly (550 Btu/h).

Heat Gain from Lighting

The energy absorbed by the structure and contents contributes to space cooling load only after a time lag, some still reradiating after the heat sources have been switched off. This may make load lower than instantaneous heat gain, thus affecting the peak load.

Instantaneous rate of heat gain from lights, q_{el} Btu/h:

$$q_{el} = 3.41\ WF_{ul}F_{sa} \tag{11.6}$$

where

W	=	total lights wattage installed
F_{ul}	=	lighting use factor (proportion in use)
F_{sa}	=	lighting special allowance factor
3.41	=	conversion factor

The **total light wattage** is obtained from the ratings of all lamps installed, both for general illumination and for display use. Ballasts are not included, but are addressed by a separate factor. Wattages of magnetic ballasts are significant; the energy consumption of high-efficiency electronic ballasts might be insignificant compared to that of the lamps.

The **lighting use factor** is the ratio of wattage in use, for the conditions under which the load estimate is being made, to total installed wattage. For commercial applications such as stores, the use factor is generally 1.0.

The **special allowance factor** is the ratio of the lighting fixtures' power consumption, including lamps and ballast, to the nominal power consumption of the lamps. For incandescent lights, this factor is 1. For fluorescent lights, it accounts for power consumed by the ballast as well as the ballast's effect on lamp power consumption. The special allowance factor can be less than 1 for electronic ballasts that lower electricity consumption below the lamp's rated power consumption. Use manufacturers' values for system (lamps + ballast) power, when available.

For high-intensity-discharge lamps (e.g. metal halide, mercury vapor, high- and low-pressure sodium vapor lamps), the actual lighting system power consumption should be available from the manufacturer of the fixture or ballast. Ballasts available for metal halide and high pressure sodium vapor lamps may have special allowance factors from about 1.3 (for low-wattage lamps) down to 1.1 (for high-wattage lamps).

An alternative procedure is to estimate the lighting heat gain on a per square foot basis. Such an approach may be required when final lighting plans are not available. Table 11.17 shows the maximum lighting power density (LPD) (lighting heat gain per square foot) allowed by ASHRAE Standard 90.1-2010 for a range of space types.

Table 11.17 Lighting Power Densities Using Space-by-Space Method
[2017F, Ch 18, Tbl 2]

Common Space Types[a]	LPD, W/ft^2	Common Space Types[a]	LPD, W/ft^2
Atrium		**Electrical/Mechanical Room**[f]	0.42
≤40 ft high	0.03/ft total height	**Emergency Vehicle Garage**	0.56
		Food Preparation Area	1.21
>40 ft high	0.40 + 0.02/ft total height	**Guest Room**	0.91
		Laboratory	
Audience Seating Area		In or as classroom	1.43
In auditorium	0.63	All other laboratories	1.81
In convention center	0.82	**Laundry/Washing Area**	0.60
In gymnasium	0.65	**Loading Dock, Interior**	0.47
In motion picture theater	1.14	**Lobby**	
In penitentiary	0.28	In facility for the visually impaired (and not used primarily by staff)[c]	1.80
In performing arts theater	2.43		
In religious building	1.53		
In sports arena	0.43	For elevator	0.64
All other audience seating areas	0.43	In hotel	1.06
		In motion picture theater	0.59
Banking Activity Area	1.01	In performing arts theater	2.00
Breakroom (*See* Lounge/ Breakroom)		All other lobbies	0.90
		Locker Room	0.75
Classroom/Lecture Hall/ Training Room		**Lounge/Breakroom**	
		In health care facility	0.92
In penitentiary	1.34	All other lounges/breakrooms	0.73
All other classrooms/lecture halls/training rooms	1.24	Enclosed and ≤250 ft^2	1.11
		Enclosed and >250 ft^2	1.11
Conference/Meeting/ Multipurpose Room	1.23	Open plan	0.98
		Office	
Confinement Cells	0.81	Enclosed	1.11
Copy/Print Room	0.72	Open plan	0.98
Corridor[b]		**Parking Area, Interior**	0.19
In facility for visually impaired (and not used primarily by staff)[c]	0.92	**Pharmacy Area**	1.68
		Restroom	
		In facility for the visually impaired (and not used primarily by staff)[c]	1.21
In hospital	0.99		
In manufacturing facility	0.41		
All other corridors	0.66	All other restrooms	0.98
Courtroom	1.72	**Sales Area**[d]	1.44
Computer Room	1.71	**Seating Area, General**	0.54
Dining Area		**Stairway**	
In penitentiary	0.96	Space containing stairway determines LPD and control requirements for stairway.	
In facility for visually impaired (and not used primarily by staff)[c]	2.65		
		Stairwell	0.69
In bar/lounge or leisure dining	1.07	**Storage Room**	
		<50 ft^2	1.24
In cafeteria or fast food dining	0.65	All other storage rooms	0.63
		Vehicular Maintenance Area	0.67
In family dining	0.89		
All other dining areas	0.65		

Table 11.17 Lighting Power Densities Using Space-by-Space Method
[2017F, Ch 18, Tbl 2] (***Continued***)

Building-Specific Space Types*	LPD, W/ft^2
Facility for Visually Impaired[c]	
Chapel (used primarily by residents)	2.21
Recreation room/common living room (and not used primarily by staff)	2.41
Automotive (*See* Vehicular Maintenance Area)	
Convention Center, Exhibit Space	1.45
Dormitory/Living Quarters	0.38
Fire Station, Sleeping Quarters	0.22
Gymnasium/Fitness Center	
Exercise area	0.72
Playing area	1.20
Health Care Facility	
Exam/treatment room	1.66
Imaging room	1.51
Medical supply room	0.74
Nursery	0.88
Nurses' station	0.71
Operating room	2.48
Patient room	0.62
Physical therapy room	0.91
Recovery room	1.15
Library	
Reading area	1.06
Stacks	1.71
Manufacturing Facility	
Detailed manufacturing area	1.29
Equipment room	0.74
Extra-high-bay area (>50 ft floor-to-ceiling height)	1.05
High-bay area (25 to 50 ft floor-to-ceiling height)	1.23
Low bay area (<25 ft floor-to-ceiling height)	1.19
Museum	
General exhibition area	1.05
Restoration room	1.02
Performing Arts Theater, Dressing Room	0.61
Post Office, Sorting Area	0.94
Religious Buildings	
Fellowship hall	0.64
Worship/pulpit/choir area	1.53
Retail Facilities	
Dressing/fitting room	0.71
Mall concourse	1.10
Sports Arena, Playing Area	
For Class I facility	3.68
For Class II facility	2.40
For Class III facility	1.80
For Class IV facility	1.20
Transportation Facility	
In baggage/carousel area	0.53
In airport concourse	0.36
At terminal ticket counter	0.80
Warehouse—Storage Area	
For medium to bulky, palletized items	0.58
For smaller, hand-carried items[e]	0.95

Source: ASHRAE Standard 90.1-2013.
[a]In cases where both a common space type and a building-specific type are listed, the building-specific space type applies.
[b]In corridors, extra lighting power density allowance is granted when corridor width is <8 ft and is not based on room/corridor ratio (RCR).
[c]A facility for the visually impaired one that can be documented as being designed to comply with light levels in ANSI/IES RP-28 and is (or will be) licensed by local/state authorities for either senior long-term care, adult day-care, senior support, and/or people with special visual needs.
[d]For accent lighting, see section 9.6.2(b) of ASHRAE Standard 90.1-2013.
[e]Sometimes called a picking area.
[f]An additional 0.53 W/ft^2 is allowed *only* if this additional lighting is controlled separately from the base allowance of 0.42 W/ft^2.

Table 11.18 provides a range of design data under typical operating conditions: airflow 1 cfm/ft^2, supply air between 59°F and 62°F, room temperature between 72°F and 75°F, and lighting heat input in a range from 0.9 to 2.6 W/ft^2. For a fluorescent luminaire without lens, Figure 11.1 gives more precise data. The data should be used with judgment.

Table 11.18 Lighting Heat Gain Parameters for Typical Operating Conditions
[2017F, Ch 18, Tbl 3]

Luminaire Category	Space Fraction	Radiative Fraction	Notes
Recessed fluorescent luminaire without lens	0.64 to 0.74	0.48 to 0.68	• Use middle values in most situations • May use higher space fraction, and lower radiative fraction for luminaire with side-slot returns • May use lower values of both fractions for direct/indirect luminaire • May use higher values of both fractions for ducted returns
Recessed fluorescent luminaire with lens	0.40 to 0.50	0.61 to 0.73	• May adjust values in the same way as for recessed fluorescent luminaire without lens
Downlight compact fluorescent luminaire	0.12 to 0.24	0.95 to 1.0	• Use middle or high values if detailed features are unknown • Use low value for space fraction and high value for radiative fraction if there are large holes in luminaire's reflector
Downlight incandescent luminaire	0.70 to 0.80	0.95 to 1.0	• Use middle values if lamp type is unknown • Use low value for space fraction if standard lamp (i.e. A-lamp) is used • Use high value for space fraction if reflector lamp (i.e. BR-lamp) is used
Non-in-ceiling fluorescent luminaire	1.0	0.5 to 0.57	• Use lower value for radiative fraction for surface-mounted luminaire • Use higher value for radiative fraction for pendant luminaire

Source: Fisher and Chantrasrisalai (2006).

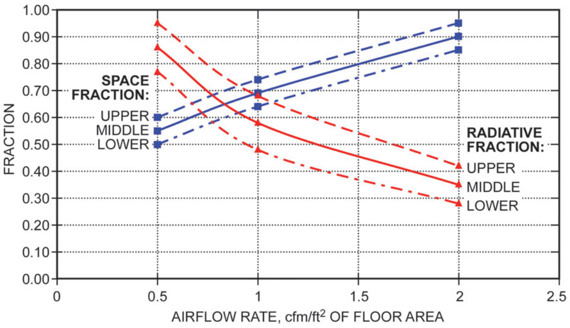


Figure 11.1 Lighting Heat Gain Parameters for Recessed Fluorescent Luminaire Without Lens
[2017F, Ch 18, Fig 3]

Heat Gain from Motors and their Loads

Instantaneous rate of heat gain from equipment operated by electric motors within a conditioned space.

$$q_{em} = 2545\ (P/E_M)\ F_{UM}\ F_{LM} \tag{11.7}$$

where

q_{em} = heat equivalent of equipment operation, Btu/h
P = motor power rating, hp
E_M = motor efficiency, decimal fraction < 1.0
F_{UM} = motor use factor 1.0 or <1.0 (proportion operating)
F_{LM} = motor load factor 1.0 or <1.0

When motor is outside the conditioned space, but load is inside,

$$q_{em} = 2545\ P\ F_{UM}\ F_{LM} \tag{11.8}$$

When motor is inside the conditioned space, but load is outside,

$$q_{em} = 2545\ P \left(\frac{1.0 - E_M}{E_M}\right) F_{UM}\ F_{LM} \tag{11.9}$$

Heat output of a motor is generally proportional to motor load, within rated overload limits. Because of typically high no-load motor current, fixed losses, and other reasons, F_{LM} is generally assumed to be unity, and no adjustment should be made for underloading or overloading unless the situation is fixed and can be accurately established, and reduced-load efficiency data can be obtained from the motor manufacturer.

Unless the manufacturer's technical literature indicates otherwise, motor heat gain normally should be equally divided between radiant and convective components for the subsequent cooling load calculations.

Table 11.19 Minimum Nominal Full-Load Efficiency for 60 HZ NEMA General Purpose Electric Motors (Subtype I) Rated 600 Volts or Less (Random Wound)*
[2017F, Ch 18, Tbl 4A]

Minimum Nominal Full Load Efficiency (%) for Motors Manufactured on or after December 19, 2010						
	Open Drip-Proof Motors			Totally Enclosed Fan-Cooled Motors		
Number of Poles ⇒	**2**	**4**	**6**	**2**	**4**	**6**
Synchronous Speed (RPM) ⇒	**3600**	**1800**	**1200**	**3600**	**1800**	**1200**
Motor Horsepower						
1	77.0	85.5	82.5	77.0	85.5	82.5
1.5	84.0	86.5	86.5	84.0	86.5	87.5
2	85.5	86.5	87.5	85.5	86.5	88.5
3	85.5	89.5	88.5	86.5	89.5	89.5
5	86.5	89.5	89.5	88.5	89.5	89.5
7.5	88.5	91.0	90.2	89.5	91.7	91.0
10	89.5	91.7	91.7	90.2	91.7	91.0
15	90.2	93.0	91.7	91.0	92.4	91.7
20	91.0	93.0	92.4	91.0	93.0	91.7
25	91.7	93.6	93.0	91.7	93.6	93.0
30	91.7	94.1	93.6	91.7	93.6	93.0
40	92.4	94.1	94.1	92.4	94.1	94.1
50	93.0	94.5	94.1	93.0	94.5	94.1
60	93.6	95.0	94.5	93.6	95.0	94.5
75	93.6	95.0	94.5	93.6	95.4	94.5
100	93.6	95.4	95.0	94.1	95.4	95.0
125	94.1	95.4	95.0	95.0	95.4	95.0
150	94.1	95.8	95.4	95.0	95.8	95.8
200	95.0	95.8	95.4	95.4	96.2	95.8
250	95.0	95.8	95.4	95.8	96.2	95.8
300	95.4	95.8	95.4	95.8	96.2	95.8
350	95.4	95.8	95.4	95.8	96.2	95.8
400	95.8	95.8	95.8	95.8	96.2	95.8
450	95.8	96.2	96.2	95.8	96.2	95.8
500	95.8	96.2	96.2	95.8	96.2	95.8

Source: ASHRAE/IES Standard 90.1-2013

*Nominal efficiencies established in accordance with NEMA Standard MG1. Design A and Design B are National Electric Manufacturers Association (NEMA) design class designations for fixed-frequency small and medium AC squirrel-cage induction motors.

Cooking Appliances

Heat gain: $q_s = q_{input} F_U F_R$, where F_U is the usage factor and F_R is the radiation factor.

Table 11.20 Recommended Rates of Radiant and Convective Heat Gain from Unhooded Electric Appliances During Idle (Ready-to-Cook) Conditions [2017F, Ch 18, Tbl 5A]

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h					
Appliance	**Rated**	**Standby**	**Sensible Radiant**	**Sensible Convective**	**Latent**	**Total**	**Usage Factor F_U**	**Radiation Factor F_R**
Cabinet: hot serving (large), insulated[a]	6,800	1,200	400	800	0	1,200	0.18	0.33
hot serving (large), uninsulated	6,800	3,500	700	2,800	0	3,500	0.51	0.20
proofing (large)[a]	17,400	1,400	1,200	0	200	1,400	0.08	0.86
proofing (small 15-shelf)	14,300	3,900	0	900	3,000	3,900	0.27	0.00
Cheesemelter[b]	8,200	3,300	1,500	1,800	0	3,300	0.41	0.45
Coffee brewing urn	13,000	1,200	200	300	700	1,200	0.09	0.17
Drawer warmers, 2-drawer (moist holding)[a]	4,100	500	0	0	200	200	0.12	0.00
Egg cooker[b]	8,100	850	200	650	0	850	0.10	0.26
Espresso machine*	8,200	1,200	400	800	0	1,200	0.15	0.33
Food warmer: steam table (2-well-type)	5,100	3,500	300	600	2,600	3,500	0.69	0.09
Freezer (small)	2,700	1,100	500	600	0	1,100	0.41	0.45
Fryer, countertop, open deep fat[b]	15,700	1,500	700	800	0	1,500	0.09	0.47
Griddle, countertop[b]	27,300	6,100	2,900	3,200	0	6,100	0.22	0.48
Hot dog roller[b]	5,500	4,200	900	3,300	0	4,200	0.77	0.22
Hot plate: single element, high speed	3,800	3,400	1,100	2,300	0	3,400	0.89	0.32
Hot-food case (dry holding)[a]	31,100	2,500	900	1,600	0	2,500	0.08	0.36
Hot-food case (moist holding)[a]	31,100	3,300	900	1,800	600	3,300	0.11	0.27
Induction hob, countertop[b]	17,100	0	0	0	0	0	0.00	0.00
Microwave oven: commercial	5,800	0	0	0	0	0	0.00	0.00
Oven: countertop conveyorized bake/finishing[b]	17,100	13,500	2,500	11,000	0	13,500	0.79	0.18
Panini[b]	6,100	2,300	700	1,600	0	2,300	0.37	0.29
Popcorn popper[b]	2,900	400	100	300	0	400	0.14	0.24
Rapid-cook oven (quartz-halogen)[a]	41,000	0	0	0	0	0	0.00	0.00
Rapid-cook oven (microwave/convection)[b]	19,400	3,900	300	3,600	0	3,900	0.20	0.08
Reach-in refrigerator[a]	4,800	1,200	300	900	0	1,200	0.25	0.25
Refrigerated prep table[a]	2,000	900	600	300	0	900	0.45	0.67
Rice cooker[b]	5,300	300	50	250	0	300	0.05	0.17
Soup warmer[b]	2,700	1,300	0	200	1,100	1,300	0.49	0.00
Steamer (bun)[b]	5,100	700	100	600	0	700	0.13	0.16
Steamer, countertop[b]	28,300	1,200	0	800	400	1,200	0.04	0.00
Toaster: 4-slice pop up (large): cooking	6,100	3,000	200	1,400	1,000	2,600	0.49	0.07
contact (vertical)[b]	8,900	2,600	600	2,000	0	2,600	0.29	0.24
conveyor (large)	32,800	10,300	3,000	7,300	0	10,300	0.31	0.29
small conveyor[b]	6,000	5,800	1,200	4,600	0	5,800	0.98	0.21
Tortilla grill[b]	7,500	3,600	900	2,700	0	3,600	0.47	0.25
Waffle iron[b]	9,200	900	200	700	0	900	0.10	0.22

Sources: Swierczyna et al. (2008, 2009); with the following exceptions as noted.
[a]Swierczyna et al. (2009) only.
[b]Additions and updates from ASHRAE research project RP-1631 (Kong and Zhang 2016; Kong et al 2016).

Table 11.21 Recommended Rates of Radiant and Convective Heat Gain from Unhooded Electric Appliances during Cooking Conditions [2017F, Ch 18, Tbl 5B]

Appliance	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h				Usage Factor F_U	Radiation Factor F_R
	Rated	Cooking	Sensible Radiant	Sensible Convective	Latent	Total		
Cheesemelter	8,200	9,300	1,500	3,700	2,000	7,200	1.13	0.16
Egg cooker	8,100	4,100	200	1,300	2,200	3,700	0.50	0.05
Fryer, countertop, open deep fryer	15,700	13,000	700	1,700	5,600	8,000	0.83	0.05
Griddle, countertop	27,300	11,200	2,900	2,200	4,400	9,500	0.41	0.26
Hot dog roller	5,500	5,400	900	2,100	2,300	5,300	0.99	0.17
Hot plate, single burner	3,800	3,400	1,100	2,100	200	3,400	0.90	0.32
Induction hob, countertop	17,100	2,200	0	1,100	1,100	2,200	0.13	0.00
Oven, conveyor	17,100	14,600	2,500	8,400	700	11,600	0.86	0.17
Microwave	5,800	8,100	0	3,200	3,400	6,600	1.39	0.00
Rapid cook	19,400	7,900	300	4,200	2,600	7,100	0.41	0.04
Panini grill	6,100	4,700	700	2,400	500	3,600	0.76	0.14
Popcorn popper	2,900	2,000	100	800	700	1,600	0.68	0.05
Rice cooker	5,300	4,000	50	300	200	550	0.75	0.01
Soup warmer	2,700	2,900	0	300	2,400	2,700	1.05	0.00
Steamer (bun)	5,100	2,700	100	800	1,700	2,600	0.53	0.04
Steamer, countertop	28,300	26,400	0	1,700	23,700	25,400	0.93	0.00
Toaster, conveyor	6,000	5,800	1,200	3,300	1,300	5,800	0.98	0.21
Vertical	8,900	6,300	600	2,400	1,100	4,100	0.71	0.10
Tortilla grill	7,500	7,500	900	4,300	2,300	7,500	1.00	0.12
Waffle maker	9,200	4,000	200	1,200	1,900	3,300	0.44	0.05

Source: ASHRAE research project RP-1631 (Zhang et al. 2015).

Table 11.22 Recommended Rates of Radiant Heat Gain from Hooded Electric Appliances During Idle (Ready-to-Cook) Conditions [2017F, Ch 18, Tbl 5C]

Appliance	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h	Usage Factor F_U	Radiation Factor F_R
	Rated	Standby	Sensible Radiant		
Broiler: underfired 3 ft	36,900	30,900	10,800	0.84	0.35
Cheesemelter*	12,300	11,900	4,600	0.97	0.39
Fryer, kettle	99,000	1,800	500	0.02	0.28
Open deep-fat, 1-vat	47,800	2,800	1,000	0.06	0.36
Pressure	46,100	2,700	500	0.06	0.19
Griddle, double-sided 3 ft (clamshell down)*	72,400	6,900	1,400	0.10	0.20
(Clamshell up)*	72,400	11,500	3,600	0.16	0.31
Flat 3 ft	58,400	11,500	4,500	0.20	0.39
Small 3 ft*	30,700	6,100	2,700	0.20	0.44
Induction cooktop*	71,700	0	0	0.00	0.00
Induction wok*	11,900	0	0	0.00	0.00
Oven, combi: combi-mode*	56,000	5,500	800	0.10	0.15
Convection mode	56,000	5,500	1,400	0.10	0.25
Oven, convection, full-sized	41,300	6,700	1,500	0.16	0.22
Half-sized*	18,800	3,700	500	0.20	0.14
Pasta cooker*	75,100	8,500	0	0.11	0.00
Range top, top off/oven on*	16,600	4,000	1,000	0.24	0.25
3 elements on/oven off	51,200	15,400	6,300	0.30	0.41
6 elements on/oven off	51,200	33,200	13,900	0.65	0.42
6 elements on/oven on	67,800	36,400	14,500	0.54	0.40
Range, hot-top	54,000	51,300	11,800	0.95	0.23
Rotisserie*	37,900	13,800	4,500	0.36	0.33
Salamander*	23,900	23,300	7,000	0.97	0.30
Steam kettle, large (60 gal) simmer lid down*	110,600	2,600	100	0.02	0.04
Small (40 gal) simmer lid down*	73,700	1,800	300	0.02	0.17
Steamer, compartment, atmospheric*	33,400	15,300	200	0.46	0.01
Tilting skillet/braising pan	32,900	5,300	0	0.16	0.00

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 11.23 Recommended Rates of Radiant Heat Gain from Hooded Gas Appliances during Idle (Ready-to-Cook) Conditions [2017F, Ch 18, Tbl 5D]

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h		
Appliance	**Rated**	**Standby**	**Sensible Radiant**	**Usage Factor F_U**	**Radiation Factor F_R**
Broiler: batch*	95,000	69,200	8,100	0.73	0.12
Chain (conveyor)	132,000	96,700	13,200	0.73	0.14
Overfired (upright)*	100,000	87,900	2,500	0.88	0.03
Underfired 3 ft	96,000	73,900	9,000	0.77	0.12
Fryer: doughnut	44,000	12,400	2,900	0.28	0.23
Open deep-fat, 1 vat	80,000	4,700	1,100	0.06	0.23
Pressure	80,000	9,000	800	0.11	0.09
Griddle: double sided 3 ft, clamshell down*	108,200	8,000	1,800	0.07	0.23
Clamshell up*	108,200	14,700	4,900	0.14	0.33
Flat 3 ft	90,000	20,400	3,700	0.23	0.18
Oven: combi: combi-mode*	75,700	6,000	400	0.08	0.07
Convection mode	75,700	5,800	1,000	0.08	0.17
Convection, full-size	44,000	11,900	1,000	0.27	0.08
Conveyor (pizza)	170,000	68,300	7,800	0.40	0.11
Deck	105,000	20,500	3,500	0.20	0.17
Rack mini-rotating*	56,300	4,500	1,100	0.08	0.24
Pasta cooker*	80,000	23,700	0	0.30	0.00
Range top: top off/oven on*	25,000	7,400	2,000	0.30	0.27
3 burners on/oven off	120,000	60,100	7,100	0.50	0.12
6 burners on/oven off	120,000	120,800	11,500	1.01	0.10
6 burners on/oven on	145,000	122,900	13,600	0.85	0.11
Range: wok*	99,000	87,400	5,200	0.88	0.06
Rethermalizer*	90,000	23,300	11,500	0.26	0.49
Rice cooker*	35,000	500	300	0.01	0.60
Salamander*	35,000	33,300	5,300	0.95	0.16
Steam kettle: large (60 gal) simmer lid down*	145,000	5,400	0	0.04	0.00
Small (10 gal) simmer lid down*	52,000	3,300	300	0.06	0.09
Medium (40 gal) simmer lid down	100,000	4,300	0	0.04	0.00
Steamer: compartment: atmospheric*	26,000	8,300	0	0.32	0.00
Tilting skillet/braising pan	104,000	10,400	400	0.10	0.04

*Items with an asterisk appear only in Swierczyna et al. (2009); all others appear in both Swierczyna et al. (2008) and (2009).

Table 11.24 Recommended Rates of Radiant Heat Gain from Hooded Solid-Fuel Appliances during Idle (Ready-to-Cook) Conditions [2017F, Ch 18, Tbl 5E]

Appliance	**Rated**	**Standby Energy Rate, Btu/h**	**Rate of Sensible Heat Gain, Btu/h**	**Usage Factor F_U**	**Radiation Factor F_R**
Broiler: solid fuel: charcoal	40 lb	42,000	6200	N/A	0.15
Broiler: solid fuel: wood (mesquite)	40 lb	49,600	7000	N/A	0.14

Source: Swierczyna et al. (2008).

Table 11.25 Recommended Rates of Radiant and Convective Heat Gain from Warewashing Equipment during Idle (Standby) or Washing Conditions [2017F, Ch 18, Tbl 5F]

	Energy Rate, Btu/h		Rate of Heat Gain, Btu/h						
			Unhooded				Hooded		
Appliance	**Rated**	**Standby/ Washing**	**Sensible Radiant**	**Sensible Convective**	**Latent**	**Total**	**Sensible Radiant**	**Usage Factor F_U**	**Radiation Factor F_R**
Dishwasher: conveyor type, hot-water sanitizing, washing	46,800	N/A	0	12,100	47,000	59,100	0	N/A	0.00
Standby	46,800	5,700	0	1,600	4,100	5,700	0	0.12	0.00
Dishwasher: conveyor type, chemical sanitizing, washing	46,800	43,600	0	11,100	35,400	46,500	0	0.93	0.00
Standby	46,800	5,700	0	1,600	4,100	5,700	0	0.12	0.00
Dishwasher: door type, hot-water sanitizing, washing	60,100	18,500	0	7,600	25,200	32,800	0	0.31	0.00
With heat recovery and vapor reduction	51,900	27,100	0	5,800	13,100	18,900	0	0.52	0.00
Standby	18,400	1,200	0	2,280	4,170	6,450	0	0.35	0.00
Dishwasher: door type, chemical sanitizing, washing	30,000	15,600	0	3,900	13,200	17,100	0	0.52	0.00
Standby	18,400	1,200	0	900	300	1,200	0	0.07	0.00
Dishwasher: door type, chemical sanitizing, dump and fill, washing	6,100	3,000	0	2,900	4,200	7,100	0	0.49	0.00
Standby	6,100	3,000	0	0	0	0	0	0.49	0.00
Pot and pan washer: door type, hot-water sanitizing, washing	53,200	36,400	0	6,000	23,500	29,500	0	0.68	0.00
With heat recovery and vapor reduction	53,200	35,200	0	5,500	19,000	24,500	0	0.66	0.00
Dishwasher: under-counter type, hot-water sanitizing, washing	28,500	7,600	800	3,200	6,900	10,900	800	0.27	0.11
With heat recovery and vapor reduction	26,600	22,800	0	2,000	1,100	3,100	0	0.86	0.00
Standby	26,600	1,700	800	500	400	1,700	800	0.06	0.47
Dishwasher: under-counter type, chemical sanitizing, washing	28,500	6,900	0	2,200	4,900	7,100	0	0.24	0.00
Standby	26,600	1,700	800	500	400	1,700	0	0.06	0.47
Booster heater	130,000	0	500	0	0	0	500	0	N/A

Sources: PG&E (2010-2016), Swierczyna et al. (2008, 2009).

Hospital and Laboratory Equipment

Heat gain varies significantly. In a laboratory, heat gain ranges from 15 to 70 Btu/h·ft^2. Medical equipment is highly varied in type and application. Table 11.26 is relevant for portable and bench-type equipment. For large equipment, such as MRI, obtain heat gain from the manufacturer.

Table 11.26 Recommended Heat Gain from Typical Medical Equipment
[2017F, Ch 18, Tbl 6]

Equipment	Nameplate, W	Peak, W	Average, W
Anesthesia system	250	177	166
Blanket warmer	500	504	221
Blood pressure meter	180	33	29
Blood warmer	360	204	114
ECG/RESP	1440	54	50
Electrosurgery	1000	147	109
Endoscope	1688	605	596
Harmonical scalpel	230	60	59
Hysteroscopic pump	180	35	34
Laser sonics	1200	256	229
Optical microscope	330	65	63
Pulse oximeter	72	21	20
Stress treadmill	N/A	198	173
Ultrasound system	1800	1063	1050
Vacuum suction	621	337	302
X-ray system	968		82
	1725	534	480
	2070		18

Source: Hosni et al. (1999)

Table 11.27 Recommended Heat Gain from Typical Laboratory Equipment
[2017F, Ch 18, Tbl 7]

Equipment	Nameplate, W	Peak, W	Average, W
Analytical balance	7	7	7
Centrifuge	138	89	87
	288	136	132
	5500	1176	730
Electrochemical analyzer	50	45	44
	100	85	84
Flame photometer	180	107	105
Fluorescent microscope	150	144	143
	200	205	178
Function generator	58	29	29
Incubator	515	461	451
	600	479	264
	3125	1335	1222
Orbital shaker	100	16	16
Oscilloscope	72	38	38
	345	99	97
Rotary evaporator	75	74	73
	94	29	28
Spectronics	36	31	31
Spectrophotometer	575	106	104
	200	122	121
	N/A	127	125
Spectro fluorometer	340	405	395
Thermocycler	1840	965	641
	N/A	233	198
Tissue culture	475	132	46
	2346	1178	1146

Source: Hosni et al. (1999).

Table 11.28 Recommended Heat Gain for Typical Desktop Computers
[2017F, Ch 18, Tbl 8A]

Description	Nameplate Power,[a] W	Peak Heat Gain,[b, d] W
Manufacturer 1		
3.0 GHz processor, 4 GB RAM, *n* = 1	NA	83
3.3 GHz processor, 8 GB RAM, *n* = 8	NA	50
3.5 GHz processor, 8 GB RAM, *n* = 2	NA	42
3.6 GHz processor, 16 GB RAM, *n* = 2	NA	66
3.3 GHz processor, 16 GB RAM, *n* = 2	NA	52
4.0 GHz processor, 16 GB RAM, *n* = 1	NA	83
3.3 GHz processor, 8 GB RAM, *n* = 1	NA	84
3.7 GHz processor, 32 GB RAM, *n* = 1	750	116
	NA	102
3.5 GHz processor, 16 GB RAM, *n* = 3[c]	550	144
	NA	93
Manufacturer 2		
3.6 GHz processor, 32 GB RAM, *n* = 8	NA	80
3.6 GHz processor, 16 GB RAM, *n* = 1	NA	78
3.4 GHz processor, 32 GB RAM, *n* = 1	NA	72
3.4 GHz processor, 24 GB RAM, *n* = 1	NA	86
3.50 GHz processor, 4 GB RAM, *n* = 1	NA	26
3.3 GHz processor, 8 GB RAM, *n* = 1	NA	78
3.20 GHz processor, 8 GB RAM, *n* = 1	NA	61
3.20 GHz processor, 4 GB RAM, *n* = 1	NA	44
2.93 GHz processor, 16 GB RAM, *n* = 1	NA	151
2.67 GHz processor, 8 GB RAM, *n* = 1	NA	137
Average 15-min peak power consumption (range)	82 (26-151)	

Source: Bach and Sarfraz (2017)
n = number of tested equipment of same configuration.
[a]Nameplate for desktop computer is present on its power supply, which is mounted inside desktop, hence not accessible for most computers, where NA = not available.
[b]For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.
[c]For tested equipment with same configuration, increasing power supply size does not increase average power consumption.
[d]Approximately 90% convective heat gain and 10% radiative heat gain.

Table 11.29 Recommended Heat Gain for Typical Laptops and Laptop Docking Station
[2017F, Ch 18, Tbl 8B]

Equipment	Description	Nameplate Power,[a] W	Peak Heat Gain,[b, c] W
Laptop computer	Manufacturer 1, 2.6 GHz processor, 8 GB RAM, n = 1	NA	46
	Manufacturer 2, 2.4 GHz processor, 4 GB RAM, n = 1	NA	59
Average 15-min peak power consumption (range)		53 (46-59)	
Laptop with docking station	Manufacturer 1, 2.7 GHz processor, 8 GB RAM, n = 1	NA	38
	1.6 GHz processor, 8 GB RAM, n = 2	NA	45
	2.0 GHz processor, 8 GB RAM, n = 1	NA	50
	2.6 GHz processor, 4 GB RAM, n = 1	NA	51
	2.4 GHz processor, 8 GB RAM, n = 1	NA	40
	2.6 GHz processor, 8 GB RAM, n = 1	NA	35
	2.7 GHz processor, 8 GB RAM, n = 1	NA	59
	3.0 GHz processor, 8 GB RAM, n = 3	NA	70
	2.9 GHz processor, 32 GB RAM, n = 3	NA	58
	3.0 GHz processor, 32 GB RAM, n = 1	NA	128
	3.7 GHz processor, 32 GB RAM, n = 1	NA	63
	3.1 GHz processor, 32 GB RAM, n = 1	NA	89
Average 15-min peak power consumption (range)		61 (26-151)	

Source: Bach and Sarfraz (2017)
n = number of tested equipment of same configuration.
[a]Voltage and amperage information for laptop computer and laptop docking station is available on power supply nameplates; however, nameplate does not provide information on power consumption, where NA = not available.
[b]For equipment peak heat gain value, the highest 15-min interval of recorded data is listed in tables.
[c]Approximately 75% convective heat gain and 25% radiative heat gain.

Table 11.30 Recommended Heat Gain for Typical Tablet PC
[2017F, Ch 18, Tbl 8C]

Description	Nameplate Power,[a] W	Peak Heat Gain,[b] W
1.7 GHz processor, 4 GB RAM, n = 1	NA	42
2.2 GHz processor, 16 GB RAM, n = 1	NA	40
2.3 GHz processor, 8 GB RAM, n = 1	NA	30
2.5 GHz processor, 8 GB RAM, n = 1	NA	31
Average 15-min peak power consumption (range)	36 (31-42)	

Source: Bach and Sarfraz (2017)
n = number of tested equipment of same configuration.
[a]Voltage and amperage information for tablet PC is available on power supply nameplate; however, nameplate does not provide information on power consumption, where NA = not available.
[b]For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.

Table 11.31 Recommended Heat Gain for Typical Monitors
[2017F, Ch 18, Tbl 8D]

Description[a]	Nameplate Power, W	Peak Heat Gain,[b, c] W
Manufacturer 1		
1397 mm LED flat screen, *n* = 1 (excluded from average because atypical size)	240	50
686 mm LED flat screen, *n* = 2	40	26
546 mm LED flat screen, *n* = 2	29	25
Manufacturer 2		
1270 mm 3D LED flat screen, *n* = 1 (excluded from average because atypical size)	94	49
Manufacturer 3		
864 mm LCD curved screen, *n* = 1 (excluded from average because atypical size and curved)	130	48
584 mm LED flat screen, *n* = 3	50	17
584 mm LED flat screen, *n* = 1	38	21
584 mm LED flat screen, *n* = 1	38	14
Manufacturer 4		
610 mm LED flat screen, *n* = 1	42	25
Manufacturer 5		
600 mm LED flat screen, *n* = 1	26	17
546 mm LED flat screen, *n* = 1	29	22
Manufacturer 6		
546 mm LED flat screen, *n* = 1	28	24
Average 15-min peak power consumption (range)	21 (14-26)	

Source: Bach and Sarfraz (2017)
n = number of tested equipment of same configuration.
[a]Screens with atypical size and shape are excluded for calculating average 15-min peak power consumption.
[b]For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.
[c]Approximately 60% convective heat gain and 40% radiative heat gain.

Table 11.32 Recommended Heat Gain for Miscellaneous Equipment
[2017F, Ch 18, Tbl 10]

Equipment	Nameplate Power,[a] W	Peak Heat Gain,[b] W
Vending machine		
Drinks, 280 to 400 items	NA	940
Snacks	NA	54
Food (e.g., for sandwiches)	NA	465
Thermal binding machine, 2 single documents up to 340 pages	350	28.5
Projector, resolution 1024 × 768	340	308
Paper shredder, up to 28 sheets	1415	265
Electric stapler, up to 45 sheets	NA	1.5
Speakers	220	15
Temperature-controlled electronics soldering station	95	16
Cell phone charger	NA	5
Battery charger		
40 V	NA	19
AA	NA	5.5
Microwave oven, 7 to 9 gal	1000 to 1550	713 to 822
Coffee maker		
Single cup	1400	385
Up to 12 cups	950	780
With grinder	1350	376
Coffee grinder, up to 12 cups	NA	73
Tea kettle, up to 6 cups	1200	1200
Dorm fridge, 3.1 ft^3	NA	57
Freezer, 18 ft^3	130	125
Fridge, 18 to 28 ft^3	NA	387 to 430
Ice maker and dispenser, 20 lb bin capacity	NA	658
Top mounted bottled water cooler	NA	114 to 350
Cash register	25	9
Touch screen computer, 15 in. standard LCD and 2.2 GHz processor	NA	58
Self-checkout machine	NA	15

Source: Bach and Sarfraz (2017)
[a]For some equipment, nameplate power consumption is not available, where NA = not available.
[b]For equipment peak heat gain value, highest 15-min interval of recorded data is listed in tables.

Table 11.33 Recommended Load Factors for Various Types of Offices
[2017F, Ch 18, Tbl 11]

Type of Use	Load Factor*, W/ft^2	Description
100% laptop, docking station		
light	0.34	167 ft^2/workstation, all laptop docking station use, 1 printer per 10
medium	0.46	125 ft^2/workstation, all laptop docking station use, 1 printer per 10
50% laptop, docking station		
light	0.44	167 ft^2/workstation, 50% laptop docking station/50% desktop, 1 printer per 10
medium	0.59	125 ft^2/workstation, 50% laptop docking station/50% desktop, 1 printer per 10
100% desktop		
light	0.54	167 ft^2/workstation, all desktop use, 1 printer per 10
medium	0.72	125 ft^2/workstation, all desktop use, 1 printer per 10
100% laptop, docking station		
2 screens	0.69	125 ft^2/workstation, all laptop docking station use, 2 screens, 1 printer per 10
100% desktop		
2 screens	0.84	125 ft^2/workstation, all laptop use, 2 screens, 1 printer per 10
3 screens	0.96	125 ft^2/workstation, all desktop use, 3 screens, 1 printer per 10
100% desktop		
heavy, 2 screens	1.02	85 ft^2/workstation, all desktop use, 2 screens, 1 printer per 8
heavy, 3 screens	1.16	85 ft^2/workstation, all desktop use, 3 screens, 1 printer per 8
100% laptop, docking station		
full on, 2 screens	1.14	85 ft^2/workstation, all laptop docking use, 2 screens, 1 printer per 8, no diversity
100% desktop		
full on, 2 screens	1.33	85 ft^2/workstation, all desktop use, 2 screens, 1 printer per 8, no diversity
full on, 3 screens	1.53	85 ft^2/workstation, all desktop use, 3 screens, 1 printer per 8, no diversity

Source: Bach and Sarfraz (2017)
*Medium-office type monochrome printer is used for load factor calculator with 15-min peak power consumption of 142 W.

Table 11.34 Diversity Factor for Different Equipment [2017F, Ch 18, Tbl 12]

Equipment	Diversity Factor, %	Diversity Factor,[a] %
Desktop PC	75	75
Laptop docking station	70	NA
Notebook computer	75[b]	75
Screen	70	60
Printer	45	NA

Source: Bach and Sarfraz (2017)
[a]2013 *ASHRAE Handbook—Fundamentals*
[b]Insufficient data from RP-1742; values based on previous data from 2013 *ASHRAE Handbook—Fundamentals* and judgment of Bach and Sarfraz (2017).

Table 11.35 Refrigerating Effect Produced by Open Refrigerated Display Fixtures

	Btu/h· ft of Fixture*		
Type of Display Fixture	**Latent Heat**	**Sensible Heat**	**Total Refrigerating Effect**
Low temperature			
Frozen Food			
Single Deck	38	207	245
Single Deck, Double Island	70	400	470
2 Deck	144	576	720
3 Deck	322	1288	1610
4 or 5 Deck	400	1600	2000
Ice Cream			
Single Deck	64	366	430
Single Deck, Double Island	70	400	470
Standard Temperature			
Meats			
Single Deck	52	298	350
Multideck	219	876	1095
Dairy			
Multideck	196	784	980
Produce			
Single Deck	36	204	240
Multideck	192	768	960

*These figures are general magnitudes for fixtures adjusted for average desired product temperatures and apply to store ambients in front of the display cases of 72°F to 74°F with 50% to 55% rh. Raising the dry bulb only 3°F to 5°F and the humidity 5% to 10% can increase heat removal 25% or more. Equally lower temperatures and humidities as in winter, have an equally marked effect on lowering heat removal from the space.

12. VENTILATION

ASHRAE Standard 62.2-2016, *Ventilation and Acceptable Indoor Air Quality in Residential Buildings*

(*See complete standard for detailed guidance.*)

Residential Ventilation for Single and Multiple Family Structures, Including Manufactured and Modular Houses

Whole-house mechanical ventilation systems are required for each dwelling unit:

$$\text{cfm} = 0.03\ (\text{ft}^2\ \text{floor area}) + 7.5\ (\text{number of bedrooms} + 1) \tag{12.1}$$

Exceptions: (a) building has no mechanical cooling and is in zone 1 or 2 of the climate zone map (see Figure 13.1), or (b) building is thermally conditioned for human occupancy for less than 876 h per year **and** if the authority having jurisdiction determines that window ventilation is sufficient.

Alternate means may be used to provide the required ventilation rate when approved by a licensed design professional. In hot, humid climates, whole-house net mechanical exhaust shall not exceed 7.5 cfm per 100 ft^2. In severe cold climates, net supply systems shall not exceed 7.5 cfm per 100 ft^2. (Climates are defined in Figure 13.1.)

Local mechanical exhaust rates are shown in Tables 12.2 and 12.3.

Ventilation openings: not less than 4% of floor nor less than 5 ft^2 for habitable rooms; and not less than 4% of floor space nor less than 1.5 ft^2 for toilets and utility rooms.

Supply ductwork for thermal conditioners except evaporative coolers, shall have a MERV 6 filter or better in accordance with ASHRAE Standard 52.2.

Airflows all refer to delivered airflow as tested, or the fans' rating at 0.25 in. w.g. with duct sizing meet the prescriptive sizing of Table 12.4.

Table 12.1 Ventilation Air Requirements, cfm [Std 62.2-2016, Tbl 4.1a]

Floor Area, ft^2	Bedrooms				
	1	**2**	**3**	**4**	**5**
<500	30	38	45	53	60
501–1000	45	53	60	68	75
1001–1500	60	68	75	83	90
1501–2000	75	83	90	98	105
2001–2500	90	98	105	113	120
2501–3000	105	113	120	128	135
3001–3500	120	128	135	143	150
3501–4000	135	143	150	158	165
4001–4500	150	158	165	173	180
4501–5000	165	173	180	188	195

Table 12.2 Demand-Controlled Local Ventilation Exhaust Airflow Rates
[Std 62.2-2016, Tbl 5.1]

Application	Airflow
Enclosed Kitchen	• Vented range hood (including appliance-range hood combinations): 100 cfm • Other kitchen exhaust fans, including downdraft: 300 cfm or a capacity of 5 ach
Nonenclosed Kitchen	• Vented range hood (including appliance-range hood combinations): 100 cfm • Other kitchen exhaust fans, including downdraft: 300 cfm
Bathroom	50 cfm

Table 12.3 Continuous Local Ventilation Exhaust Airflow Rates
[Std 62.2-2016, Tbl 5.2]

Application	Airflow
Enclosed Kitchen	5 ach, based on kitchen volume
Bathroom	20 cfm

Table 12.4 Prescriptive Duct Sizing [Std 62.2-2016, Tbl 5.3]

Duct Type	**Flex Duct**								**Smooth Duct**							
Fan Airflow Rating, cfm @ 0.25 in. of water	50	80	100	125	150	200	250	300	50	80	100	125	150	200	250	300
Diameter[a], in.	**Maximum Length[b,c,d], ft**															
3	×	×	×	×	×	×	×	×	5	×	×	×	×	×	×	×
4	56	4	×	×	×	×	×	×	114	31	10	×	×	×	×	×
5	NL	81	42	16	2	×	×	×	NL	152	91	51	28	4	×	×
6	NL	NL	158	91	55	18	1	×	NL	NL	NL	168	112	53	25	9
7	NL	NL	NL	NL	161	78	40	19	NL	NL	NL	NL	NL	148	88	54
8 and above	NL	NL	NL	NL	NL	189	111	69	NL	NL	NL	NL	NL	NL	198	133

a. For noncircular ducts, calculate the diameter as four times the cross-sectional area divided by the perimeter.
b. This table assumes no elbows. Deduct 15 ft of allowable duct length for each elbow.
c. NL = no limit on duct length of this size.
d. × = not allowed; any length of duct of this size with assumed turns and fitting will exceed the rated pressure drop.

ASHRAE Standard 62.1-2016, *Ventilation for Acceptable Indoor Air Quality*

(*See complete standard for detailed guidance.*)

General

Use of natural ventilation systems is permitted in lieu of or in conjunction with mechanical ventilation. Naturally ventilated spaces shall be permanently open to operable wall or roof openings to the outdoors; free openable area at least 4% of net occupiable floor area. If interior spaces are ventilated through adjoining rooms, free area between rooms shall be permanently unobstructed and at least 8% of the area of the interior room, nor less than 25 ft^2. Occupants must have ready access to the openings.

All airstream surfaces shall be designed to resist mold growth and resist erosion. Ductwork construction shall meet SMACNA standards. In accordance with manufacturer instructions, fuel-burning appliances shall have sufficient air for combustion and adequate removal of combustion products, which shall be vented directly outdoors. Filters or air cleaners with minimum MERV 8 by ASHRAE Standard 52.2 shall be provided upstream of all cooling coils or other devices with wetted surfaces through which air is supplied to occupiable space. Relative humidity should be 65% or less when system performance is analyzed with outdoor at the design dew point and mean coincident dry bulb, sensible and latent space interior loads at cooling design values, and space solar loads at zero. Drain pans slope minimum 1/8 in. per ft to outlet at lowest point, and drain line shall have P-trap or other seal when drain pan is at negative static pressure relative to the outlet. Drain pan shall extend from leading edge of the coil to a distance of half the vertical dimension of the coil.

Discharge from noncombustion equipment that captures contaminants generated by the equipment shall be discharged directly outdoors.

Investigate outdoor air quality. Survey and document local and regional outdoor air quality, with description of noticeable air problems and conditions regarding its acceptability. If unacceptable, treat it. Cleaning for ozone is required only if in a high-ozone area (see Informative Appendix F of the standard) and if the minimum design outdoor airflow is 1.5 air changes or more.

Outdoor air intakes shall be located so the shortest distance from intake to any specific contaminant source shall equal or exceed values in Table 12.5 or that result from the calculation method in Normative Appendix B of ASHRAE Standard 62.1.

Design intakes to manage rain and snow entrainment and include bird screens.

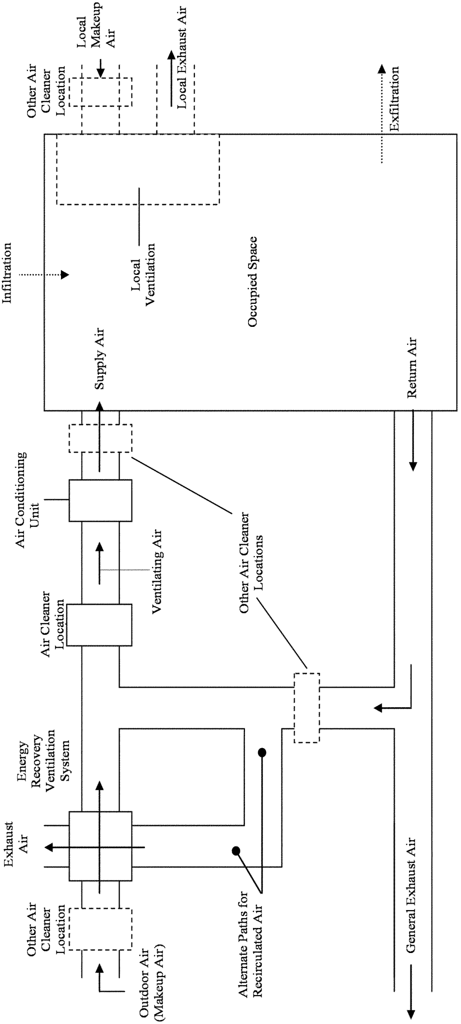


Figure 12.1 Ventilation System [Std 62.1-2016, Fig 3.1]

Table 12.5 Air Intake Minimum Separation Distance [Std 62.1-2016, Tbl 5.5.1]

Object	Minimum Distance, ft
Class 2 air exhaust/relief outlet[a]	10
Class 3 air exhaust/relief outlet[a]	15
Class 4 air exhaust/relief outlet[b]	30
Plumbing vents terminating less than 3 ft above the level of the outdoor air intake	10
Plumbing vents terminating at least 3 ft above the level of the outdoor air intake	3
Vents, chimneys, and flues from combustion appliances and equipment[c]	15
Garage entry, automobile loading area, or drive-in queue[d]	15
Truck loading area or dock, bus parking/idling area[d]	25
Driveway, street, or parking place[d]	5
Thoroughfare with high traffic volume	25
Roof, landscaped grade, or other surface directly below intake[e,f]	1
Garbage storage/pick-up area, dumpsters	15
Cooling tower intake or basin	15
Cooling tower exhaust	25

a. This requirement applies to the distance from the outdoor air intakes for one ventilation system to the exhaust outlets and relief outlets for any other ventilation system.

b. Minimum distance listed does not apply to laboratory fume hood exhaust air outlets. Separation criteria for fume hood exhaust shall be in compliance with ANSI/AIHA Z9.5. Informative Appendix J contains sources of additional information on separation criteria. These include the *ACGIH Industrial Ventilation Manual*, *ASHRAE Handbook—HVAC Applications*, *ASHRAE Laboratory Design Guide*, and NSF/ANSI 49.

c. The minimum distances relative to fuel-fired appliances shall be as required by ANSI Z223.1/NFPA 54 for fuel gas burning appliances and equipment, NFPA 31 for oil burning appliances and equipment, and NFPA 211 for other combustion appliances and equipment.

d. Distance measured to closest place that vehicle exhaust is likely to be located.

e. The minimum separation distance shall not apply where outdoor surfaces below the air intake are sloped more than 45 degrees from horizontal or where such surfaces are less than 1 in. in width.

f. Where snow accumulation is expected, the surface of the snow at the expected average snow depth shall be considered to be a surface directly below an intake.

Air classifications:

- Class 1: Air with low contaminant concentration, low sensory-irritation intensity, and inoffensive odor.
- Class 2: Air with moderate contaminant concentration, mild sensory-irritation intensity, or mildly offensive odors. (Class 2 air also includes air that is not necessarily harmful or objectionable but that is inappropriate for transfer or recirculation to spaces used for different purposes.)
- Class 3: Air with significant contaminant concentration, significant sensory-irritation intensity, or offensive odor.
- Class 4: Air with highly objectionable fumes or gases or with potentially dangerous particles, bioaerosols, or gases, at concentrations high enough to be considered harmful.

The Ventilation Rate Procedure, the Natural Ventilation Procedure, or the Indoor Air Quality (IAQ) Procedure, or a combination thereof, shall be used to design ventilation systems. The IAQ procedure is based on analysis of contaminant sources, concentrations, and targets, and perceived acceptability targets.

Table 12.6 Airstreams or Sources [Std 62.1-2016, Tbl 5.16.1]

Description	Air Class
Diazo printing equipment discharge	4
Commercial kitchen grease hoods	4
Commercial kitchen hoods other than grease	3
Laboratory hoods	4*
Residential kitchen hoods	3
Hydraulic elevator machine room	2

*Air Class 4 unless determined otherwise by the Environmental Health and Safety professional responsible to the owner or to the owner's designee

Procedures from ASHRAE Standard 62.1-2016

6.1 General. The Ventilation Rate Procedure, the IAQ Procedure, the Natural Ventilation Procedure, or a combination thereof shall be used to meet the requirements of this section. In addition, the requirements for exhaust ventilation in Section 6.5 shall be met regardless of the method used to determine minimum outdoor airflow rates.

Informative Note: Although the intake airflow determined using each of these approaches may differ significantly because of assumptions about the design, any of these approaches is a valid basis for design.

6.1.1 Ventilation Rate Procedure. The prescriptive design procedure presented in Section 6.2, in which outdoor air intake rates are determined based on space type/application, occupancy level, and floor area, shall be permitted to be used for any zone or system.

Informative Note: The Ventilation Rate Procedure minimum rates are based on contaminant sources and source strengths that are typical for the listed occupancy categories.

6.1.2 IAQ Procedure. This performance-based design procedure presented in Section 6.3, in which the building outdoor air intake rates and other system design parameters are based on an analysis of contaminant sources, contaminant concentration limits, and level of perceived indoor air acceptability, shall be permitted to be used for any zone or system.

6.1.3 Natural Ventilation Procedure. The prescriptive design procedure presented in Section 6.4, in which outdoor air is provided through openings to the outdoors, shall be permitted to be used for any zone or portion of a zone in conjunction with mechanical ventilation systems in accordance with Section 6.4.

6.2 Ventilation Rate Procedure. The outdoor air intake flow (V_{ot}) for a ventilation system shall be determined in accordance with Sections 6.2.1 through 6.2.7.

Informative Note: Additional explanation of terms used below is contained in Normative Appendix A, along with a ventilation system schematic (Figure 12.2).

6.2.1 Outdoor Air Treatment. Each ventilation system that provides outdoor air through a supply fan shall comply with the following subsections.

Exception: Systems supplying air for enclosed parking garages, warehouses, storage rooms, janitor's closets, trash rooms, recycling areas, shipping/receiving/distribution areas.

Informative Note: Occupied spaces ventilated with outdoor air that is judged to be unacceptable are subject to reduced air quality when outdoor air is not cleaned prior to introduction to the occupied spaces.

6.2.1.1 Particulate Matter Smaller than 10 Micrometers (PM10). In buildings located in an area where the national standard or guideline for PM10 is exceeded, particle filters or air-cleaning devices shall be provided to clean the outdoor air at any location prior to its introduction to occupied spaces. Particulate matter filters or air cleaners shall have an efficiency reporting value (MERV) of not less than 6 where rated in accordance with ASHRAE Standard 52.2.

Informative Note: See Informative Appendix F for resources regarding selected PM10 national standards and guidelines.

6.2.1.2 Particulate Matter Smaller than 2.5 Micrometers (PM2.5). In buildings located in an area where the national standard or guideline for PM2.5 is exceeded, particle filters or air-cleaning devices shall be provided to clean the outdoor air at any location prior to its introduction to occupied spaces. Particulate matter filters or air cleaners shall have an efficiency reporting value (MERV) of not less than 11 where rated in accordance with ASHRAE Standard 52.2.

Informative Note: See Informative Appendix F for resources regarding selected PM2.5 national standards and guidelines.

6.2.1.3 Ozone. Air-cleaning devices for ozone shall be provided when the most recent three-year average annual fourth-highest daily maximum eight-hour average ozone concentration exceeds 0.107 ppm.

Such air-cleaning devices shall have a volumetric ozone removal efficiency of not less than 40% where installed, operated, and maintained in accordance with manufacturer recommendations and shall be approved by the authority having jurisdiction. Such devices shall be operated where the outdoor ozone levels are expected to exceed 0.107 ppm.

Exceptions: Air cleaning for ozone shall not be required where

1. the system design outdoor air intake flow is 1.5 ach or less,
2. controls are provided that sense outdoor ozone level and reduce intake airflow to 1.5 ach or less while complying with the outdoor airflow requirements of Section 6, or
3. outdoor air is brought into the building and heated by direct-fired makeup air units.

Informative Note: See Informative Appendix F for a map of United States locations exceeding the most recent three-year average annual fourth-highest daily maximum eight-hour average ozone concentration of 0.107 ppm.

6.2.1.4 Other Outdoor Contaminants. In buildings located in an area where the national standard for one or more contaminants not addressed in Section 6.2.1 is exceeded, any design assumptions and calculations related to the impact on indoor air quality shall be included in the design documents.

6.2.2 Zone Calculations. Ventilation zone parameters shall be determined in accordance with Sections 6.2.2.1 through 6.2.2.3 for ventilation zones served by the ventilation system.

6.2.2.1 Breathing Zone Outdoor Airflow. The outdoor airflow required in the breathing zone (V_{bz}) of the occupiable space or spaces in a ventilation zone shall be not less than the value determined in accordance with Equation 12.2:

$$V_{bz} = R_p \times P_z + R_a \times A_z \tag{12.2}$$

where

A_z = zone floor area, the net occupiable floor area of the ventilation zone, ft^2

P_z = zone population, the number of people in the ventilation zone during use

R_p = outdoor airflow rate required per person as determined from Table 12.7

Informative Note: These values are based on adapted occupants.

R_a = outdoor airflow rate required per unit area as determined from Table 12.7

Informative Note: Equation 12.2 accounts for people-related sources and area-related sources independently in the determination of the outdoor air rate required at the breathing zone. The use of Equation 12.2 in the context of this standard does not necessarily imply that simple addition of outdoor airflow rates for different sources can be applied to any other aspect of indoor air quality.

6.2.2.1.1 Design Zone Population. Design zone population (P_z) shall equal the largest (peak) number of people expected to occupy the ventilation zone during typical use.

Exceptions:

1. Where the number of people expected to occupy the ventilation zone fluctuates, zone population equal to the average number of people shall be permitted, provided such average is determined in accordance with Section 6.2.6.2.
2. Where the largest or average number of people expected to occupy the ventilation zone cannot be established for a specific design, an estimated value for

zone population shall be permitted, provided such value is the product of the net occupiable area of the ventilation zone and the default occupant density listed in Table 12.7.

6.2.2.2 Zone Air Distribution Effectiveness. The zone air distribution effectiveness (E_z) shall be not greater than the default value determined using Table 6.2.2.2 of Standard 62.1-2016.

Informative Note: For some configurations, the default value depends upon space and supply air temperature.

6.2.2.3 Zone Outdoor Airflow. The zone outdoor airflow (V_{oz}) provided to the ventilation zone by the supply air distribution system shall be determined in accordance with Equation 12.3:

$$V_{oz} = V_{bz}/E_z \tag{12.3}$$

Ventilation

6.2.3 Single-Zone Systems. For ventilation systems wherein one or more air handlers supply a mixture of outdoor air and recirculated air to only one ventilation zone, the outdoor air intake flow (V_{ot}) shall be determined in accordance with Equation 12.4:

$$V_{ot} = V_{oz} \tag{12.4}$$

6.2.4 100% Outdoor Air Systems. For ventilation systems wherein one or more air handlers supply only outdoor air to one or more ventilation zones, the outdoor air intake flow (V_{ot}) shall be determined in accordance with Equation 12.5:

$$V_{ot} = \Sigma_{all\ zones} V_{oz} \tag{12.5}$$

6.2.5 Multiple-Zone Recirculating Systems. For ventilation systems wherein one or more air handlers supply a mixture of outdoor air and recirculated air to more than one ventilation zone, the outdoor air intake flow (V_{ot}) shall be determined in accordance with Sections 6.2.5.1 through 6.2.5.4.

6.2.5.1 Primary Outdoor Air Fraction. Primary outdoor air fraction (Z_{pz}) shall be determined for ventilation zones in accordance with Equation 12.6:

$$Z_{pz} = V_{oz}/V_{pz} \tag{12.6}$$

where V_{pz} is the zone primary airflow to the ventilation zone, including outdoor air and recirculated air.

a. For VAV-system design purposes, V_{pz} is the lowest zone primary airflow value expected at the design condition analyzed.
b. In some cases, it is permitted to determine these parameters for only selected zones as outlined in Normative Appendix A.

6.2.5.2 System Ventilation Efficiency. The system ventilation efficiency (E_v) shall be determined in accordance with Table 6.2.5.2 or Normative Appendix A of Standard 62.1-2016.

6.2.5.3 Uncorrected Outdoor Air Intake. The uncorrected outdoor air intake (V_{ou}) flow shall be determined in accordance with Equation 12.7:

$$V_{ou} = D\Sigma_{all\ zones}(R_p \times P_z) + \Sigma_{all\ zones}(R_a \times A_z) \tag{12.7}$$

6.2.5.3.1 Occupant Diversity. The occupant diversity ratio (D) shall be determined in accordance with Equation 12.8 to account for variations in population within the ventilation zones served by the system.

$$D = P_s/\Sigma_{all\ zones} P_z \tag{12.8}$$

where the system population (P_s) is the total population in the area served by the system.

Exception: Alternative methods to account for occupant diversity shall be permitted, provided the resulting V_{ou} value is not less than that determined using Equation 12.7.

Informative Note: The uncorrected outdoor air intake (V_{ou}) is adjusted for occupant diversity, but it is not corrected for system ventilation efficiency.

Table 12.7 Minimum Ventilation Rates in Breathing Zone

[Std 62.1-2016, Tbl 6.2.2.1]

(Table 12.7 shall be used in conjunction with the accompanying notes.)

Occupancy Category	People Outdoor Air Rate R_p cfm/person	Area Outdoor Air Rate R_a cfm/ft²	Notes	Default Values: Occupant Density (see Note 4) #/1000 ft²	Default Values: Combined Outdoor Air Rate (see Note 5) cfm/person	Air Class
Correctional Facilities						
Cell	5	0.12		25	10	2
Dayroom	5	0.06		30	7	1
Guard stations	5	0.06		15	9	1
Booking/waiting	7.5	0.06		50	9	2
Educational Facilities						
Daycare (through age 4)	10	0.18		25	17	2
Daycare sickroom	10	0.18		25	17	3
Classrooms (ages 5–8)	10	0.12		25	15	1
Classrooms (age 9 plus)	10	0.12		35	13	1
Lecture classroom	7.5	0.06	H	65	8	1
Lecture hall (fixed seats)	7.5	0.06	H	150	8	1
Art classroom	10	0.18		20	19	2
Science laboratories	10	0.18		25	17	2
University/college laboratories	10	0.18		25	17	2
Wood/metal shop	10	0.18		20	19	2
Computer lab	10	0.12		25	15	1
Media center	10	0.12	A	25	15	1
Music/theater/dance	10	0.06	H	35	12	1
Multiuse assembly	7.5	0.06	H	100	8	1
Food and Beverage Service						
Restaurant dining rooms	7.5	0.18		70	10	2
Cafeteria/fast-food dining	7.5	0.18		100	9	2
Bars, cocktail lounges	7.5	0.18		100	9	2
Kitchen (cooking)	7.5	0.12		20	14	2
General						
Break rooms	5	0.06	H	25	7	1
Coffee stations	5	0.06	H	20	8	1
Conference/meeting	5	0.06	H	50	6	1
Corridors	—	0.06	H	—		1
Occupiable storage rooms for liquids or gels	5	0.12	B	2	65	2
Hotels, Motels, Resorts, Dormitories						
Bedroom/living room	5	0.06	H	10	11	1
Barracks sleeping areas	5	0.06	H	20	8	1
Laundry rooms, central	5	0.12		10	17	2
Laundry rooms within dwelling units	5	0.12		10	17	1
Lobbies/prefunction	7.5	0.06	H	30	10	1
Multipurpose assembly	5	0.06	H	120	6	1
Office Buildings						
Breakrooms	5	0.12		50	7	1
Main entry lobbies	5	0.06	H	10	11	1
Occupiable storage rooms for dry materials	5	0.06		2	35	1
Office space	5	0.06	H	5	17	1
Reception areas	5	0.06	H	30	7	1
Telephone/data entry	5	0.06	H	60	6	1

Table 12.7 Minimum Ventilation Rates in Breathing Zone
[Std 62.1-2016, Tbl 6.2.2.1] ***(Continued)***

(Table 12.7 shall be used in conjunction with the accompanying notes.)

Occupancy Category	People Outdoor Air Rate R_p cfm/person	Area Outdoor Air Rate R_a cfm/ft²	Notes	Default Values: Occupant Density (see Note 4) #/1000 ft²	Default Values: Combined Outdoor Air Rate (see Note 5) cfm/person	Air Class
Miscellaneous Spaces						
Bank vaults/safe deposit	5	0.06	H	5	17	2
Banks or bank lobbies	7.5	0.06	H	15	12	1
Computer (not printing)	5	0.06	H	4	20	1
Freezer and refrigerated spaces (<50°F)	10	0	E	0	0	2
General manufacturing (excludes heavy industrial and processes using chemicals)	10	0.18		7	36	3
Pharmacy (prep. area)	5	0.18		10	23	2
Photo studios	5	0.12		10	17	1
Shipping/receiving	10	0.12	B	2	70	2
Sorting, packing, light assembly	7.5	0.12		7	25	2
Telephone closets	—	0.00		—		1
Transportation waiting	7.5	0.06	H	100	8	1
Warehouses	10	0.06	B	—		2
Public Assembly Spaces						
Auditorium seating area	5	0.06	H	150	5	1
Places of religious worship	5	0.06	H	120	6	1
Courtrooms	5	0.06	H	70	6	1
Legislative chambers	5	0.06	H	50	6	1
Libraries	5	0.12		10	17	1
Lobbies	5	0.06	H	150	5	1
Museums (children's)	7.5	0.12		40	11	1
Museums/galleries	7.5	0.06	H	40	9	1
Residential						
Dwelling unit	5	0.06	F,G, H	F		1
Common corridors	—	0.06	H			1
Retail						
Sales (except as below)	7.5	0.12		15	16	2
Mall common areas	7.5	0.06	H	40	9	1
Barbershop	7.5	0.06	H	25	10	2
Beauty and nail salons	20	0.12		25	25	2
Pet shops (animal areas)	7.5	0.18		10	26	2
Supermarket	7.5	0.06	H	8	15	1
Coin-operated laundries	7.5	0.12		20	14	2
Sports and Entertainment						
Gym, sports arena (play area)	20	0.18	E	7	45	2
Spectator areas	7.5	0.06	H	150	8	1
Swimming (pool & deck)	—	0.48	C	—		2
Disco/dance floors	20	0.06	H	100	21	2
Health club/aerobics room	20	0.06		40	22	2
Health club/weight rooms	20	0.06		10	26	2
Bowling alley (seating)	10	0.12		40	13	1
Gambling casinos	7.5	0.18		120	9	1
Game arcades	7.5	0.18		20	17	1
Stages, studios	10	0.06	D, H	70	11	1

GENERAL NOTES FOR TABLE 12.7

1 Related requirements: The rates in this table are based on all other applicable requirements of this standard being met.

2 Environmental Tobacco Smoke: This table applies to ETS-free areas. Refer to Section 5.17 for requirements for buildings containing ETS areas and ETS-free areas.

3 Air density: Volumetric airflow rates are based on dry air density of 0.075 lb_{da}/ft^3 at a barometric pressure of 1 atm and an air temperature of 70°F. Rates shall be permitted to be adjusted for actual density.

4 Default occupant density: The default occupant density shall be used where the actual occupant density is not known.

5 Default combined outdoor air rate (per person): Rate is based on the default occupant density.

6 Unlisted occupancies: Where the occupancy category for a proposed space or zone is not listed, the requirements for the listed occupancy category that is most similar in terms of occupant density, activities, and building construction shall be used.

ITEM-SPECIFIC NOTES FOR TABLE 12.7

A For high-school and college libraries, the values shown for "Public Assembly Spaces—Libraries" shall be used.

B Rate may not be sufficient where stored materials include those having potentially harmful emissions.

C Rate does not allow for humidity control. "Deck area" refers to the area surrounding the pool that is capable of being wetted during pool use or when the pool is occupied. Deck area that is not expected to be wetted shall be designated as an occupancy category.

D Rate does not include special exhaust for stage effects such as dry ice vapors and smoke.

E Where combustion equipment is intended to be used on the playing surface or in the space, additional dilution ventilation, source control, or both shall be provided.

F Default occupancy for dwelling units shall be two persons for studio and one-bedroom units, with one additional person for each additional bedroom.

G Air from one residential dwelling shall not be recirculated or transferred to any other space outside of that dwelling.

H Ventilation air for this occupancy category shall be permitted to be reduced to zero when the space is in occupied-standby mode.

6.2.5.3.2 Design System Population. Design system population (P_s) shall equal the largest (peak) number of people expected to occupy all ventilation zones served by the ventilation system during use.

Informative Note: Design system population is always equal to or less than the sum of design zone population for all zones in the area served by the system because all zones may not be simultaneously occupied at design population.

6.2.5.4 Outdoor Air Intake. The design outdoor air intake flow (V_{ot}) shall be determined in accordance with Equation 12.9:

$$V_{ot} = V_{ou}/E_v \tag{12.9}$$

6.2.6 Design for Varying Operating Conditions

6.2.6.1 Variable Load Conditions. Ventilation systems shall be designed to be capable of providing not less than the minimum ventilation rates required in the breathing zone where the zones served by the system are occupied, including all full- and part-load conditions.

Informative Note: The minimum outdoor air intake flow may be less than the design value at part-load conditions.

6.2.6.2 Short-Term Conditions. Where it is known that peak occupancy will be of short duration, ventilation will be varied or interrupted for a short period of time, or both, the design shall be permitted to be based on the average conditions over a time period (T) determined by Equation 12.10:

$$T = 3v/V_{bz} \tag{12.10}$$

where

T = averaging time period, min

v = the volume of the ventilation zone where averaging is being applied, ft^3

V_{bz} = the breathing zone outdoor airflow calculated using Equation 12.2 and the design value of the zone population (P_z), cfm

Acceptable design adjustments based on this optional provision include the following:

a. Zones with fluctuating occupancy: The zone population (P_z) shall be permitted to be averaged over time (T).

Table 12.8 System Ventilation Efficiency [Std 62.1-2016, Tbl 6.2.5.1]

Max (Z_{pz})	E_v
≤0.15	1.0
≤0.25	0.9
≤0.35	0.8
≤0.45	0.7
≤0.55	0.6
>0.55	Use Normative Appendix A of Standard 62.1-2016

NOTES:

1. "Max (Z_{pz})" refers to the largest value of Z_{pz}, calculated using Equation 12.6, among all the ventilation zones served by the system.
2. For values of Max (Z_{pz}) between 0.15 and 0.55, the corresponding value of E_v may be determined by interpolating the values in the table.
3. The values of E_v in this table are based on a 0.15 average outdoor air fraction for the system. For systems with higher values of the average outdoor air fraction, this table may result in unrealistically low values of E_v and the use of Normative Appendix A may yield more practical results.

Ventilation

b. Zones with intermittent interruption of supply air: The average outdoor airflow supplied to the breathing zone over time (T) shall be not less than the breathing zone outdoor airflow (V_{bz}) calculated using Equation 12.2.

c. Systems with intermittent closure of the outdoor air intake: The average outdoor air intake over time (T) shall be not less than the minimum outdoor air intake (V_{ot}) calculated using Equation 12.4, 12.5, or 12.9 as appropriate.

6.2.7 Dynamic Reset. The system shall be permitted to be designed to reset the outdoor air intake flow (V_{ot}), the space or ventilation zone airflow (V_{oz}) as operating conditions change, or both.

6.2.7.1 Demand Control Ventilation (DCV). DCV shall be permitted as an optional means of dynamic reset.

Exception: CO_2-based DCV shall not be applied in zones with indoor sources of CO_2 other than occupants or with CO_2 removal mechanisms, such as gaseous air cleaners.

6.2.7.1.1 For DCV zones in the occupied mode, breathing zone outdoor airflow (V_{bz}) shall be reset in response to current population.

6.2.7.1.2 For DCV zones in the occupied mode, breathing zone outdoor airflow (V_{bz}) shall be not less than the building component ($R_a \times A_z$) for the zone.

Exception: Breathing zone outdoor airflow shall be permitted to be reduced to zero for zones in occupied standby mode for the occupancy categories indicated in Table 12.7, provided that airflow is restored to V_{bz} whenever occupancy is detected.

6.2.7.1.3 Documentation. A written description of the equipment, methods, control sequences, setpoints, and the intended operational functions shall be provided. A table shall be provided that shows the minimum and maximum outdoor intake airflow for each system.

6.2.7.2 Ventilation Efficiency. Variations in the efficiency with which outdoor air is distributed to the occupants under different ventilation system airflows and temperatures shall be permitted as an optional basis of dynamic reset.

6.2.7.3 Outdoor Air Fraction. A higher fraction of outdoor air in the air supply due to intake of additional outdoor air for free cooling or exhaust air makeup shall be permitted as an optional basis of dynamic reset.

6.3 Indoor Air Quality (IAQ) Procedure. Breathing zone outdoor airflow (V_{bz}) shall be determined in accordance with Sections 6.3.1 through 6.3.5.

6.3.1 Contaminant Sources. Each contaminant of concern, for purposes of the design, shall be identified. For each contaminant of concern, indoor sources and outdoor sources shall be identified, and the emission rate for each contaminant of concern from each source shall be determined. Where two or more contaminants of concern target the same organ system, these contaminants shall be considered to be a contaminant mixture.

Informative Note: Informative Appendix C provides information for some potential contaminants of concern, including the organs they affect.

6.3.2 Contaminant Concentration. For each contaminant of concern, a concentration limit and its corresponding exposure period and an appropriate reference to a cognizant authority shall be specified. For each contaminant mixture of concern, the ratio of the concentration of each contaminant to its concentration limit shall be determined, and the sum of these ratios shall be not greater than one.

Exception: Consideration of odors in determining concentration limits shall not be required.

Informative Note:

1. Odors are addressed in Section 6.3.4.2.
2. Informative Appendix C includes concentration guidelines for some potential contaminants of concern.

6.3.3 Perceived Indoor Air Quality. The design level of indoor air acceptability shall be specified in terms of the percentage of building occupants, visitors, or both expressing satisfaction with perceived IAQ.

6.3.4 Design Approach. Zone and system outdoor airflow rates shall be the larger of those determined in accordance with Section 6.3.4.1 and either Section 6.3.4.2 or 6.3.4.3, based on emission rates, concentration limits, and other relevant design parameters.

6.3.4.1 Mass Balance Analysis. Using a steady-state or dynamic mass-balance analysis, the minimum outdoor airflow rates required to achieve the concentration limits specified in Section 6.3.2 shall be determined for each contaminant or contaminant mixture of concern within each zone served by the system.

Informative Note:

1. Informative Appendix E includes steady-state mass-balance equations that describe the impact of air cleaning on outdoor air and recirculation rates for ventilation systems serving a single zone.
2. In the completed building, measurement of the concentration of contaminants or contaminant mixtures of concern may be useful as a means of checking the accuracy of the design mass-balance analysis, but such measurement is not required for compliance.

6.3.4.2 Subjective Evaluation. Using a subjective occupant evaluation conducted in the completed building, the minimum outdoor airflow rates required to achieve the level of acceptability specified in Section 6.3.3 shall be determined within each zone served by the system.

Informative Note:

1. Informative Appendix C presents one approach to subjective occupant evaluation.
2. Level of acceptability often increases in response to increased outdoor airflow rates, increased level of indoor or outdoor air cleaning, or decreased indoor or outdoor contaminant emission rate.

6.3.4.3 Similar Zone. The minimum outdoor airflow rates shall be not less than those found in accordance with Section 6.3.4.2 for a substantially similar zone.

6.3.5 Combined IAQ Procedure and Ventilation Rate Procedure. The IAQ Procedure in conjunction with the Ventilation Rate Procedure shall be permitted to be applied to a zone or system. In this case, the Ventilation Rate Procedure shall be used to determine the required zone minimum outdoor airflow, and the IAQ Procedure shall be used to determine the additional outdoor air or air cleaning necessary to achieve the concentration limits of the contaminants and contaminant mixtures of concern.

Informative Note: The improvement of indoor air quality through the use of air cleaning or provision of additional outdoor air in conjunction with minimum ventilation rates may be quantified using the IAQ Procedure.

6.3.6 Documentation. Where the IAQ Procedure is used, the following information shall be included in the design documentation: the contaminants and contaminant mixtures of concern considered in the design process, the sources and emission rates of the contaminants of concern, the concentration limits and exposure periods and the references for these limits, and the analytical approach used to determine ventilation rates and air-cleaning requirements. The contaminant monitoring and occupant or visitor evaluation plans shall also be included in the documentation.

6.4 Natural Ventilation Procedure. Natural ventilation systems shall be designed in accordance with this section and shall include mechanical ventilation systems designed in accordance with Section 6.2, Section 6.3, or both.

Exceptions:

1. An engineered natural ventilation system, where approved by the authority having jurisdiction, need not meet the requirements of Section 6.4.
2. The mechanical ventilation systems shall not be required where
   a. natural ventilation openings that comply with the requirements of Section 6.4 are permanently open or have controls that prevent the openings from being closed during periods of expected occupancy or
   b. the zone is not served by heating or cooling equipment.

6.4.1 Floor Area to Be Ventilated. Spaces, or portions of spaces, to be naturally ventilated shall be located within a distance based on the ceiling height, as determined by Sections 6.4.1.1, 6.4.1.2, or 6.4.1.3, from operable wall openings that meet the requirements of Section 6.4.2. For spaces with ceilings that are not parallel to the floor, the ceiling height shall be determined in accordance with Section 6.4.1.4.

6.4.1.1 Single Side Opening. For spaces with operable openings on one side of the space, the maximum distance from the operable openings shall be not more than $2H$, where H is the ceiling height.

6.4.1.2 Double Side Opening. For spaces with operable openings on two opposite sides of the space, the maximum distance from the operable openings shall be not more than $5H$, where H is the ceiling height.

6.4.1.3 Corner Openings. For spaces with operable openings on two adjacent sides of a space, the maximum distance from the operable openings shall be not more than $5H$ along a line drawn between the two openings that are farthest apart. Floor area outside that line shall comply with Section 6.4.1.1.

6.4.1.4 Ceiling Height. The ceiling height (H) to be used in Sections 6.4.1.1 through 6.4.1.3 shall be the minimum ceiling height in the space.

Exception: For ceilings that are increasing in height as distance from the openings is increased, the ceiling height shall be determined as the average height of the ceiling within 6 m (20 ft) from the operable openings.

6.4.2 Location and Size of Openings. Spaces or portions of spaces to be naturally ventilated shall be permanently open to operable wall openings directly to the outdoors. The openable area shall be not less than 4% of the net occupiable floor area. Where openings are covered with louvers or otherwise obstructed, openable area shall be based on the net free unobstructed area through the opening. Where interior rooms, or portions of rooms, without direct openings to the outdoors are ventilated through adjoining rooms, the opening between rooms shall be permanently unobstructed and have a free area of not less than 8% of the area of the interior room or less than 25 ft^2 (2.3 m^2).

6.4.3 Control and Accessibility. The means to open required operable openings shall be readily accessible to building occupants whenever the space is occupied. Controls shall be designed to coordinate operation of the natural and mechanical ventilation systems.

6.5 Exhaust Ventilation. The Prescriptive Compliance Path or the Performance Compliance Path shall be used to meet the requirements of this section. Exhaust makeup air shall be permitted to be any combination of outdoor air, recirculated air, or transfer air.

6.5.1 Prescriptive Compliance Path. The design exhaust airflow shall be determined in accordance with the requirements in Table 6.5 of Standard 62.1-2016.

6.5.2 Performance Compliance Path. The exhaust airflow shall be determined in accordance with the following subsections.

6.5.2.1 Contaminant Sources. Contaminants or mixtures of concern for purposes of the design shall be identified. For each contaminant or mixture of concern, indoor sources (occupants, materials, activities, and processes) and outdoor sources shall be identified, and the emission rate for each contaminant of concern from each source shall be determined.

Informative Note: Informative Appendix C provides information for some potential contaminants of concern.

6.5.2.2 Contaminant Concentration. For each contaminant of concern, a concentration limit and its corresponding exposure period and an appropriate reference to a cognizant authority shall be specified.

Informative Note: Informative Appendix C includes concentration guidelines for some potential contaminants of concern.

6.5.2.3 Monitoring and control systems shall be provided to automatically detect contaminant levels of concern and modulate exhaust airflow such that contaminant levels are maintained at not greater than the specified contaminant concentration limits.

6.6 Design Documentation Procedures. Design criteria and assumptions shall be documented and made available for operation of the system after installation. See Sections 4.3, 5.1.3, 5.16.4, 6.2.7.1.4, and 6.3.6 regarding assumptions to be detailed in the documentation.

Ventilation

NORMATIVE APPENDIX A
MULTIPLE-ZONE SYSTEMS

This appendix presents an alternative procedure for calculating the system ventilation efficiency (E_v) that must be used when Standard 62.1-2016 Table 6.2.5.2 values are not used. In this alternative procedure, E_v is equal to the lowest calculated value of the zone ventilation efficiency (E_{vz}) (see Equation 12.12 below).

Informative Note: Figure 12.2 contains a ventilation system schematic depicting most of the quantities used in this appendix.

A1. SYSTEM VENTILATION EFFICIENCY

For any multiple-zone recirculating system, the system ventilation efficiency (E_v) shall be calculated in accordance with Sections A1.1 through A1.3.

A1.1 Average Outdoor Air Fraction. The average outdoor air fraction (X_s) for the ventilation system shall be determined in accordance with Equation 12.11:

$$X_s = V_{ou}/V_{ps} \tag{12.11}$$

where the uncorrected outdoor air intake (V_{ou}) is found in accordance with Section 6.2.5.3, and the system primary airflow (V_{ps}) is found at the condition analyzed.

Informative Note: For VAV-system design purposes, V_{ps} is the highest expected system primary airflow at the design condition analyzed. System primary airflow at design is usually less than the sum of design zone primary airflow values because primary airflow seldom peaks simultaneously in all VAV zones.

A1.2 Zone Ventilation Efficiency. The zone ventilation efficiency (E_{vz}) shall be determined in accordance with Section A1.2.1 or A1.2.2.

A1.2.1 Single Supply Systems. For single supply systems, wherein all of the air supplied to each ventilation zone is a mixture of outdoor air and system-level recirculated air, zone ventilation efficiency (E_{vz}) shall be determined in accordance with Equation 12.12. Examples of single supply systems include constant-volume reheat, single-duct VAV, single-fan dual-duct, and multizone systems.

$$E_{vz} = 1 + X_s - Z_{pz} \tag{12.12}$$

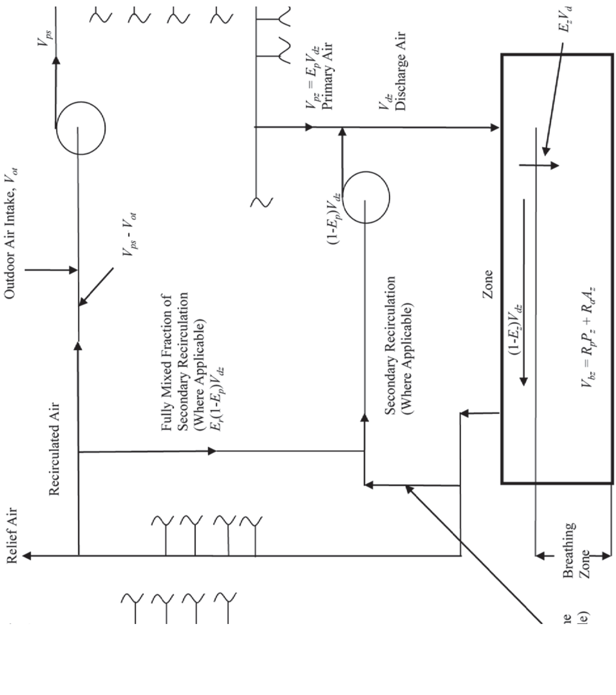


Figure 12.2 Ventilation System Schematic [Std 62.1-2016, Fig A.1]

where the average outdoor air fraction for the system (X_s) is determined in accordance with Equation 12.11, and the primary outdoor air fraction for the zone (Z_{pz}) is determined in accordance with Section 6.2.5.1.

A1.2.2 Secondary Recirculation Systems. For secondary recirculation systems wherein all or part of the supply air to each ventilation zone is recirculated air (air that has not been directly mixed with outdoor air) from other zones, zone ventilation efficiency (E_{vz}) shall be determined in accordance with Equation 12.13. Examples of secondary recirculation systems include dual-fan dual-duct and fan-powered mixing-box systems, and systems that include transfer fans for conference rooms.

$$E_{vz} = (F_a + X_s \times F_b - Z_{pz} \times E_p \times F_c)/F_a \quad (12.13)$$

where system air fractions F_a, F_b, and F_c are determined in accordance with Equations 12.14, 12.15, and 12.16, respectively.

$$F_a = E_p + (1 - E_p) \times E_r \quad (12.14)$$

$$F_b = E_p \quad (12.15)$$

$$F_c = 1 - (1 - E_z) \times (1 - E_r) \times (1 - E_p) \quad (12.16)$$

Where the zone primary air fraction (E_p) is determined in accordance with Equation 12.17, zone secondary recirculation fraction (E_r) is determined by the designer based on system configuration, and zone air distribution effectiveness (E_z) is determined in accordance with Section 6.2.2.2.

$$E_p = V_{pz}/V_{dz} \quad (12.17)$$

where V_{dz} is zone discharge airflow.

Informative Notes:

1. For plenum return systems with secondary recirculation (e.g., fan-powered VAV with plenum return), E_r is usually less than 1.0, although values may range from 0.1 to 1.2, depending upon the location of the ventilation zone relative to other zones and the air handler. For ducted return systems with secondary recirculation (e.g., fan-powered VAV with ducted return), E_r is typically 0.0, while for those with system-level recirculation (e.g, dual-fan dual-duct systems with ducted return), E_r is typically 1.0. For other system types, E_r is typically 0.75.
2. For single-zone and single-supply systems, E_p *is* 1.0.

A1.3 System Ventilation Efficiency. The system ventilation efficiency shall equal the lowest zone ventilation efficiency among all ventilation zones served by the air handler in accordance with Equation 12.18:

$$E_v = \text{minimum } (E_{vz}) \quad (12.18)$$

A2. DESIGN PROCESS

The system ventilation efficiency and, therefore, the outdoor air intake flow for the system (V_{ot}) determined as part of the design process are based on the design and minimum expected supply airflows to individual ventilation zones as well as the design outdoor air requirements to the zones. For VAV system design purposes, zone ventilation efficiency (E_{vz}) for each ventilation zone shall be found using the minimum expected zone primary airflow (V_{pz}) and using the highest expected system primary airflow (V_{ps}) at the design condition analyzed.

Informative Note: Increasing the zone supply airflow values during the design process, particularly to the critical zones requiring the highest fraction of outdoor air, reduces the system outdoor air intake flow requirement determined in the calculation.

A2.1 Selecting Zones for Calculation. Zone ventilation efficiency (E_{vz}) shall be calculated for all ventilation zones.

Exception: Because system ventilation efficiency (E_v) is determined by the minimum value of the zone ventilation efficiency (E_{vz}) in accordance with Equation 12.18, calculation of is not required for any ventilation zone that has an E_{vz} value that is equal to or larger than that of the ventilation zone for which a calculation has been made.

Informative Note: The value of E_{vz} for a ventilation zone will be equal to or larger than that for another ventilation zone if all of the following are true relative to the other ventilation zone:

a. Floor area per occupant (A_z/P_z) is no lower.
b. Minimum zone discharge airflow rate per unit area (V_{dz}/A_z) is no lower.
c. Primary air fraction (E_p) is no lower.
d. Zone air distribution effectiveness (E_z) is no lower.
e. Area outdoor air rate (R_a) is no higher.
f. People outdoor air rate (R_p) is no higher.

A3. SYMBOLS

A_z ***zone floor area:*** the net occupiable floor area of the ventilation zone, ft^2.

D ***occupant diversity:*** the ratio of the system population to the sum of the zone populations.

E_p ***primary air fraction:*** the fraction of primary air in the discharge air to the ventilation zone

E_r ***secondary recirculation fraction:*** in systems with secondary recirculation of return air, the fraction of secondary recirculated air to the zone that is representative of average system return air rather than air directly recirculated from the zone.

E_v ***system ventilation efficiency:*** the efficiency with which the system distributes air from the outdoor air intake to the breathing zone in the ventilation-critical zone, which requires the largest fraction of outdoor air in the primary airstream. E_v shall be determined in accordance with Section 6.2.5.2 or Section A1.

E_{vz} ***zone ventilation efficiency:*** the efficiency with which the system distributes air from the outdoor air intake to the breathing zone in any particular ventilation zone.

E_z ***zone air distribution effectiveness:*** a measure of the effectiveness of supply air distribution to the breathing zone. E_z is determined in accordance with Section 6.2.2.2.

F_a ***supply air fraction:*** the fraction of supply air to the ventilation zone that includes sources of air from outside the zone.

F_b ***mixed-air fraction:*** the fraction of supply air to the ventilation zone from fully mixed primary air.

F_c ***outdoor air fraction:*** the fraction of outdoor air to the ventilation zone that includes sources of air from outside the zone.

P_s ***system population:*** the simultaneous number of occupants in the area served by the ventilation system.

P_z ***zone population:*** see Section 6.2.2.1.

R_a ***area outdoor air rate:*** see Section 6.2.2.1.

R_p ***people outdoor air rate:*** see Section 6.2.2.1.

V_{bz} ***breathing zone outdoor airflow:*** see Section 6.2.2.1.

V_{dz} ***zone discharge airflow:*** the expected discharge (supply) airflow to the zone that includes primary airflow and secondary recirculated airflow, cfm (L/s).

V_{ot} ***outdoor air intake flow:*** see Sections 6.2.3, 6.2.4, and 6.2.5.4.

V_{ou} ***uncorrected outdoor air intake:*** see Section 6.2.5.3.

V_{oz} ***zone outdoor airflow:*** see Section 6.2.2.3.

V_{ps} ***system primary airflow:*** the total primary airflow supplied to all zones served by the system from the air-handling unit at which the outdoor air intake is located.

V_{pz} ***zone primary airflow:*** see Section 6.2.5.1.

X_s ***average outdoor air fraction:*** at the primary air handler, the fraction of outdoor air intake flow in the system primary airflow.

Z_{pz} ***primary outdoor air fraction:*** the outdoor air fraction required in the primary air supplied to the ventilation zone prior to the introduction of any secondary recirculation air.

Table 12.9 Design Parameters [Std 170-2013, Tbl 7.1]

(See complete standard for detailed guidance.)

Function of Space	Pressure Relation-ship to Adjacent Areas (n)	Mini-mum Out-door ach	Mini-mum-Total ach	All Room Air Exhauste d Directly to Out-doors (j)	Air Recircu-lated by Means of Room Units (a)	Design Relative Humidit y (k), %	Design Temp. (l), °F
SURGERY AND CRITICAL CARE							
Operating room (Class B and C) (m), (n), (o)	Positive	4	20	NR	No	20–60	68–75
Operating/surgical cystoscopic rooms, (m), (n) (o)	Positive	4	20	NR	No	20–60	68–75
Delivery room (Caesarean) (m), (n), (o)	Positive	4	20	NR	No	20–60	68–75
Substerile service area	NR	2	6	NR	No	NR	NR
Recovery room	NR	2	6	NR	No	20–60	70–75
Critical and intensive care	NR	2	6	NR	No	30–60	70–75
Intermediate care (s)	NR	2	6	NR	NR	max 60	70–75
Wound intensive care (burn unit)	NR	2	6	NR	No	40–60	70–75
Newborn intensive care	Positive	2	6	NR	No	30–60	72–78
Treatment room (p)	NR	2	6	NR	NR	20–60	70–75
Trauma room (crisis or shock) (c)	Positive	3	15	NR	No	20–60	70–75
Medical/anesthesia gas storage (r)	Negative	NR	8	Yes	NR	NR	NR
Laser eye room	Positive	3	15	NR	No	20–60	70–75
ER waiting rooms	Negative	2	12	Yes (q)	NR	max 65	70–75
Triage	Negative	2	12	Yes (q)	NR	max 60	70–75
ER decontamination	Negative	2	12	Yes	No	NR	NR
Radiology waiting rooms	Negative	2	12	Yes (q), (w)	NR	max 60	70–75
Procedure room (Class A surgery) (o), (d)	Positive	3	15	NR	No	20–60	70–75
Emergency department exam/ treatment room (p)	NR	2	6	NR	NR	max 60	70–75
INPATIENT NURSING							
Patient room	NR	2	4 (y)	NR	NR	max 60	70–75
Nourishment area or room	NR	NR	2	NR	NR	NR	NR
Toilet room	Negative	NR	10	Yes	No	NR	NR
Newborn nursery suite	NR	2	6	NR	No	30–60	72–78
Protective environment room (t)	Positive	2	12	NR	No	max 60	70–75
AII room (u)	Negative	2	12	Yes	No	max 60	70–75
Combination AII/PE room	Positive	2	12	Yes	No	Max 60	70–75
AII anteroom (u)	(e)	NR	10	Yes	No	NR	NR
PE anteroom (t)	(e)	NR	10	NR	No	NR	NR
Combination AII/PE anteroom	(e)	NR	10	Yes	No	NR	NR
Labor/delivery/recovery/ postpartum (LDRP) (s)	NR	2	6	NR	NR	max 60	70–75
Labor/delivery/recovery (LDR) (s)	NR	2	6	NR	NR	max 60	70–75
Patient Corridor	NR	NR	2	NR	NR	NR	NR
NURSING FACILITY							
Resident room	NR	2	2	NR	NR	NR	70–75
Resident gathering/activity/ dining	NR	4	4	NR	NR	NR	70–75
Resident unit corridor	NR	NR	4	NR	NR	NR	NR
Physical therapy	Negative	2	6	NR	NR	NR	70–75
Occupational therapy	NR	2	6	NR	NR	NR	70–75
Bathing room	Negative	NR	10	Yes	No	NR	70–75
RADIOLOGY (v)							
X-ray (diagnostic and treatment)	NR	2	6	NR	NR	max 60	72–78
X-ray (surgery/critical care and catheterization)	Positive	3	15	NR	No	max 60	70–75
Darkroom (g)	Negative	2	10	Yes	No	NR	NR

Table 12.9 Design Parameters *(Continued)* [Std 170-2013, Tbl 7.1] ***(Continued)***
(See complete standard for detailed guidance.)

Function of Space	Pressure Relation-ship to Adjacent Areas (n)	Mini-mum Out-door ach	Mini-mum-Total ach	All Room Air Exhauste d Directly to Out-doors (j)	Air Recircu-lated by Means of Room Units (a)	Design Relative Humidit y (k), %	Design Temp. (l), °F
DIAGNOSTIC AND TREATMENT							
Bronchoscopy, sputum collection, and pentamidine administration (n)	Negative	2	12	Yes	No	NR	68–73
Laboratory, general (v)	Negative	2	6	NR	NR	NR	70–75
Laboratory, bacteriology (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, biochemistry (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, cytology (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, glasswashing	Negative	2	10	Yes	NR	NR	NR
Laboratory, histology (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, microbiology (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, nuclear medicine (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, pathology (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, serology (v)	Negative	2	6	Yes	NR	NR	70–75
Laboratory, sterilizing	Negative	2	10	Yes	NR	NR	70–75
Laboratory, media transfer (v)	Positive	2	4	NR	NR	NR	70–75
Nonrefrigerated body-holding room (h)	Negative	NR	10	Yes	No	NR	70–75
Autopsy room (n)	Negative	2	12	Yes	No	NR	68–75
Pharmacy (b)	Positive	2	4	NR	NR	NR	NR
Examination room	NR	2	6	NR	NR	max 60	70–75
Medication room	NR	2	4	NR	NR	max 60	70–75
Gastrointestinal endoscopy procedure room (x)	NR	2	6	NR	No	20–60	68–73
Endoscope cleaning	Negative	2	10	Yes	No	NR	NR
Treatment room (x)	NR	2	6	NR	NR	max 60	70–75
Hydrotherapy	Negative	2	6	NR	NR	NR	72–80
Physical therapy	Negative	2	6	NR	NR	Max 65	72–80
Dialysis treatment area	NR	2	6	NR	NR	NR	72–78
Dialyzer reprocessing room	Negative	NR	10	Yes	No	NR	NR
Nuclear medicine hot lab	Negative	NR	6	Yes	No	NR	70–75
Nuclear medicine treatment room	Negative	2	6	Yes	NR	NR	70–75
STERILIZING							
Sterilizer equipment room	Negative	NR	10	Yes	No	NR	NR
CENTRAL MEDICAL AND SURGICAL SUPPLY							
Soiled or decontamination room	Negative	2	6	Yes	No	NR	72–78
Clean workroom	Positive	2	4	NR	No	max 60	72–78
Sterile storage	Positive	2	4	NR	NR	max 60	72–78
SERVICE							
Food preparation center (i)	NR	2	10	NR	No	NR	72–78
Warewashing	Negative	NR	10	Yes	No	NR	NR
Dietary storage	NR	NR	2	NR	No	NR	72–78
Laundry, general	Negative	2	10	Yes	No	NR	NR
Soiled linen sorting and storage	Negative	NR	10	Yes	No	NR	NR
Clean linen storage	Positive	NR	2	NR	NR	NR	72–78
Linen and trash chute room	Negative	NR	10	Yes	No	NR	NR
Bedpan room	Negative	NR	10	Yes	No	NR	NR
Bathroom	Negative	NR	10	Yes	No	NR	72–78
Janitor's closet	Negative	NR	10	Yes	No	NR	NR
SUPPORT SPACE							
Soiled workroom or soiled holding	Negative	2	10	Yes	No	NR	NR
Clean workroom or clean holding	Positive	2	4	NR	NR	NR	NR
Hazardous material storage	Negative	2	10	Yes	No	NR	NR

Note: NR = no requirement

Operation and Maintenance

Provide an O&M manual together with final system design drawings, updated and maintained on site.

Table 12.10 Minimum Maintenance Activity and Frequency for Ventilation System Equipment and Associated Components
[Std 62.1-2016, Tbl 8.2]

Inspection/Maintenance Task	Frequency*
a. Investigate system for water intrusion or accumulation. Rectify as necessary.	As necessary
b. Verify that the space provided for routine maintenance and inspection of open cooling tower water systems, closed cooling tower water systems, and evaporative condensers is unobstructed.	Monthly
c. Open cooling tower water systems, closed cooling tower water systems, and evaporative condensers shall be treated to limit the growth of microbiological contaminants, including *legionella sp.*	Monthly
d. Verify that the space provided for routine maintenance and inspection of equipment and components is unobstructed.	Quarterly
e. Check pressure drop and scheduled replacement date of filters and air-cleaning devices. Clean or replace as necessary to ensure proper operation.	Quarterly
f. Check ultraviolet lamp. Clean or replace as needed to ensure proper operation.	Quarterly
g. Visually inspect dehumidification and humidification devices. Clean and maintain to limit fouling and microbial growth. Measure relative humidity and adjust system controls as necessary.	Quarterly
h. Maintain floor drains and trap primer located in air plenums or rooms that serve as air plenums to prevent transport of contaminants from the floor drain to the plenum.	Semiannually
i. Check ventilation and indoor air quality related control systems and devices for proper operation. Clean, lubricate, repair, adjust, or replace as needed to ensure proper operation.	Semiannually
j. Check P-traps in floor drains located in plenums or rooms that serve as air plenums. Prime as needed to ensure proper operation.	Semiannually
k. Check fan belt tension. Check for belt wear and replace if necessary to ensure proper operation. Check sheaves for evidence of improper alignment or evidence of wear and correct as needed.	Semiannually
l. Check variable-frequency drive for proper operation. Correct as needed.	Semiannually
m. Check for proper operation of cooling or heating coil for damage or evidence of leaks. Clean, restore, or replace as required.	Semiannually
n. Visually inspect outdoor air intake louvers, bird screens, mist eliminators, and adjacent areas for cleanliness and integrity; clean as needed; remove all visible debris or visible biological material observed and repair physical damage to louvers, screens, or mist eliminators if such damage impairs the item from providing the required outdoor air entry.	Semiannually
o. Visually inspect natural ventilation openings and adjacent areas for cleanliness and integrity; clean as needed. Remove all visible debris or visible biological material observed and repair physical damage to louvers, and screens if such damage impairs the item from providing the required outdoor air entry. Manual and/or automatic opening apparatus shall be physically tested for proper operation and repaired or replaced as necessary.	Semiannually

Table 12.10 Minimum Maintenance Activity and Frequency for Ventilation System Equipment and Associated Components
[Std 62.1-2016, Tbl 8.2] ***(Continued)***

Inspection/Maintenance Task	Frequency*
p. Verify the operation of the outdoor air ventilation system and any dynamic minimum outdoor air controls.	Annually
q. Check air filter fit and housing seal integrity. Correct as needed.	Annually
r. Check control box for dirt, debris, and/or loose terminations. Clean and tighten as needed.	Annually
s. Check motor contactor for pitting or other signs of damage. Repair or replace as needed.	Annually
t. Check fan blades and fan housing. Clean, repair, or replace as needed to ensure proper operation.	Annually
u. Check integrity of all panels on equipment. Replace fasteners as needed to ensure proper integrity and fit/finish of equipment.	Annually
v. Assess field serviceable bearings. Lubricate if necessary.	Annually
w. Check drain pans, drain lines, and coils for biological growth. Check adjacent areas for evidence of unintended wetting. Repair and clean as needed.	Annually
x. Check for evidence of buildup or fouling on heat exchange surfaces. Restore as needed to ensure proper operation.	Annually
y. Inspect unit for evidence of moisture carryover from cooling coils beyond the drain pan. Make corrections or repairs as necessary.	Annually
z. Check for proper damper operation. Clean, lubricate, repair, replace, or adjust as needed to ensure proper operation.	Annually
aa. Visually inspect areas of moisture accumulation for biological growth. If present, clean or disinfect as needed.	Annually
ab. Check condensate pump. Clean or replace as needed.	Annually
ac. Visually inspect exposed ductwork and external piping for insulation and vapor barrier for integrity. Correct as needed.	Annually
ad. Verify the accuracy of permanently mounted sensors whose primary function is outdoor air delivery monitoring, outdoor air delivery verification, or dynamic minimum outdoor air control, such as flow stations at an air handler and those used for demand-control ventilation. A sensor failing to meet the accuracy specified in the O&M manual shall be recalibrated or replaced. Performance verification shall include output comparison to a measurement reference standard consistent with those specified for similar devices in ASHRAE Standard 41.2 or ASHRAE Standard 111.	5 years
ae. Verify the total quantity of outdoor air delivered by air handlers set to minimum outdoor air mode. If measured minimum airflow rates are less than the design minimum rate documented in the O&M manual, ± a 10% balancing tolerance, (1) confirm the measured rate does not conform with the provisions of this standard and (2) adjust or modify the air-handler components to correct the airflow deficiency. Ventilation systems shall be balanced in accordance with ASHRAE Standard 111 or its equivalent, at least to the extent necessary to verify conformance with the total outdoor airflow and space supply airflow requirements of this standard. **Exception:** Units under 2000 cfm of supply air are exempt from this requirement.	5 years

* Minimum frequencies may be increased or decreased if indicated in the O&M manual.

** National Institute of Standards and Technology, U.S. Department of Commerce, Gaithersburg, MD.

13. ENERGY-CONSERVING DESIGN

Sustainability

Recognition of the impact of the building industry's activities on the earth's ecosystem is changing the way it approaches the design, construction, operation, maintenance, reuse, and demolition of what it creates—namely addressing the environmental and long-term economic consequences of its actions. While this *sustainable design* ethic—or *sustainability*—covers things beyond the purview of HVAC&R, design for the efficient use of energy resources is a key element of sustainable design.

The basic approach to energy-efficient design is reducing loads and required power, improving transport systems, and providing efficient components and "intelligent" controls. This includes understanding the relationship between energy and power, maintaining simplicity, using self-imposed budgets, and applying energy-smart design practices.

An example of a budget designers have set for themselves for office buildings in a typical mid-USA climate:

Installed lighting	0.8 W/ft^2	Thermal power	20 Btu/h·ft^2
Space sensible cooling	15 Btu/h·ft^2	Hydronic system head	65 ft water
Space heating load	10 Btu/h·ft^2	Water chiller (water-cooled)	0.50 kW/ton
Fan system pressure	3.0 in. water	Chilled-water auxiliaries	0.12 kW/ton
Air circulation	1 cfm/ft^2	Annual electric energy	15 kW/ft^2·y
Overall electric power	3.0 W/ft^2	Annual thermal energy	5 Btu/ft^2·y°F·day

Then, as design proceeds, compare with budget:

1. Minimize impact of building's functional requirements—to reduce, redistribute, or shift (delay) loads.
2. Minimize loads—look at peak and part-load operation.
3. Maximize subsystem efficiency—including opportunities to reclaim, redistribute, and store energy for future use.
4. Study alternative ways to integrate subsystems into the building—use easily understood design solutions to foster simplicity of operation.

HVAC&R System Design

- Consider separate systems to serve areas expected to operate on widely different schedules or design conditions.
- Arrange systems so spaces with relatively constant and weather-independent loads are served by systems separate from systems serving perimeter spaces.
- Sequence supply of cooling and heating to prevent simultaneous operation of heating and cooling systems to the same space.
- Provide controls to allow operation in an occupied mode and an unoccupied mode.
- Where diurnal temperature swings and humidity levels permit, consider coupling air distribution and building mass to allow nighttime cooling to reduce requirement of daytime mechanical cooling.
- Where climate allows, consider mixed-mode systems of HVAC and natural ventilation.
- Select energy conversion devices matched to load increments.
- Select the most efficient equipment practical at both design and part-load operating conditions.
- Seriously consider life-cycle purchasing technique for large power devices.
- Transport energy by the most energy-efficient means.
- Provide intelligent control system that provides information to operators and managers.

Summary

In designing HVAC&R systems, the need to address immediate issues such as economics, performance, and space constraints should not prevent designers from fully considering different energy sources. Consider the viability and dependability of energy resources for the long-term operation of the building. Energy standards and legislation represent only the minimum that can be achieved; strive to better utilize energy.

Energy Efficiency Standards

ANSI/ASHRAE/IES Standard 90.1-2016, *Energy Standard for Buildings Except Low-Rise Residential Buildings*

The standard includes minimum energy efficiency requirements for new buildings or portions of buildings and their systems; new systems and equipment in existing buildings.

There is a strong move to provide considerably more energy savings than required by this standard. The standard has had frequent addenda and is revised every three years.

Prescriptive Path for Compliance Highlights

Section 5. Building Envelope: Tables in the standard cover nine climate zones (see Figure 13.1) for nonresidential, residential, and semiheated occupancies for minimum allowable insulating value of envelope elements and maximum allowable solar heat gain of fenestration.

Section 6. HVAC; minimum equipment efficiencies. Required controls. Allowable fan power for supply air systems. Hydronic systems pump power. Heat rejection equipment fans. Exhaust energy recovery. Heat recovery systems for service water heating systems. Radiant heating for unenclosed spaces.

Section 7. Service water heating—minimum equipment efficiencies.

Section 8. Power: Feeder conductors sized for maximum voltage drop of 2% at design load; branch circuit conductors 3%.

Section 9. Lighting: Limitations on lighting power densities, controls required.

Section 10. Other equipment: Minimum allowable electric motor efficiencies.

Normative Appendix A. Rated R-value of insulation and assembly U-factor, C-factor, and F-factor determinations.

Normative Appendix C. Methodology for building envelope trade-off option in Section 5.6.

Alternative to Prescriptive Methods of Compliance

Section 11. Energy cost budget method.

Normative Appendix G. Performance rating method.

ASHRAE Standard 90.2-2007, *Energy Efficient Design of Low-Rise Residential Buildings*

Prescriptive minimum requirements for envelope and equipment with alternate annual energy cost method of compliance.

ASHRAE Standard 100-2006, *Energy Conservation in Existing Buildings*

A building or a complex of buildings complies when the following requirements have been met and recorded on Form A of the standard and the party determining compliance has (1) conducted an energy survey as required by the standard in Section 5, (2) stated in writing that the operation and maintenance requirements in Section 5 have been met, and (3) has stated in writing that building and equipment modifications in Section 7 have been met.

More stringent and more detailed requirements can be expected in future editions of the standard.

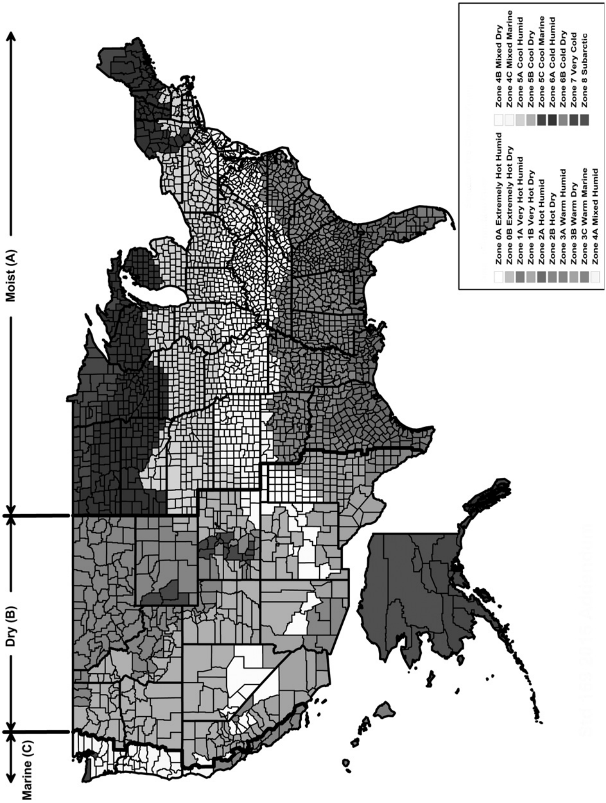
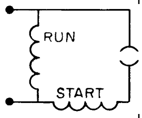
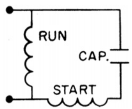
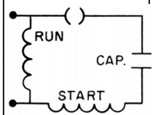
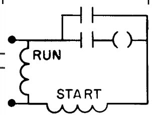
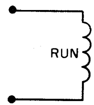
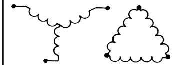
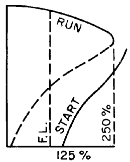
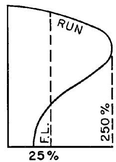
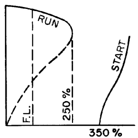
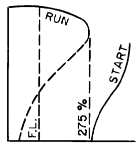
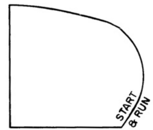


Figure 13.1 Climate Zones for United States Locations [Std 90.1-2016, Fig Annex1-1]

14. ELECTRICAL

Table 14.1 Characteristics of AC Motors (Nonhermetic) [2016S, Ch 45, Tbl 4]

	Split-Phase	Permanent Split-Capacitor	Capacitor-Start Induction-Run	Capacitor-Start Capacitor-Run	Shaded-Pole	Polyphase, 60-Hz
Connection Diagram						
Speed Torque Curves						
Starting Method	Centrifugal switch	None	Centrifugal switch	Centrifugal switch	None	Motor controller
Ratings, hp	0.05 to 0.5	0.05 to 5	0.05 to 5	0.05 to 5	0.01 to 0.25	0.5 and up
Full-Load Speeds at 60-Hz (Two-Pole, Four-Pole)	3450 to 1725	3450 to 1725	3450 to 1725	3500 to 1750	3100 to 1550	3500 to 1750
Torque[a] **Locked Rotor** **Breakdown**	 125 to 150% 250 to 300%	 25% 250 to 300%	 250 to 350% 250 to 300%	 250% 250%	 25% 125%	 150 to 350% 250 to 350%
Speed Classification	Constant	Constant	Constant	Constant	Constant or adjustable	Constant
Full-Load Power Factor	60%	95%	65%	95%	60%	80%
Efficiency	Medium	High	Medium	High	Low	High-Medium

[a] Expressed as percent of rated horsepower torque.

Table 14.2 Motor Full-Load Amperes

Horse power	Recommended Starter Size Three Phase 230 V	460 V	Three-Phase AC Squirrel-Cage and Wound-Rotor (Induction Type) 200 V	230 V	460 V	Recommended Starter Size Single Phase 230 V	Single-Phase AC 115 V	200 V	230 V	Horse power
1/6						00	4.4	2.5	2.2	1/6
1/4						00	5.8	3.3	2.9	1/4
1/2	00	00	2.3	2	1	00	9.8	5.6	4.9	1/2
3/4	00	00	3.2	2.8	1.4	00	13.8	7.9	6.9	3/4
1	00	00	4.1	3.6	1.8	00	16	9.2	8	1
1.5	00	00	6.0	5.2	2.6	0	20	11.5	10	1.5
2	0	00	7.8	6.8	3.4	0	24	13.8	12	2
3	0	0	11.0	9.6	4.8	1	34	19.6	17	3
5	1	0	17.5	15.2	7.6	1	56	32.2	28	5
7.5	1	1	25.3	22	11	2	80	46	40	7.5
10	2	1	32.2	28	14	2	100	57.5	50	10
15	2	2	48.3	42	21	3				15
20	3	2	62.1	54	27					20
25	3	2	78.2	68	34					25
30	3	3	92	80	40					30
40	4	3	119.6	104	52					40
50	4	3	149.5	130	65					50
60	5	4	177.1	154	77					60
75	5	4	220.8	192	96					75
100	5	4	285.2	248	124					100
125	6	5	358.8	312	156					125
150	6	5	414	360	180					150
200	6	5	552	480	240					200

Values are for motors with normal torque characteristics running at usual belted speeds.

Table 14.3 Useful Electrical Formulas

To Find	Direct Current	Single Phase	Three Phase
Amperes when horsepower known	$\frac{hp \times 746}{E \times \eta}$	$\frac{hp \times 746}{E \times \eta \times F}$	$\frac{hp \times 746}{1.73 \times E \times \eta \times F}$
Amperes when kilowatts known	$\frac{kW \times 1000}{E}$	$\frac{kW \times 1000}{E \times F}$	$\frac{kW \times 1000}{1.73 \times E \times F}$
Amperes when kVA known		$\frac{kVA \times 1000}{E}$	$\frac{kVA \times 1000}{1.73 \times E}$
Kilowatts	$\frac{I \times E}{1000}$	$\frac{I \times E \times F}{1000}$	$\frac{I \times E \times 1.73 \times F}{1000}$
kVA		$\frac{I \times E}{1000}$	$\frac{I \times E \times 1.73}{1000}$
Horsepower—(output)	$\frac{I \times E \times \eta}{746}$	$\frac{I \times E \times \eta \times F}{746}$	$\frac{I \times E \times 1.73 \times \eta \times F}{746}$

I = amperes; E = volts; η = efficiency expressed as decimal; F = power factor; kW = kilowatts; kVA = kilovolt-amperes; hp = horsepower.

Motor Controllers

Three-phase constant-speed induction motor controllers are usually full-voltage except when the starting current must be reduced in larger motors to meet power system limitations; such motor controllers may be of various row types. All are used for starting and stopping the motor and include overcurrent protection.

Variable-Speed Drives (VSDs)

By far the most energy-efficient means of varying flow of fans and pumps driven by electric motors are VSDs. Their application involves careful consideration of their effects (here VSD is considered synonymous with variable-frequency drive [VFD], pulse-width modulated drive [PWM drive], adjustable-speed drive [ASD], and adjustable-frequency drive [AFD].) A VSD consists of a pulse-width-modulation controller with insulated-gate bipolar transistors (IGBTs) and an induction motor. The IGBT changes the characteristics of waveforms applied to a motor due to the speed at which the IGBT cycles on and off. At switching speed up to 20 k Hz, the impedance in the connecting cable is far less than the motor impedance, particularly for small motors, causing pulse reflectance at the motor terminals to form damaging high voltage. NEMA motor standard MG1 states PWM drive limits and establishes a peak of 1600 V and a minimum rise time of 0.1 μs for motors rated less than 600 V. Typical manufacturer maximum voltage withstand levels range from 1000 V to 1800 V. When specifying motors for operation on VSDs, the voltage withstand level based on the dV/dt of the drive and the known cable distance should be specified.

Harmonics caused by the portion of a VSD converting line power LDC affect input lines and are termed *line-side harmonics*. Output line harmonics are caused solely by the inverter section of the VSD and are known as load side or motor harmonics. Generally, PWM drives containing internal bus reaction or three-phase AC line reactors do not cause interference with other electrical equipment. There may be problems when a VSD is switched onto a standby generator, or when power factor correction capacitors are used.

15. FUELS AND COMBUSTION

Table 15.1 Maximum Capacity of Gas Pipe in Cubic Feet per Hour
[2017F, Ch 22, Tbl 40]

Nominal Iron Pipe Size, in.	Internal Diameter, in.	Length of Pipe, ft													
		10	20	30	40	50	60	70	80	90	100	125	150	175	200
1/4	0.364	32	22	18	15	14	12	11	11	10	9	8	8	7	6
3/8	0.493	72	49	40	34	30	27	25	23	22	21	18	17	15	14
1/2	0.622	132	92	73	63	56	50	46	43	40	38	34	31	28	26
3/4	0.824	278	190	152	130	115	105	96	90	84	79	72	64	59	55
1	1.049	520	350	285	245	215	195	180	170	160	150	130	120	110	100
1 1/4	1.380	1,050	730	590	500	440	400	370	350	320	305	275	250	225	210
1 1/2	1.610	1,600	1,100	890	760	670	610	560	530	490	460	410	380	350	320
2	2.067	3,050	2,100	1,650	1,450	1,270	1,150	1,050	990	930	870	780	710	650	610
2 1/2	2.469	4,800	3,300	2,700	2,300	2,000	1,850	1,700	1,600	1,500	1,400	1,250	1,130	1,050	980
3	3.068	8,500	5,900	4,700	4,100	3,600	3,250	3,000	2,800	2,600	2,500	2,200	2,000	1,850	1,700
4	4.026	17,500	12,000	9,700	8,300	7,400	6,800	6,200	5,800	5,400	5,100	4,500	4,100	3,800	3,500

Note: Capacity is in cubic feet per hour at gas pressures of 0.5 psig or less and a pressure drop of 0.3 in. of water; specific gravity = 0.60.

Table 15.2 Typical API Gravity, Density, and Heating Value of Standard Grades of Fuel Oil [2017F, Ch 28, Tbl 10]

Grade No.	API Gravity	Density, lb/gal	Heating Value, Btu/gal
1	38 to 45	6.950 to 6.675	137,000 to 132,900
2	30 to 38	7.296 to 6.960	141,800 to 137,000
4	20 to 28	7.787 to 7.396	148,100 to 143,100
5L	17 to 22	7.940 to 7.686	150,000 to 146,800
5H	14 to 18	8.080 to 7.890	152,000 to 149,400
6	8 to 15	8.448 to 8.053	155,900 to 151,300

Types of Fuel Oils

Fuel oils for heating are broadly classified as distillate fuel oils (lighter oils) or residual fuel oils (heavier oils). ASTM has established specifications for fuel oil properties which subdivide the oils into various grades. Grades No. 1 and 2 are distillate fuel oils. Grades 4,5 (Light), 5 (Heavy), and 6 are residual fuel oils. Specifications for the grades are based on required characteristics of fuel oils for use in different types of burners. The ANSI standard specification for fuel oils is ASTM Standard D396.

Grade No. 1 is a light distillate intended for vaporizing-type burners. High volatility is essential to continued evaporation of the fuel oil with minimum residue.

Grade No. 2 is a heavier (API Gravity) distillate than No. 1. It is used primarily with pressure-atomizing (gun) burners that spray the oil into a combustion chamber. The atomized oil vapor mixed with air and burns. This grade is used in most domestic burners and many medium capacity commercial-industrial burners.

Grade No. 4 is an intermediate fuel that is considered either a light residual or a heavy distillate. Intended for burners that atomize oils of higher viscosity than domestic burners can handle, its permissible viscosity range allows it to be pumped and atomized at relatively low storage temperatures.

Grade No. 5 (Light) is a residual fuel of intermediate viscosity for burners that handle fuel more viscous than No. 4 without preheating. Preheating may be necessary in some equipment for burning and, in colder climates, for handling.

Grade No. 5 (Heavy) is a residual fuel more viscous than No. 5 (Light), but intended for similar purposes. Preheating is usually necessary for burning and, in colder climates, for handling.

Grade No. 6, sometimes referred to as Bunker C, is a high-viscosity oil used mostly in commercial and industrial heating. It requires preheating in the storage tank to permit pumping, and additional preheating at the burner to permit atomizing.

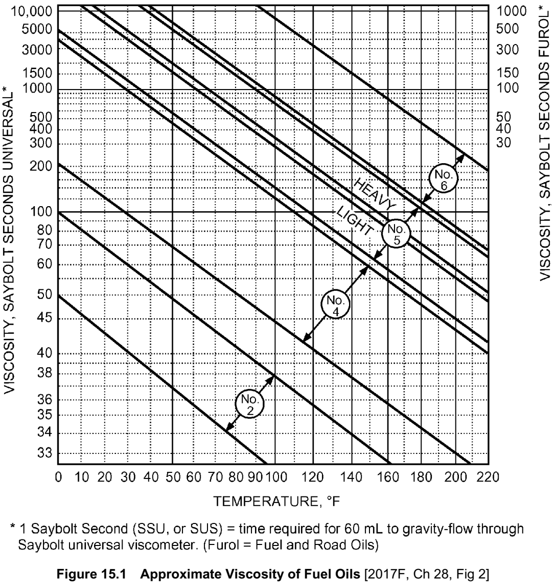


Figure 15.1 Approximate Viscosity of Fuel Oils [2017F, Ch 28, Fig 2]

Types and Properties of Liquid Fuels for Engines

The primary stationary engine fuels are diesel and gas turbine oils, natural gases, and liquefied petroleum gases. Other fuels include sewage gas, manufactured gas, and gas mixtures. Gasoline and the JP series of gas turbine fuels are rarely used for stationary engines.

Properties of the three grades of diesel fuel oils (1-D, 2-D, and 4-D) are listed in ASTM Standard D975.

Grade No. 2-D includes the class of lower volatility distillate gas oils. These fuels are used in high-speed engines with relatively high loads and uniform speeds, or in engines not requiring fuels with the higher volatility or other properties specified for Grade No. 1-D.

Grade No. 4-D covers the class of more viscous distillates and blends of these distillates with residual fuel oils. These fuels are used in low- and medium-speed engines involving sustained loads at essentially constant speed.

Property specifications and test methods for Grade No. 1-D, 2-D, and 4-D diesel fuel oils are essentially identical to specifications of Grade No. 1, 2, and 4 fuel oils, respectively. However, diesel fuel oils have an additional specification for **cetane number**, which measures ignition quality and influences combustion roughness. Cetane number requirements depend on engine design, size, speed and load variations, and starting and atmospheric conditions. An increase in cetane number over values actually required does not improve engine performance. Thus, the cetane number should be as low as possible to assure maximum fuel availability. ASTM Standard D975 provides several methods for estimating cetane number from other fuel oil properties.

ASTM Standard D2880 for gas turbine fuel oils relates gas turbine fuel oil grades to fuel and diesel fuel oil grades.

Table 15.3 Approximate Air Requirements for Stoichiometric Combustion of Various Fuels [2017F, Ch 28, Tbl 13]

Type of Fuel	Theoretical Air Required for Combustion
Solid fuels	lb/lb fuel
Anthracite	9.6
Semibituminous	11.2
Bituminous	10.3
Lignite	6.2
Coke	11.2
Liquid fuels	lb/gal fuel
No. 1 fuel oil	103
No. 2 fuel oil	106
No. 5 fuel oil	112
No. 6 fuel oil	114
Gaseous fuels	ft^3/ft^3 fuel
Natural gas	9.6
Butane	31.1
Propane	24.0

Table 15.4 Approximate Maximum Theoretical (Stoichiometric) CO_2 Values and CO_2 Values of Various Fuels with Different Percentages of Excess Air [2017F, Ch 28, Tbl 13]

Type of Fuel	Theoretical or Maximum CO_2, %	Percent CO_2 at Given Excess Air Values 20%	40%	60%
Gaseous Fuels				
Natural gas	12.1	9.9	8.4	7.3
Propane gas (commercial)	13.9	11.4	9.6	8.4
Butane gas (commercial)	14.1	11.6	9.8	8.5
Mixed gas (natural and carbureted water gas)	11.2	12.5	10.5	9.1
Carbureted water gas	17.2	14.2	12.1	10.6
Coke oven gas	11.2	9.2	7.8	6.8
Liquid Fuels				
No. 1 and 2 fuel oil	15.0	12.3	10.5	9.1
No. 6 fuel oil	16.5	13.6	11.6	10.1
Solid Fuels				
Bituminous coal	18.2	15.1	12.9	11.3
Anthracite	20.2	16.8	14.4	12.6
Coke	21.0	17.5	15.0	13.0

Table 15.5 Recommended Nominal Size for Fuel Oil Suction Lines from Tank to Pump (Distillate Grades No. 1 and No. 2) [2017F, Ch 22, Tbl 42]

Pumping Rate, gph	Length of Run in Feet at Maximum Suction Lift of 10 ft: 25	50	75	100	125	150	175	200	250	300
10	1/2	1/2	1/2	1/2	1/2	1/2	1/2	3/4	3/4	1
40	1/2	1/2	1/2	1/2	1/2	3/4	3/4	3/4	3/4	1
70	1/2	1/2	3/4	3/4	3/4	3/4	3/4	1	1	1
100	1/2	3/4	3/4	3/4	3/4	1	1	1	1	1 1/4
130	1/2	3/4	3/4	1	1	1	1	1	1 1/4	1 1/4
160	3/4	3/4	3/4	1	1	1	1	1 1/4	1 1/4	1 1/4
190	3/4	3/4	1	1	1	1	1 1/4	1 1/4	1 1/4	2
220	3/4	1	1	1	1	1 1/4	1 1/4	1 1/4	1 1/4	2

Table 15.6 Recommended Nominal Size for Fuel Oil Suction Lines from Tank to Pump (Residual Grades No. 5 and No. 6) [2017F, Ch 22, Tbl 41]

Pumping Rate, gph	Length of Run in Feet at Maximum Suction Lift of 15 ft: 25	50	75	100	125	150	175	200	250	300
10	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	1 1/2	2	2	2 1/2	2 1/2
40	1 1/2	1 1/2	1 1/2	2	2	2 1/2	2 1/2	2 1/2	2 1/2	3
70	1 1/2	2	2	2	2	2 1/2	2 1/2	2 1/2	3	3
100	2	2	2	2 1/2	2 1/2	3	3	3	3	3
130	2	2	2 1/2	2 1/2	2 1/2	3	3	3	3	4
160	2	2	2 1/2	2 1/2	2 1/2	3	3	3	4	4
190	2	2 1/2	2 1/2	2 1/2	3	3	3	4	4	4
220	2 1/2	2 1/2	2 1/2	3	3	3	4	4	4	4

Notes: 1. Pipe sizes smaller than 1 in. IPS are not recommended for use with residual grade fuel oils.
2. Lines conveying fuel oil from pump discharge port to burners and tank return may be reduced by one or two sizes, depending on piping length and pressure losses.

16. OWNING AND OPERATING

Maintenance Costs

The maintenance cost of mechanical systems varies widely depending on configuration, equipment locations, accessibility, system complexity, service duty, geography, and system reliability requirements.

Dohrmann and Alereza (1986) obtained maintenance costs and HVAC system information from 342 buildings located in 35 states in the United States. In 1983 U.S. dollars, data collected showed a mean HVAC system maintenance cost of $0.32/ft^2 per year, with a median cost of $0.24/ft^2 per year. Building age has a statistically significant but minor effect on HVAC maintenance costs. Analysis also indicated that building size is not statistically significant in explaining cost variation. The type of maintenance program or service agency that building management chooses can also have a significant effect on total HVAC maintenance costs. Although extensive or thorough routine and preventive maintenance programs cost more to administer, they usually extend equipment life; improve reliability; and reduce system downtime, energy costs, and overall life-cycle costs.

Some maintenance cost data are available, both in the public domain and from proprietary sources used by various commercial service providers. These sources may include equipment manufacturers, independent service providers, insurers, government agencies (e.g., the U.S. General Services Administration), and industry-related organizations [e.g., the Building Owners and Managers Association (BOMA)] and service industry publications. More traditional, widely used products and components are likely to have statistically reliable records. However, design changes or modifications necessitated by industry changes, such as alternative refrigerants, may make historical data less relevant.

Newer HVAC products, components, system configurations, control systems and protocols, and upgraded or revised system applications present an additional challenge. Care is required when using data not drawn from broad experience or field reports. In many cases, maintenance information is proprietary or was sponsored by a particular entity or group. Particular care should be taken when using such data. It is the user's responsibility to obtain these data and to determine their appropriateness and suitability for the application being considered.

ASHRAE research project TRP-1237 (Abramson et al. 2005) developed a standardized Internet-based data collection tool and database on HVAC equipment service life and maintenance costs. The database was seeded with data on 163 buildings from around the country. Maintenance cost data were gathered for total HVAC system maintenance costs from 100 facilities. In 2004 dollars, the mean HVAC maintenance cost from these data was $0.47/ft^2, and the median cost was $0.44/ft^2. Table 16.1 compares these figures with estimates reported by Dohrmann and Alereza (1986), both in terms of contemporary dollars, and in 2004 dollars, and shows that the cost per square foot varies widely between studies.

Estimating Maintenance Costs

Total HVAC maintenance cost for new and existing buildings with various types of equipment may be estimated several ways, using several resources. Equipment maintenance requirements can be obtained from the equipment manufacturers for large or custom pieces of equipment. Estimating in-house labor requirements can be difficult; BOMA provides guidance on this topic. Many independent mechanical service companies provide preventative maintenance contracts. These firms typically have proprietary estimating programs developed through their experience, and often provide generalized maintenance costs to engineers and owners upon request, without obligation.

Table 16.1 Comparison of Maintenance Costs Between Studies
[2015A, Ch 37, Tbl 6]

Survey	Cost per ft^2, as Reported		Consumer Price Index	Cost per ft^2, 2004 Dollars	
	Mean	Median		Mean	Median
Dohrmann and Alereza (1986)	$0.32	$0.24	99.6	$0.61	$0.46
Abramson et al. (2005)	$0.47	$0.44	188.9	$0.47	$0.44

When evaluating various HVAC systems during design or retrofit, the absolute magnitude of maintenance costs may not be as important as the relative costs. Whichever estimating method or resource is selected, it should be used consistently throughout any evaluation. Mixing information from different resources in an evaluation may provide erroneous results.

Applying simple costs per unit of building floor area for maintenance is highly discouraged. Maintenance costs can be generalized by system types. When projecting maintenance costs for different HVAC systems, the major system components need to be identified with a required level of maintenance. The potential long-term costs of environmental issues on maintenance costs should also be considered.

Table 16.2 Owning and Operating Cost Data and Summary [2015A, Ch 37, Tbl 1]

OWNING COSTS			
I. Initial Cost of System		_______	
II. Periodic Costs			
A. Income taxes	_______		
B. Property taxes	_______		
C. Insurance	_______		
D. Rent	_______		
E. Other periodic costs	_______		
Total Periodic Costs		_______	
III. Replacement Cost		_______	
IV. Salvage Value		_______	
Total Owning Costs			_______
OPERATING COSTS			
V. Annual Utility, Fuel, Water, etc., Costs			
A. Utilities			
1. Electricity	_______		
2. Natural gas	_______		
3. Water/sewer	_______		
4. Purchased steam	_______		
5. Purchased hot/chilled water	_______		
B. Fuels			
1. Propane	_______		
2. Fuel oil	_______		
3. Diesel	_______		
4. Coal	_______		
C. On-site generation of electricity	_______		
D. Other utility, fuel, water, etc., costs	_______		
Total		_______	
VI. Annual Maintenance Allowances/Costs			
A. In-house labor	_______		
B. Contracted maintenance service	_______		
C. In-house materials	_______		
D. Other maintenance allowances/costs (e.g., water treatment)	_______		
Total		_______	
VII. Annual Administration Costs		_______	
Total Annual Operating Costs			_______
TOTAL ANNUAL OWNING AND OPERATING COSTS			_______

Simple Payback

This ignores inflation and the time value of money. The annual revenue stream cost savings and other factors are estimated and divided into the initial capital outlay; the result is the simple payback time in years.

Life-Cycle Costs

A representation in present dollars of the cost of an investment over its lifetime is useful for evaluating mutually exclusive alternatives that have the same anticipated lifetime.

A *discount rate* is required for a life-cycle-cost calculation. The discount rate represents the cost of capital to building owners. In essence, it is the rate on a loan (or bond) adjusted to account for inflation and taxes. A 3% real discount rate is typical for energy policy analyses. Higher rates are often used by private investors for economic evaluation of commercial construction. To account for inflation and fuel escalation, either lower the discount rate or inflate future energy and maintenance costs.

Life-cycle cost is calculated by determining the present worth of the cost of an investment. For system alternatives, it is

$$\mathrm{LCC} = \mathrm{IC} + \mathrm{ESPWF}(\mathrm{COST}_{energy} + \mathrm{COST}_{maint}) \qquad (16.1)$$

where

LCC	=	life-cycle cost
IC	=	initial cost premium of alternative
ESPWF	=	equal series present worth factor (see Table 16.3)
COST_{energy}	=	yearly energy cost saving
COST_{maint}	=	yearly maintenance cost reduction

ESPWF for other lifetimes and discount rates can be calculated from

$$\mathrm{ESPWF} = \frac{(1+d)^n - 1}{d(1+d)^n} \qquad (16.2)$$

where n = lifetime in years and d = discount rate in percent/100.

Note that ESPWF can only be used when annual costs remain constant.

Capital Recovery Factors

The future equal payments to repay a present value of money is determined by the capital recovery factor, which is the reciprocal of the present worth factor for a series of equal payments.

$$\mathrm{CRF} = \frac{i(1+i)^n}{(1+i)^n - 1} = \frac{i}{1-(1+i)^n} \qquad (16.3)$$

where i is the compound interest rate.

Improved Payback Analysis

Similar to simple payback but cost of money is considered.

$$n = \frac{\ln[\mathrm{CRF}/(i - \mathrm{CRF})]}{\ln(1+i)} \qquad (16.4)$$

Table 16.3 Equal Series Present Worth Factors (ESPWFs)

Lifetime (years)	Discount Rate							
	2.5%	3.0%	3.5%	4.0%	4.5%	7%	10%	15%
7	6.35	6.23	6.11	6.00	5.89	5.39	4.8	4.16
10	8.75	8.53	8.32	8.11	7.91	7.02	6.14	5.02
15	12.38	11.94	11.52	11.12	10.74	9.11	7.61	5.85

Table 16.4 Annual Capital Recovery Factors [2003A, Ch 36, Tbl 5]

Years	Rate of Return or Interest Rate, % per Year				
	3.5	4.5	6	8	10
2	0.52640	0.53400	0.54544	0.56077	0.57619
4	0.27225	0.27874	0.28859	0.30192	0.31547
6	0.18767	0.19388	0.20336	0.21632	0.22961
8	0.14548	0.15161	0.16104	0.17401	0.18744
10	0.12024	0.12638	0.13587	0.14903	0.16275
12	0.10348	0.10967	0.11928	0.13270	0.14676
14	0.09157	0.09782	0.10758	0.12130	0.13575
16	0.08268	0.08902	0.09895	0.11298	0.12782
18	0.07582	0.08224	0.09236	0.10670	0.12193

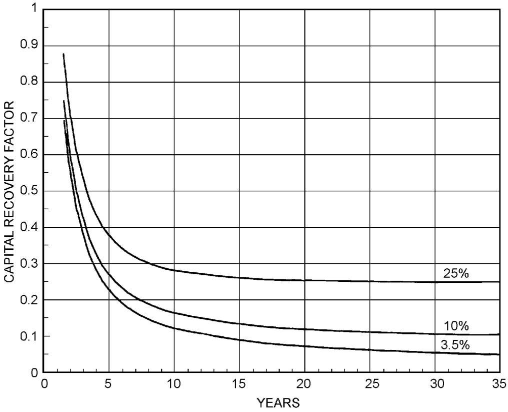


Figure 16.1 Capital Recovery Factor Versus Time [2003A, Ch 36, Fig 1]

17. SOUND

Sound Pressure and Sound Pressure Level

Sound intensity is difficult to measure directly, but sound pressure is relatively easy to measure; the human ear and microphones are pressure-sensitive devices. A decibel scale for sound pressure can be created in a manner analogous to the decibel scale for sound intensity, with a reference pressure of 20 μPa, which corresponds to the approximate threshold of hearing. Because pressure squared is proportional to intensity, sound pressure level is

$$L_p = 10 \log(p/p_{ref})^2 \text{ re } p_{ref} \tag{17.1}$$

Because p/p_{ref} is 20 μPa, which is 2×10^{-5} Pa, and since $10 \log p^2 = 20 \log p$,

$$L_p = 20 \log(p/2 \times 10^{-5}) \text{ re } 20\ \mu\text{Pa} \tag{17.2}$$

where p is the root mean square (rms) value of pressure in micropascals. Or

$$L_p = 20 \log p + 94 \text{ db re } 20\ \mu\text{Pa} \tag{17.3}$$

The human ear responds across a broad range of sound pressures. The linear range scale for sound pressure in Table 17.1 is awkward in this form; therefore, the equivalent logarithmic notations should be used.

Table 17.1 Typical Sound Pressures and Sound Pressure Levels
[2017F, Ch 8, Tbl 1]

Source	Sound Pressure, Pa	Sound Pressure Level, dB re 20 μPa	Subjective Reaction
Military jet takeoff at 100 ft	200	140	Extreme danger
Artillery fire at 10 ft	63.2	130	
Passenger jet takeoff at 50 ft	20	120	Threshold of pain
Loud rock band	6.3	110	Threshold of discomfort
Automobile horn at 10 ft	2	100	
Unmuffled large diesel engine at 130 ft	0.6	90	Very loud
Accelerating diesel truck at 50 ft	0.2	80	
Freight train at 100 ft	0.06	70	
Conversational speech at 3 ft	0.02	60	
Window air conditioner at 3 ft	0.006	50	Moderate
Quiet residential area	0.002	40	
Whispered conversation at 6 ft	0.0006	30	
Buzzing insect at 3 ft	0.0002	20	
Threshold of good hearing	0.00006	10	Faint
Threshold of excellent youthful hearing	0.00002	0	Threshold of hearing

Combining Sound Levels

To estimate the levels from multiple sources from the levels from each source, the intensities (not the levels) must be added. Thus, the levels must first be converted to find intensities, the intensities summed, and then converted to a level again, so the combination of multiple levels L_1, L_2, etc., produces a level L_{sum} given by

$$L_{sum} = 10 \log \sum_i 10^{L_i/10} \tag{17.4}$$

where for sound pressure level (L_p), $10^{L_i/10}$ is p_i^2/p_{ref}^2, and L_i is the sound pressure level for the ith source.

A simpler and slightly less accurate method is outlined in Table 17.2. This method, although not exact, results in errors of 1 dB or less. The process with a series of levels may be shortened by combining the largest with the next largest, then combining this sum with the third largest, then the fourth largest, and so on until the combination of the remaining levels is 10 dB lower than the combined level. The process may then be stopped.

Sound Power and Sound Power Level

A fundamental characteristic of an acoustic source is its ability to radiate energy. Some energy input excites the source, which radiates some fraction of this energy in the form of sound. Since unit power radiated through a unit sphere yields unit intensity, the power reference base, established by international agreement, is 1 picowatt (pW) (10^{-12} W). The reference quantity used should be stated explicitly. A definition of sound power level is, therefore

$$L_w = \log w/(10^{-12}\text{W}) \text{ dB re 1 pW} \quad \text{or} \quad L_w = 10 \log w + 120 \text{ dB re 1 pW} \tag{17.5}$$

Table 17.2 Combining Two Sound Levels [2017F, Ch 8, Tbl 3]

Difference between levels to be combined, dB	0 to 1	2 to 4	5 to 9	10 and More
Number of decibels to add to highest level to obtain combined level	3	2	1	0

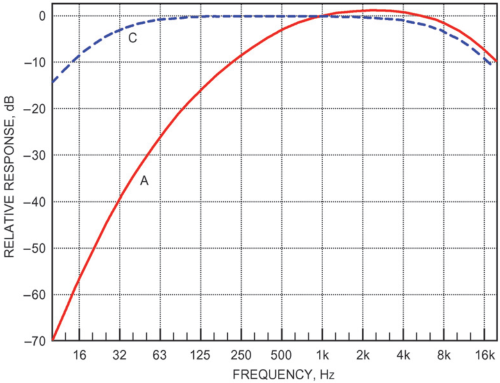


Figure 17.1 Curves Showing A- and C-Weighting Responses for Sound Level Meters [2017F, Ch 8, Fig 1]

Table 17.3 Midband and Approximate Upper and Lower Cutoff Frequencies for Octave and 1/3 Octave Band Filters [2017F, Ch 8, Tbl 4]

Octave Bands, Hz			1/3 Octave Bands, Hz		
Lower	**Midband**	**Upper**	**Lower**	**Midband**	**Upper**
			11.2	12.5	14
11.2	16	22.4	14	16	18
			18	20	22.4
			22.4	25	28
22.4	31.5	45	28	31.5	35.5
			35.5	40	45
			45	50	56
45	63	90	56	63	71
			71	80	90
			90	100	112
90	125	180	112	125	140
			140	160	180
			180	200	224
180	250	355	224	250	280
			280	315	355
			355	400	450
355	500	710	450	500	560
			560	630	710
			710	800	900
710	1,000	1,400	900	1,000	1,120
			1,120	1,250	1,400
			1,400	1,600	1,800
1,400	2,000	2,800	1,800	2,000	2,240
			2,240	2,500	2,800
			2,800	3,150	3,550
2,800	4,000	5,600	3,550	4,000	4,500
			4,500	5,000	5,600
			5,600	6,300	7,100
5,600	8,000	11,200	7,100	8,000	9,000
			9,000	10,000	11,200
			11,200	12,500	14,000
11,200	16,000	22,400	14,000	16,000	18,000
			18,000	20,000	22,400

Table 17.4 Design Guidelines for HVAC-Related Background Sound in Rooms [2015A, Ch 48, Tbl 1]

Room Types		Octave Band Analysis[a] NC/RC[b]	Approximate Overall Sound Pressure Level[a] dBA[c]	dBC[c]
Rooms with Intrusion from Outdoor Noise Sources[d]	Traffic noise	N/A	45	70
	Aircraft flyovers	N/A	45	70
Residences, Apartments, Condominiums	Living areas	30	35	60
	Bathrooms, kitchens, utility rooms	35	40	60
Hotels/Motels	Individual rooms or suites	30	35	60
	Meeting/banquet rooms	30	35	60
	Corridors and lobbies	40	45	65
	Service/support areas	40	45	65
Office Buildings	Executive and private offices	30	35	60
	Conference rooms	30	35	60
	Teleconference rooms	25	30	55
	Open-plan offices	40	45	65
	Corridors and lobbies	40	45	65
Courtrooms	Unamplified speech	30	35	60
	Amplified speech	35	40	60
Performing Arts Spaces	Drama theaters, concert and recital halls	20	25	50
	Music teaching studios	25	30	55
	Music practice rooms	30	35	60
Hospitals and Clinics	Patient rooms	30	35	60
	Wards	35	40	60
	Operating and procedure rooms	35	40	60
	Corridors and lobbies	40	45	65
Laboratories	Testing/research w/minimal speech communication	50	55	75
	Extensive phone use and speech communication	45	50	70
	Group teaching	35	40	60
Churches, Mosques, Synagogues	General assembly with critical music programs[e]	25	30	55
Schools[f]	Classrooms	30	35	60
	Large lecture rooms with speech amplification	30	35	60
	Large lecture rooms without speech amplification	25	30	55
Libraries		30	35	60
Indoor Stadiums, Gymnasiums	Gymnasiums and natatoriums[g]	45	50	70
	Large-seating-capacity spaces with speech amplification[g]	50	55	75

N/A = Not applicable

[a]Values and ranges are based on judgment and experience, and represent general limits of acceptability for typical building occupancies.

[b]NC: this metric plots octave band sound levels against a family of reference curves, with the number rating equal to the highest tangent line value.

RC: when sound quality in the space is important, the RC metric provides a diagnostic tool to quantify both the speech interference level and spectral imbalance.

[c]dBA and dBC: these are overall sound pressure level measurements with A- and C-weighting, and serve as good references for a fast, single-number measurement. They are also appropriate for specification in cases where no octave band sound data are available for design.

[d]Intrusive noise is addressed here for use in evaluating possible non-HVAC noise that is likely to contribute to background noise levels.

[e]An experienced acoustical consultant should be retained for guidance on acoustically critical spaces (below RC 30) and for all performing arts spaces.

[f]Some educators and others believe that HVAC-related sound criteria for schools, as listed in previous editions of this table, are too high and impede learning for affected groups of all ages. See ANSI/ASA Standard S12.60 for classroom acoustics and a justification for lower sound criteria in schools. The HVAC component of total noise meets the background noise requirement of that standard if HVAC-related background sound is approximately NC/RC 25. Within this category, designs for K-8 schools should be quieter than those for high schools and colleges.

[g]RC or NC criteria for these spaces need only be selected for the desired speech and hearing conditions.

Table 17.5 Comparison of Sound Rating Methods [2015A, Ch 48, Tbl 4]

Method	Overview	Considers Speech Interference Effects	Evaluates Sound Quality	Components Presently Rated by Each Method
dBA	No quality assessment Frequently used for outdoor noise ordinances	Yes	No	Cooling towers Water chillers Condensing units
NC	Can rate components Limited quality assessment Does not evaluate low-frequency rumble	Yes	Somewhat	Air terminals Diffusers
RC Mark II	Used to evaluate systems Should not be used to evaluate components Evaluates sound quality Provides improved diagnostics capability	Yes	Yes	Not used for component rating
NCB	Can rate components Some quality assessment	Yes	Somewhat	See NC
RNC	Some quality assessment Attempts to quantify fluctuations	Yes	Somewhat	Not used for component rating

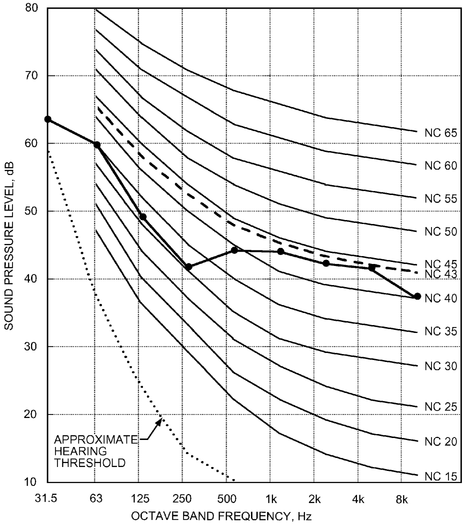
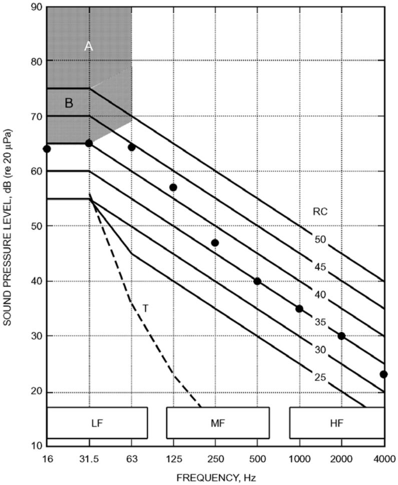


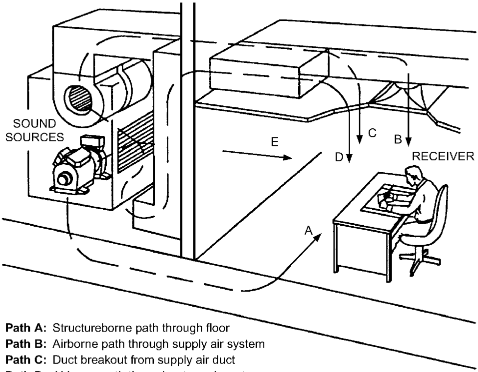
Figure 17.2 NC (Noise Criteria) Curves and Typical Spectrum (Curve with Symbols)
[2017F, Ch 8, Fig 7]

Sound



Note:
- Noise levels for lightweight wall and ceiling constructions:
  – In shaded region B are likely to generate vibration that may be perceptible. There is a slight possibility of rattles in light fixtures, doors, windows, etc.
  – In shaded region A have a high probability of generating easily perceptible noise- induced vibration. Audible rattling in light fixtures, doors, windows, etc. may be anticipated.
- Regions LF, MF, and HF are explained in the text.
- Solid dots are sound pressure levels for the example discussed in the text.

Figure 17.3 Room Criteria Curves, Mark II [2015A, Ch 48, Fig 6]



Path A: Structureborne path through floor
Path B: Airborne path through supply air system
Path C: Duct breakout from supply air duct
Path D: Airborne path through return air system
Path E: Airborne path through mechanical equipment room wall

Figure 17.4 Typical Paths of Noise and Vibration Propagation in HVAC Systems
[2015A, Ch 48, Fig 1]

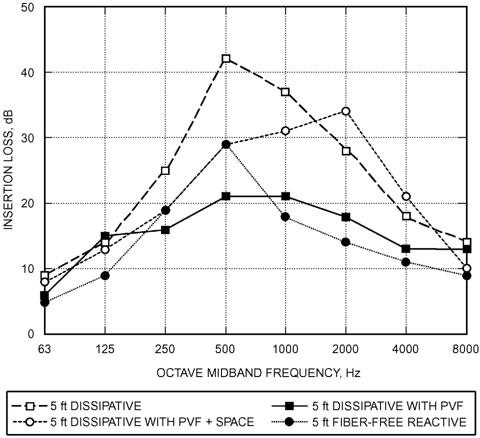
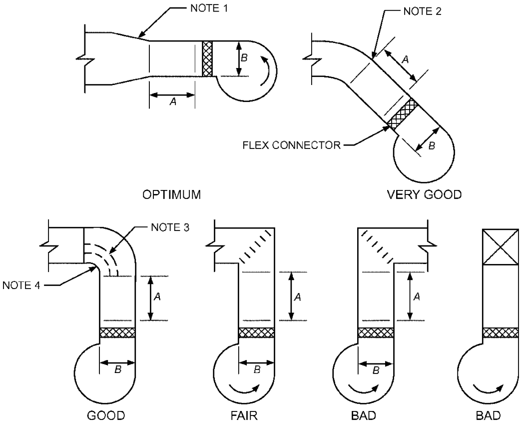


Figure 17.5 Comparison of 5 ft Dissipative and Reactive Silencer Performance—Film Liner to Conform to NFPA 90A [2015A, Ch 48, Fig 23]



Notes:

1. Slopes of 1 in 7 preferred. Slopes of 1 in 4 permitted below 2000 fpm.
2. Dimension *A* should be at least 1.5 times *B*, where *B* is largest discharge duct dimension.
3. Rugged turning vanes should extend full radius of elbow.
4. Minimum 6 in. radius required.

Figure 17.6 Various Outlet Configurations for Centrifugal Fans and Their Possible Rumble Conditions [2015A, Ch 48, Fig 25]

A number of AHRI, AMCA, CTI, and ANSI sound standards are used by equipment manufacturers to provide accurate sound data. Manufacturer-supplied data in accordance with the appropriate standard should be used in preference to any earlier empirical information in evaluating the noise resulting from a particular equipment item.

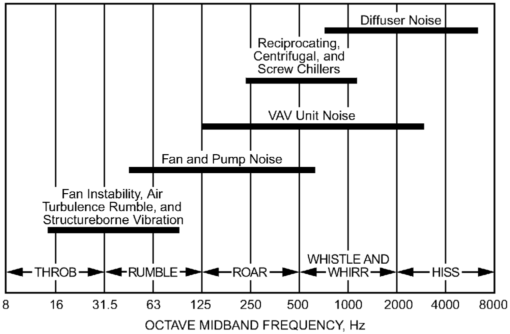


Figure 17.7 Frequencies at Which Different Types of Mechanical Equipment Generally Control Sound Spectra [2015A, Ch 48, Fig 4]

Table 17.6 Sound Transmission Class (STC) and Transmission Loss Values of Typical Mechanical Equipment Room Wall, Floor, and Ceiling Types, dB
[2015A, Ch 48, Tbl 40]

		Octave Midband Frequency, Hz						
Room Construction Type	**STC**	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**
8 in. CMU*	50	35	35	41	44	50	57	64
8 in. CMU with 5/8 in. GWB* on furring strips	53	33	32	44	50	56	59	65
5/8 in. GWB on both sides of 3 5/8 in. metal studs	38	18	16	33	47	55	43	47
5/8 in. GWB on both sides of 3 5/8 in. metal studs with fiberglass insulation in cavity	49	16	23	44	58	64	52	53
2 layers of 5/8 in. GWB on both sides of 3 5/8 in. metal studs with fiberglass insulation in cavity	56	19	32	50	62	67	58	63
Double row of 3 5/8 in. metal studs, 1 in. apart, each with 2 layers of 5/8 in. GWB and fiberglass insulation in cavity	64	23	40	54	62	71	69	74
6 in. solid concrete floor/ceiling	53	40	40	40	49	58	67	76
6 in. solid concrete floor with 4 in. isolated concrete slab and fiberglass insulation in cavity	72	44	52	58	73	87	97	100
6 in. solid concrete floor with two layers of 5/8 in. GWB hung on spring isolators with fiberglass insulation in cavity	84	53	63	70	84	93	104	105

Note: Actual material composition (e.g., density, porosity, stiffness) affects transmission loss and STC values.

*CMU = concrete masonry unit; GWB = gypsum wallboard.

Table 17.7 Sound Sources, Transmission Paths, and Recommended Noise Reduction Methods [2015A, Ch 48, Tbl 6]

Sound Source	Path No.
Circulating fans; grilles; registers; diffusers; unitary equipment in room	1
Induction coil and fan-powered VAV mixing units	1, 2
Unitary equipment located outside of room served; remotely located air-handling equipment, such as fans, blowers, dampers, duct fittings, and air washers	2, 3
Compressors, pumps, and other reciprocating and rotating equipment (excluding air-handling equipment)	4, 5, 6
Cooling towers; air-cooled condensers	4, 5, 6, 7
Exhaust fans; window air conditioners	7, 8
Sound transmission between rooms	9, 10

No.	Transmission Paths	Noise Reduction Methods
1	Direct sound radiated from sound source to ear	Direct sound can be controlled only by selecting quiet equipment.
	Reflected sound from walls, ceiling, and floor	Reflected sound is controlled by adding sound absorption to the room and to equipment location.
2	Air- and structureborne sound radiated from casings and through walls of ducts and plenums is transmitted through walls and ceiling into room	Design duct and fittings for low turbulence; locate high-velocity ducts in noncritical areas; isolate ducts and sound plenums from structure with neoprene or spring hangers.
3	Airborne sound radiated through supply and return air ducts to diffusers in room and then to listener by Path 1	Select fans for minimum sound power; use ducts lined with sound-absorbing material; use duct silencers or sound plenums in supply and return air ducts.
4	Noise transmitted through equipment room walls and floors to adjacent rooms	Locate equipment rooms away from critical areas; use masonry blocks or concrete for equipment room walls and floor.
5	Vibration transmitted via building structure to adjacent walls and ceilings, from which it radiates as noise into room by Path 1	Mount all machines on properly designed vibration isolators; design mechanical equipment room for dynamic loads; balance rotating and reciprocating equipment.
6	Vibration transmission along pipes and duct walls	Isolate pipe and ducts from structure with neoprene or spring hangers; install flexible connectors between pipes, ducts, and vibrating machines.
7	Noise radiated to outside enters room windows	Locate equipment away from critical areas; use barriers and covers to interrupt noise paths; select quiet equipment.
8	Inside noise follows Path 1	Select quiet equipment.
9	Noise transmitted to an air diffuser in a room, into a duct, and out through an air diffuser in another room	Design and install duct attenuation to match transmission loss of wall between rooms.
10	Sound transmission through, over, and around room partition	Extend partition to ceiling slab and tightly seal all around; seal all pipe, conduit, duct, and other partition penetrations.

18. VIBRATION

$$\text{Natural frequency,} \quad f_n = \frac{1}{2\pi}\sqrt{\frac{k}{M}} \tag{18.1}$$

where k is the stiffness of vibration isolator (force per unit deflection) and M is the mass of equipment supported by the isolator.

$$f_n = \frac{3.13}{\sqrt{\delta_{st}}}\text{Hz} \tag{18.2}$$

where δ_{st} is the static deflection of the isolator in inches.

Transmissibility is the ratio of the amplitudes of the force transmitted to the building structure to the exciting force produced by the vibrating equipment. Transmissibility is inversely proportional to the square of the disturbing frequency f_d to the natural frequency f_n.

$$T = \left[\frac{1}{1-(f_d/f_n)^2}\right] \tag{18.3}$$

At $f_d = f_n$, resonance occurs. Vibration isolation is effective only at a f_d/f_n ratio > 3.5.

When supporting structure stiffness is not large with respect to stiffness of isolator, it becomes a two-degree of freedom system. In this case, choose an isolator that will provide static deflection eight to ten times that of the estimated floor static deflection due to the added weight of the equipment. Seismic snubbers must be included in or with isolators to limit equipment movement.

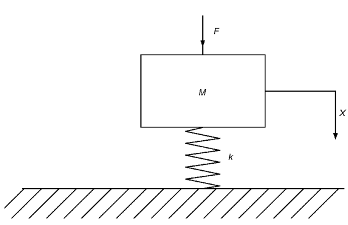


Figure 18.1 Single-Degree-of-Freedom System [2017F, Ch 8, Fig 8]

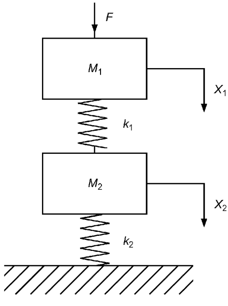


Figure 18.2 Two-Degree-of-Freedom System [2017F, Ch 8, Fig 11]

Table 18.1 Selection Guide for Vibration Isolation [2015A, Ch 48, Tbl 47]

			Equipment Location (Notes for Table 18.1, Item 1)												
						Floor Span									
			Slab on Grade			Up to 20 ft			20 to 30 ft			30 to 40 ft			
Equipment Type	Horse-power and Other	RPM	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Reference Notes
Refrigeration Machines and Chillers															
Water-cooled reciprocating	All	All	A	2	0.25	A	4	0.75	A	4	1.50	A	4	2.50	2,3,12
Water-cooled centrifugal, scroll	All	All	A	1	0.25	A	4	0.75	A	4	1.50	A	4	1.50	2,3,4,8,12
Water-cooled screw	All	All	A	1	1.00	A	4	1.5	A	4	2.50	A	4	2.50	2,3,4,12
Absorption	All	All	A	4	0.25	A	4	0.75	A	4	1.50	A	4	1.50	
Air-cooled recip., scroll	All	All	A	1	0.25	A	4	1.50	A	4	1.50	A	4	2.50	2,4,5,12
Air-cooled screw	All	All	A	4	1.00	A	4	1.50	B	4	2.50	B	4	2.50	2,4,5,8,12
Air Compressors and Vacuum Pumps															
Tank-mounted horiz.	≤10	All	A	3	0.75	A	3	0.75	A	3	1.50	A	3	1.50	3,15
	≥10	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,15
Tank-mounted vert.	All	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,15
Base-mounted	All	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,14,15
Large reciprocating	All	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,14,15
Pumps															
Close-coupled	≤7.5	All	B	2	0.25	C	3	0.75	C	3	0.75	C	3	0.75	16
	≥7.5	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	16
Inline	5 to 25	All	A	3	0.75	A	3	1.50	A	3	1.50	A	3	1.50	
	≥25	All	A	3	1.50	A	3	1.50	A	3	1.50	A	3	2.50	
End suction and double-suction/split case	≤40	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	16
	50 to 125	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	2.50	10,16
	≥150	All	C	3	0.75	C	3	1.50	C	3	2.50	C	3	3.50	10,16
Packaged pump systems	All	All	A	3	0.75	A	3	0.75	A	3	1.50	C	3	2.50	

Table 18.1 Selection Guide for Vibration Isolation [2015A, Ch 48, Tbl 47] **(Continued)**

			Equipment Location (Notes for Table 18.1, Item 1)												
						Floor Span									
			Slab on Grade			Up to 20 ft			20 to 30 ft			30 to 40 ft			
Equipment Type	**Horse-power and Other**	**RPM**	**Base Type**	**Isolator Type**	**Min. Defl., in.**	**Base Type**	**Isolator Type**	**Min. Defl., in.**	**Base Type**	**Isolator Type**	**Min. Defl., in.**	**Base Type**	**Isolator Type**	**Min. Defl., in.**	**Reference Notes**
Cooling Towers	All	Up to 300	A	1	0.25	A	4	3.50	A	4	3.50	A	4	3.50	5,8,18
		301 to 500	A	1	0.25	A	4	2.50	A	4	2.50	A	4	2.50	5,18
		501 and up	A	1	0.25	A	4	0.75	A	4	0.75	A	4	0.75	5,18
Boilers															
Fire-tube	All	All	A	1	0.25	B B	4	0.75	B	4	1.50	B	4	2.50	4
Water-tube, copper fin	All	All	A	1	0.12	A	1	0.12	A	1	0.12	B	4	0.25	
Axial Fans, Plenum Fans, Cabinet Fans, Fan Sections, Centrifugal Inline Fans															
Up to 22 in. diameter	All	All	A	2	0.25	A	3	0.75	A	3	0.75	C	3	0.75	4,9,8
24 in. diameter and up	≤2 in. SP	Up to 300	B	3	2.50	C	3	3.50	C	3	3.50	C	3	3.50	9,8
		301 to 500	B	3	0.75	B	3	1.50	C	3	2.50	C	3	2.50	9,8
		501 and up	B	3	0.75	B	3	1.50	B	3	1.50	B	3	1.50	9,8
		Up to 300	C	3	2.50	C	3	3.50	C	3	3.50	C	3	3.50	3,8,9
	≥2.1 in. SP	301 to 500	C	3	1.50	C	3	1.50	C	3	2.50	C	3	2.50	3,8,9
		501 and up	C	3	0.75	C	3	1.50	C	3	1.50	C	3	2.50	3,8,9
Centrifugal Fans															
Up to 22 in. diameter	All	All	B	2	0.25	B	3	0.75	B	3	0.75	B	3	1.50	9,19
24 in. diameter and up	≤40	Up to 300	B	3	2.50	B	3	3.50	B	3	3.50	B	3	3.50	8,19
		301 to 500	B	3	1.50	B	3	1.50	B	3	2.50	B	3	2.50	8,19
		501 and up	B	3	0.75	B	3	0.75	B	3	0.75	B	3	1.50	8,19
	≥50	Up to 300	C	3	2.50	C	3	3.50	C	3	3.50	C	3	3.50	2,3,8,9,19
		301 to 500	C	3	1.50	C	3	1.50	C	3	2.50	C	3	2.50	2,3,8,9,19
		501 and up	C	3	1.00	C	3	1.50	C	3	1.50	C	3	2.50	2,3,8,9,19
Propeller Fans															
Wall-mounted	All	All	A	1	0.25	A	1	0.25	A	1	0.25	A	1	0.25	
Roof-mounted	All	All	A	1	0.25	A	1	0.25	B	4	1.50	D	4	1.50	

Table 18.1 Selection Guide for Vibration Isolation [2015A, Ch 48, Tbl 47] **(Continued)**

			Equipment Location (Notes for Table 18.1, Item 1)												
						Floor Span									
			Slab on Grade			Up to 20 ft			20 to 30 ft			30 to 40 ft			
Equipment Type	Horse-power and Other	RPM	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Reference Notes
Heat Pumps, Fan-Coils, Computer Room Units	All	All	A	3	0.75	A	3	0.75	A	3	0.75	A/D	3	1.50	
Condensing Units	All	All	A	1	0.25	A	4	0.75	A	4	1.50	A/D	4	1.50	
Packaged AH, AC, H, and V Units															
All	≤10	All	A	3	0.75	A	3	0.75	A	3	0.75	A	3	0.75	19
	≤15	Up to 300	A	3	0.75	A	3	3.50	A	3	3.50	C	3	3.50	2,4,8,19
		301 to 500	A	3	0.75	A	3	2.50	A	3	2.50	A	3	2.50	4,19
	≤4 in. SP	501 and up	A	3	0.75	A	3	1.50	A	3	1.50	A	3	1.50	4,19
	>15, >4 in. SP	Up to 300	B	3	0.75	C	3	3.50	C	3	3.50	C	3	3.50	2,3,4,8,9
		301 to 500	B	3	0.75	C	3	1.50	C	3	2.50	C	3	2.50	2,3,4,9
		501 and up	B	3	0.75	C	3	1.50	C	3	1.50	C	3	2.50	2,3,4,9
Packaged Rooftop Equipment	All	All	A/D	1	0.25	D	3	0.75	See Reference Note 17						5,6,8,17
Ducted Rotating Equipment															
Small fans, fan-powered boxes	≤600 cfm		A	3	0.50	A	3	0.50	A	3	0.50	A	3	0.50	7
	≥601 cfm		A	3	0.75	A	3	0.75	A	3	0.75	A	3	0.75	7
Engine-Driven Generators	All	All	A	3	0.75	C	3	1.50	C	3	2.50	C	3	3.50	2,3,4
Piping and Ducts (See sections on Isolating Vibration and Noise in Piping Systems and Isolating Duct Vibration for isolator selection.)															

Base Types:
A. No base, isolators attached directly to equipment (Note 28)
B. Structural steel rails or base (Notes 29 and 30)
C. Concrete inertia base (Note 31)
D. Curb-mounted base (Note 32)

Isolator Types:
1. Pad, rubber, or glass fiber (Notes 20 and 21)
2. Rubber floor isolator or hanger (Notes 20 and 25)
3. Spring floor isolator or hanger (Notes 22, 23, and 26)
4. Restrained spring isolator (Notes 22 and 24)
5. Thrust restraint (Note 27)
6. Air spring (Note 25)

Notes for Table: Selection Guide for Vibration Isolation

These notes are keyed to the column titled Reference Notes in 2015A, Ch 48, Tbl 47 and to other reference numbers throughout the table. Although the guide is conservative, cases may arise where vibration transmission to the building is still excessive. If the problem persists after all short circuits have been eliminated, it can almost always be corrected by altering the support path (e.g., from ceiling to floor), increasing isolator deflection, using low-frequency air springs, changing operating speed, improving rotating component balancing, or, as a last resort, changing floor frequency by stiffening or adding more mass. Assistance from a qualified vibration consultant can be very useful in resolving these problems.

Note 1. Isolator deflections shown are based on a reasonably expected floor stiffness according to floor span and class of equipment. Certain spaces may dictate higher levels of isolation. For example, bar joist roofs may require a static deflection of 1.5 in. over factories, but 2.5 in. over commercial office buildings.

Note 2. For large equipment capable of generating substantial vibratory forces and structureborne noise, increase isolator deflection, if necessary, so isolator stiffness is less than one-tenth the stiffness of the supporting structure, as defined by the deflection due to load at the equipment support.

Note 3. For noisy equipment adjoining or near noise-sensitive areas, see the section on Mechanical Equipment Room Sound Isolation.

Note 4. Certain designs cannot be installed directly on individual isolators (type A), and the equipment manufacturer or a vibration specialist should be consulted on the need for supplemental support (base type).

Note 5. Wind load conditions must be considered. Restraint can be achieved with restrained spring isolators (type 4), supplemental bracing, snubbers, or limit stops. Also see Chapter 55 of the 2015 *ASHRAE Handbook—HVAC Applications*.

Note 6. Certain types of equipment require a curb-mounted base (type D). Airborne noise must be considered.

Note 7. See section on Resilient Pipe Hangers and Supports for hanger locations adjoining equipment and in equipment rooms.

Note 8. To avoid isolator resonance problems, select isolator deflection so that resonance frequency is 40% or less of the lowest normal operating speed of equipment (see Chapter 8 in the 2013 *ASHRAE Handbook—Fundamentals*). Some equipment, such as variable-frequency drives, and high-speed equipment, such as screw chillers and vaneaxial fans, contain very-high-frequency vibration. This equipment creates new technical challenges in the isolation of high-frequency noise and vibration from a building's structure. Structural resonances both internal and external to the isolators can significantly degrade their performance at high frequencies. Unfortunately, at present no test standard exists for measuring the high-frequency dynamic properties of isolators, and commercially available products are not tested to determine their effectiveness for high frequencies. To reduce the chance of high-frequency vibration transmission, add a minimum 0.75 in. thick elastomeric pad (type 1, Note 20) to the base plate of spring isolators (type 3, Note 22, 23, 24). For some sensitive locations, air springs (Note 25) may be required. If equipment is located near extremely noise-sensitive areas, follow the recommendations of an acoustical consultant.

Note 9. To limit undesirable movement, thrust restraints (type 5) are required for all ceiling-suspended and floor-mounted units operating at 2 in. of water or more total static pressure.

Note 10. Pumps over 75 hp may need extra mass and restraints.

Note 11. See text for full discussion.

Isolation for Specific Equipment

Note 12. Refrigeration Machines: Large centrifugal, screw, and reciprocating refrigeration machines may generate very high noise levels; special attention is required when such equipment is installed in upper-story locations or near noise-sensitive areas. If equipment is located near extremely noise-sensitive areas, follow the recommendations of an acoustical consultant.

Note 13. Compressors: The two basic reciprocating compressors are (1) single- and double-cylinder vertical, horizontal or L-head, which are usually air compressors; and (2) Y, W, and multihead or multicylinder air and refrigeration compressors. Single- and double-cylinder compressors generate high vibratory forces requiring large inertia bases (type C) and are generally not suitable for upper-story locations. If this equipment must be installed in an upper-story location or at-grade location near noise-sensitive areas, the expected maximum unbalanced force data must be obtained from the equipment manufacturer and a vibration specialist consulted for design of the isolation system.

Note 14. Compressors: When using Y, W, and multihead and multicylinder compressors, obtain the magnitude of unbalanced forces from the equipment manufacturer so the need for an inertia base can be evaluated.

Note 15. Compressors: Base-mounted compressors through 5 hp and horizontal tank-type air compressors through 10 hp can be installed directly on spring isolators (type 3) with structural bases (type B) if required, and compressors 15 to 100 hp on spring isolators (type 3) with inertia bases (type C) weighing 1 to 2 times the compressor weight.

Note 16. Pumps: Concrete inertia bases (type C) are preferred for all flexible-coupled pumps and are desirable for most close-coupled pumps, although steel bases (type B) can be used. Close-coupled pumps should not be installed directly on individual isolators (type A) because the impeller usually overhangs the motor support base, causing the rear mounting to be in tension. The primary requirements for type C bases are strength and shape to accommodate base elbow supports. Mass is not usually a factor, except for pumps over 75 hp, where extra mass helps limit excess movement due to starting torque and forces. Concrete bases (type C) should be designed for a thickness of one-tenth the longest dimension with minimum thickness as follows: (1) for up to 30 hp, 6 in.; (2) for 40 to 75 hp, 8 in.; and (3) for 100 hp and up, 12 in.

Pumps over 75 hp and multistage pumps may exhibit excessive motion at start-up (“heaving”); supplemental restraining devices can be installed if necessary. Pumps over 125 hp may generate high starting forces; consult a vibration specialist.

Note 17. Packaged Rooftop Air-Conditioning Equipment: This equipment is usually installed on lightweight structures that are susceptible to sound and vibration transmission problems. The noise problems are compounded further by curb-mounted equipment, which requires large roof openings for supply and return air.

The table shows type D vibration isolator selections for all spans up to 20 ft, but extreme care must be taken for equipment located on spans of over 20 ft, especially if construction is open web joists or thin, lightweight slabs. The recommended procedure is to determine the additional deflection caused by equipment in the roof. If additional roof deflection is 0.25 in. or less, the isolator should be selected for up to 10 times the additional roof deflection. If additional roof deflection is over 0.25 in., supplemental roof stiffening should be installed to bring the roof deflection down below 0.25 in., or the unit should be relocated to a stiffer roof position.

For mechanical units capable of generating high noise levels, mount the unit on a platform above the roof deck to provide an air gap (buffer zone) and locate the unit away from the associated roof penetration to allow acoustical treatment of ducts before they enter the building.

Some rooftop equipment has compressors, fans, and other equipment isolated internally. This isolation is not always reliable because of internal short-circuiting, inadequate static deflection, or panel resonances. It is recommended that rooftop equipment over 300 lb be isolated externally, as if internal isolation was not used.

Note 18. Cooling Towers: These are normally isolated with restrained spring isolators (type 4) directly under the tower or tower dunnage. High-deflection isolators proposed for use directly under the motor-fan assembly must be used with extreme caution to ensure stability and safety under all weather conditions. See Note 5.

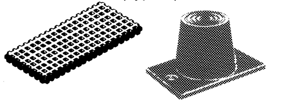
Note 19. Fans and Air-Handling Equipment: Consider the following in selecting isolation systems for fans and air-handling equipment:

1. Fans with wheel diameters of 22 in. and less and all fans operating at speeds up to 300 rpm do not generate large vibratory forces. For fans operating under 300 rpm, select isolator deflection so the isolator natural frequency is 40% or less than the fan speed. For example, for a fan operating at 275 rpm, 0.4 ´ 275 = 110 rpm. Therefore, an isolator natural frequency of 110 rpm or lower is required. This can be accomplished with a 3 in. deflection isolator (type 3).
2. Flexible duct connectors should be installed at the intake and discharge of all fans and air-handling equipment to reduce vibration transmission to air duct structures.
3. Inertia bases (type C) are recommended for all class 2 and 3 fans and air-handling equipment because extra mass allows the use of stiffer springs, which limit heaving movements.
4. Thrust restraints (type 5) that incorporate the same deflection as isolators should be used for all fan heads, all suspended fans, and all base-mounted and suspended air-handling equipment operating at 2 in. or more total static pressure. Restraint movement adjustment must be made under normal operational static pressures.

Vibration Isolators: Materials, Types, and Configurations

Notes 20 through 32 include figures to assist in evaluating commercially available isolators for HVAC equipment. The isolator selected for a particular application depends on the required deflection, life, cost, and compatibility with associated structures.

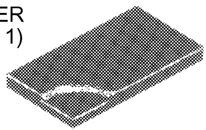
RUBBER PADS (Type 1)



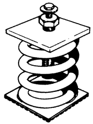
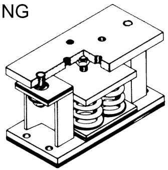
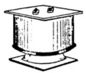
RUBBER MOUNTS (Type 2)

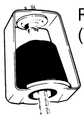
Note 20. Rubber isolators are available in pad (type 1) and molded (type 2) configurations. Pads are used in single or multiple layers. Molded isolators come in a range of 30 to 70 durometer (a measure of stiffness). Material in excess of 70 durometer is usually ineffective because durometers are not a measure of stiffness of an isolator. Isolators are designed for up to 0.5 in. deflection, but are used where 0.3 in. or less deflection is required. Solid rubber and composite fabric and rubber pads are also available. They provide high load capacities with small deflection and are used as noise barriers under columns and for pipe supports. These pad types work well only when they are properly loaded and the weight load is evenly distributed over the entire pad surface. Metal loading plates can be used for this purpose.

GLASS FIBER PADS (Type 1)

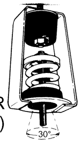


Note 21. Glass fiber with elastic coating (type 1). This type of isolation pad is precompressed molded fiberglass pads individually coated with a flexible, moisture-impervious elastomeric membrane. Natural frequency of fiberglass vibration isolators should be essentially constant for the operating load range of the supported equipment. Weight load is evenly distributed over the entire pad surface. Metal loading plates can be used for this purpose.

SPRING ISOLATOR (Type 3) 	**Note 22.** Steel springs are the most popular and versatile isolators for HVAC applications because they are available for almost any deflection and have a virtually unlimited life. Spring isolators may have a rubber acoustical barrier to reduce transmission of high-frequency vibration and noise that can migrate down the steel spring coil. They should be corrosion protected if installed outdoors or in a corrosive environment. The basic types include the following: **Note 23.** *Open spring isolators* (type 3) consist of top and bottom load plates with adjustment bolts for leveling equipment. Springs should be designed with a horizontal stiffness of at least 80% of the vertical stiffness (k_x/k_y) to ensure stability. Similarly, the springs should have a minimum ratio of 0.8 for the diameter divided by the deflected spring height.
RESTRAINED SPRING ISOLATOR (Type 4) 	**Note 24.** *Restrained spring isolators* (type 4) have hold-down bolts to limit vertical as well as horizontal movement. They are used with (a) equipment with large variations in mass (e.g., boilers, chillers, cooling towers) to restrict movement and prevent strain on piping when water is removed, (b) outdoor equipment, such as condensing units and cooling towers, to prevent excessive movement due to wind loads, and (c) with any equipment subject to seismic forces. Spring criteria should be the same as open spring isolators, and snubbers should have adequate clearance so that they are activated only when a temporary restraint is needed. See Chapter 55 of the 2015 *ASHRAE Handbook—HVAC Applications* for typical snubber types. *Closed mounts* or *housed spring isolators* consist of two telescoping housings separated by a resilient material. These provide lateral snubbing and some vertical damping of equipment movement, but do not limit the vertical movement. Additional vertical snubbers must be used where vertical travel must be limited (see Chapter 55 of the 2015 *ASHRAE Handbook—HVAC Applications*). Care should be taken in selection and installation to minimize binding and short circuiting.
AIR SPRINGS (Type 6)  ROLLING LOBE  BELLOWS	**Note 25.** Air springs (type 6) can be designed for any frequency, but are economical only in applications with natural frequencies of 1.33 Hz or less (6 in. or greater deflection). They do not transmit high-frequency noise and are often used to replace high-deflection springs on problem jobs (e.g., large transformers on upper-floor installations). A constant air supply (an air compressor with an air dryer) and leveling valves are typically required.



RUBBER HANGER (Type 2)

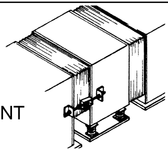


SPRING HANGER (Type 3)

Note 26. Isolation hangers (types 2 and 3) are used for suspended pipe and equipment and have rubber, springs, or a combination of spring and rubber elements. Criteria should be similar to open spring isolators, though lateral stability is less important. Where support rod angular misalignment is a concern, use hangers that have sufficient clearance and/or incorporate rubber bushings to prevent the rod from touching the housing. Swivel or traveler arrangements may be necessary for connections to piping systems subject to large thermal movements.

Precompressed spring hangers incorporate some means of precompression or preloading of the isolator spring to minimize movement of the isolated equipment or system. These are typically used on piping systems that can change weight substantially between installation and operation.

THRUST RESTRAINT (Type 5)

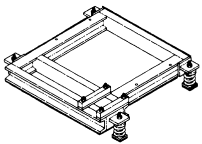


Note 27. Thrust restraints (type 5) are similar to spring hangers or isolators and are installed in pairs to resist the thrust caused by air pressure. These are typically sized to limit lateral movement to 0.25 in. or less.

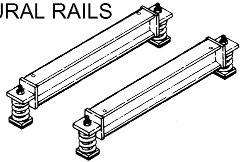
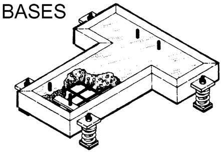
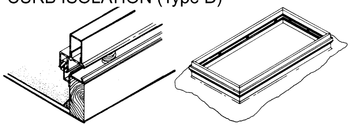
DIRECT ISOLATION (Type A)

Note 28. Direct isolation (type A) is used when equipment is unitary and rigid and does not require additional support. Direct isolation can be used with large chillers, some fans, packaged air-handling units, and air-cooled condensers. If there is any doubt that the equipment can be supported directly on isolators, use structural bases (type B) or inertia bases (type C), or consult the equipment manufacturer.

STRUCTURAL BASES (Type B)



Note 29. Structural bases (type B) are used where equipment cannot be supported at individual locations and/or where some means is necessary to maintain alignment of component parts in equipment. These bases can be used with spring or rubber isolators (types 2 and 3) and should have enough rigidity to resist all starting and operating forces without supplemental hold-down devices. Bases are made in rectangular configurations using structural members with a depth equal to one-tenth the longest span between isolators. Typical base depth is between 4 and 12 in., except where structural or alignment considerations dictate otherwise.

STRUCTURAL RAILS (Type B) 	**Note 30.** Structural rails (type B) are used to support equipment that does not require a unitary base or where the isolators are outside the equipment and the rails act as a cradle. Structural rails can be used with spring or rubber isolators and should be rigid enough to support the equipment without flexing. Usual practice is to use structural members with a depth one-tenth of the longest span between isolators, typically between 4 and 12 in., except where structural considerations dictate otherwise.
CONCRETE BASES (Type C) 	**Note 31.** Concrete bases (type C) are used where the supported equipment requires a rigid support (e.g., flexible-coupled pumps) or excess heaving motion may occur with spring isolators. They consist of a steel pouring form usually with welded-in reinforcing bars, provision for equipment hold-down, and isolator brackets. Like structural bases, concrete bases should be sized to support piping elbow supports, rectangular or T-shaped, and for rigidity, have a depth equal to one-tenth the longest span between isolators. Base depth is typically between 6 and 12 in. unless additional depth is specifically required for mass, rigidity, or component alignment.
CURB ISOLATION (Type D) 	**Note 32.** Curb isolation systems (type D) are specifically designed for curb-supported rooftop equipment and have spring isolation with a watertight, and sometimes airtight, assembly. *Rooftop rails* consist of upper and lower frames separated by nonadjustable springs and rest on top of architectural roof curbs. *Isolation curbs* incorporate the roof curb into their design as well. Both kinds are designed with springs that have static deflections in the 1 to 3 in. range to meet the design criteria described in type 3. Flexible elastomeric seals are typically most effective for weatherproofing between the upper and lower frames. A continuous sponge gasket around the perimeter of the top frame is typically applied to further weatherproof the installation.

19. HVAC SYSTEMS AND EQUIPMENT

For discussion of boilers, compressors, chillers, and cooling towers, see the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

Furnaces

Furnaces are self-enclosed, permanently installed major appliances that provide heated air through ductwork to the space being heated. In addition, a furnace may provide the indoor fan necessary for circulating heated or cooled air from a split or single-package air conditioner or heat pump. Furnaces may be used in either residential or commercial applications, and may be grouped according to the following characteristics:

- Heat source: electricity, natural gas/propane (fan assisted, condensing or noncondensing), or oil (forced draft with power atomizing burner)
- Installation location: within conditioned space (indoors), or outside conditioned space (either outdoors, or inside the structure but not within the conditioned space)
- Combustion air source: direct vent (outdoor air) or indoor air
- Mounting arrangement and airflow: horizontal, vertical upflow, vertical downflow, or multiposition

Furnaces that use electricity as a heat source include one or more resistance-type heating elements that heats the circulating air either directly or through a metal sheath that encloses the resistance element. In gas- or oil-fired furnaces, combustion occurs in the heat exchanger sections or in a combustion chamber, with direct-spark, hot-surface, or electric ignition. Circulating air passes over the outer surfaces of a heat exchanger so that it does not contact the fuel or the products of combustion, which are passed to the outdoor atmosphere through a vent.

In North America, natural gas is the most common fuel supplied for residential heating, and the central-system forced-air furnace (Figure 19.1) is the most common way of heating with natural gas.

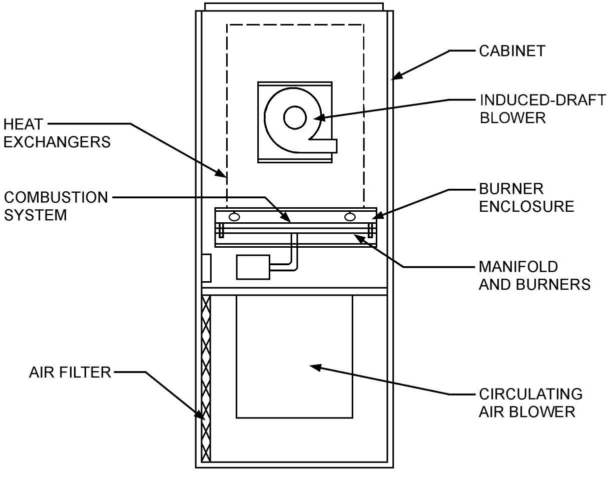


Figure 19.1 Induced-Draft Gas Furnace [2016S, Ch 33, Fig 1]

Furnaces with gas-fired burners have heat exchangers that are typically made either of left/right sets of formed parts that are joined together to form a clamshell, finless tubes bent into a compact form, or finned-tube (condensing) heat exchangers. Standard indoor furnace heat exchangers are generally made of alloy steel. Common corrosion-resistant materials include aluminized steel and stainless steel. Furnaces certified for use downstream of a cooling coil must have corrosion-resistant heat exchangers.

Heat exchangers of oil-fired furnaces are normally heavy-gage steel formed into a welded assembly. Hot flue products flow through the inside of the heat exchanger into the chimney, and conditioned air flows over the outside of the heat exchanger and into the air supply plenum.

Fan-assisted combustion furnaces use a small blower to induce flue products through the furnace. Induced-draft furnaces may or may not have a relief air opening, but they meet the same safety requirements regardless. Residential furnaces built since 1987 are equipped with a blocked-vent shutoff switch to shut down the furnace in case the vent becomes blocked.

Direct-vent furnaces use outdoor air for combustion. Outdoor air is supplied to the furnace combustion chamber by direct connections between the furnace and the outdoor air. If the vent or the combustion air supply becomes blocked, the furnace control system will shut down the furnace.

ANSI Standard Z21.47/CSA 2.3 classifies venting systems. Central furnaces are categorized by temperature and pressure attained in the vent and by the steady-state efficiency attained by the furnace. Although ANSI Standard Z21.47/CSA 2.3 uses 83% as the steady-state efficiency dividing central furnace categories, a general rule of thumb is as follows:

Category I: nonpositive vent pressure and flue loss of 17% or more
Category II: nonpositive vent pressure and flue loss less than 17%
Category III: positive vent pressure and flue loss of 17% or more
Category IV: positive vent pressure and flue loss less than 17%

Furnaces rated in accordance with ANSI Standard Z21.47/CSA 2.3 that are not direct vent are marked to show that they are in one of these four venting categories.

Ducted-system, oil-fired, forced-air furnaces are usually forced draft.

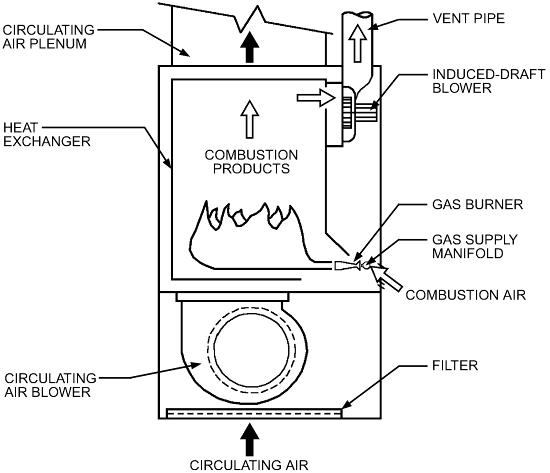


Figure 19.2 Upflow Category I Furnace with Induced-Draft Blower [2016S, Ch 33, Fig 2]

Evaporative Cooling

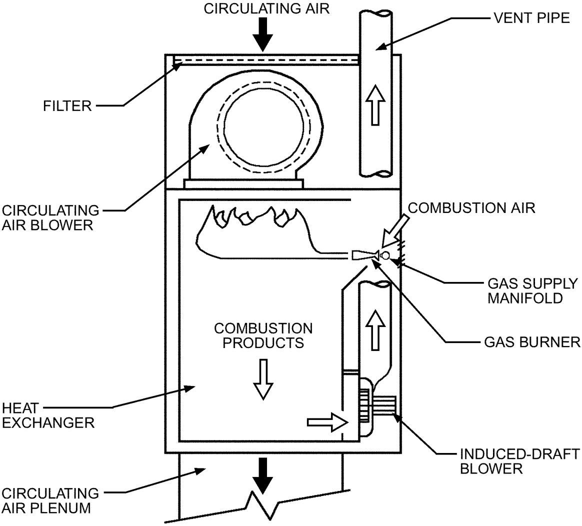


Figure 19.3 Downflow (Counterflow) Category I Furnace with Induced-Draft Blower
[2016S, Ch 33, Fig 3]

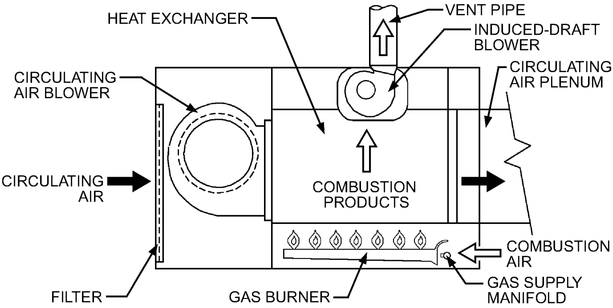


Figure 19.4 Horizontal Category I Furnace with Induced-Draft Blower
[2016S, Ch 33, Fig 4]

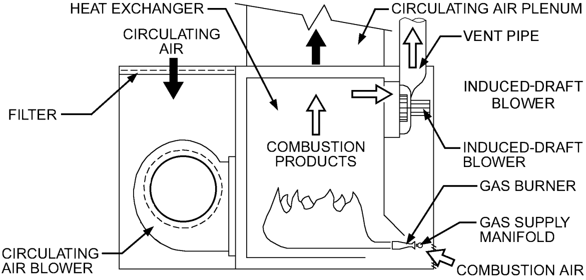


Figure 19.5 Basement (Lowboy) Category I Furnace with Induced-Draft Blower
[2016S, Ch 33, Fig 5]

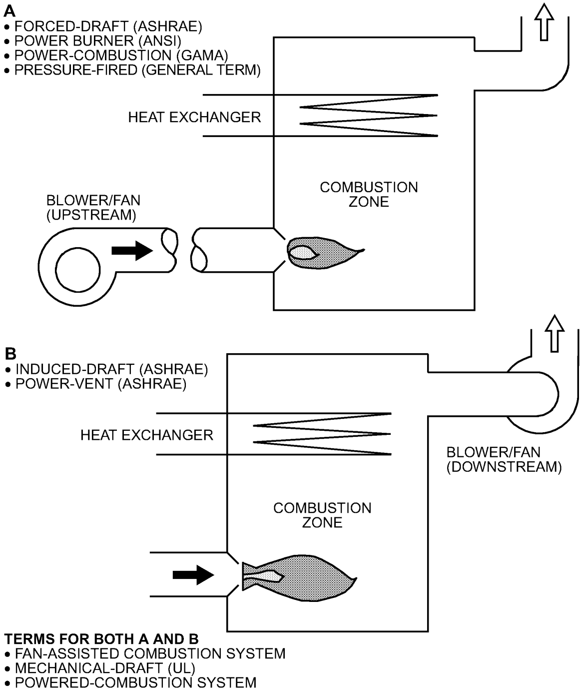


Figure 19.6 Terminology Used to Describe Fan-Assisted Combustion [2016S, Ch 33, Fig 6]

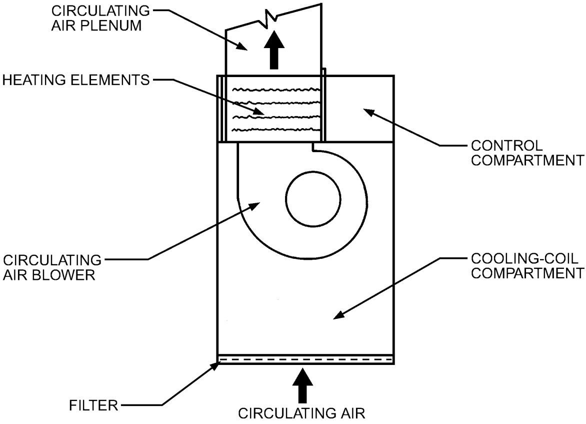


Figure 19.7 Electric Forced-Air Furnace [2016S, Ch 33, Fig 7]

Furnaces with capacities above 150,000 Btu/h are classifiied as commercial furnaces. The other basic differences between residential and commercial furnaces are available options such as economizers, outdoor air dampers, and the type of electrical service required (three-phase).

Commercial heating equipment comes in almost as many flow arrangements and design variations as residential equipment. Some are identical to residential equipment, whereas others are unique to commercial applications. Some commercial units function as a part of a ducted system, and others operate as unducted space heaters.

Externally, the furnace is controlled by a low-voltage room thermostat.

Several types of gas valves perform various functions within the furnace. The type of valve available relates closely to the type of ignition device used. **Two-stage valves**, available on some furnaces, operate at full gas input or at a reduced rate, and are controlled by either a two-stage thermostat or a software algorithm programmed in the furnace control system.

The **fan control switch** controls the circulating air blower. This switch may be temperature-sensitive and exposed to the circulating airstream in the furnace cabinet, or it may be an electronically operated relay. Blower start-up is typically delayed about 1 min after burner start-up. This delay gives the heat exchangers time to warm up and reduces the flow of cold air when the blower comes on. Blower shutdown is also delayed several minutes after burner shutdown to remove residual heat from the heat exchangers and to improve the annual efficiency of the furnace.

The **limit switch** prevents overheating in the event of severe reduction in circulating airflow. This temperature-sensitive switch is exposed to the circulating airstream and shuts off the heat source (e.g., gas valve or electric element) if the temperature of air leaving the furnace is excessive. The fan control and limit switches are sometimes incorporated in the same housing and may be operated by the same thermostatic element. In the United States, the **blocked-vent shutoff switch** and **flame rollout switch** shut off the gas valve if the vent is blocked or when insufficient combustion air is present.

Furnaces using fan-assisted combustion feature a **pressure switch** to verify the flow of combustion air before opening the gas valve.

Electronic control systems are available in furnaces to provide sequencing of the inducer prepurge, ignition, circulating air blower operation, and inducer postpurge functions according to an algorithm provided by the manufacturer.

Furnaces can be installed inside or outside a building. For ideal air distribution, locate the unit in the center of the structure being heated. Furnaces are typically located in a closet, mechanical room, basement, attic, crawlspace, garage, or outdoors.

The type of fuel selected for heating is based on relative fuel cost, number of heating degree-days, and availability of utilities in the area. The most common fuel is natural gas because of its clean burning characteristics, and because of the continuous supply of this fuel through underground distribution networks to most urban settings. Propane and oil fuels are also commonly used. These fuels require on-site storage and periodic fuel deliveries. Electric heat is also continuously available through electrical power grids and is common especially where natural gas is not provided, or where the heating demand is small relative to the cooling demand.

Furnaces are clearly marked for the type of fuel to be used. In some cases, a manufacturer-approved conversion kit may be necessary to convert a furnace from one fuel type to another. If the fuel type is changed after the original installation, the conversion must be done by a qualified service person per the manufacturer's instructions and using the manufacturer's specified conversion kit. After conversion, the unit must be properly inspected by the local code authority.

All fuel-burning furnaces must be properly vented to the outdoors. Metal vents, masonry chimneys, and plastic vents are commonly used for furnaces. Manufacturers provide installation instructions for venting their furnaces, and Chapter 35 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has a detailed discussion on venting.

Air for combustion enters the combustion zone through louvers or pipes. Outdoor air usually has lower levels of pollutants than are typically found in air from indoors, garages, utility rooms, and basements.

The furnace's heating capacity (i.e., the maximum heating rate the furnace can provide) is provided on the appliance rating plate; it is also available through the manufacturers' product literature.

Other factors should be considered when determining furnace capacity. Thermostat setback recovery may require additional heating capacity. Increasing furnace capacity may increase space temperature swing, and thus reduce comfort. Two-stage or step-modulating equipment could help by using the unit's maximum capacity to meet the setback recovery needs, and providing a lower stage of heating capacity at other times.

Fuel-burning furnaces are typically subdivided into two primary categories:

- **Condensing** furnaces typically have high efficiencies, ranging from 89 to 98%, because they have a specially designed secondary heat exchanger that extracts the heat of vaporization of water vapor in the exhaust. The dew-point temperatures of flue gases of condensing furnaces are significantly above the vent temperature, so plastic or other corrosion-resistant venting material is required. Condensing furnaces must be plumbed for condensate disposal.
- **Noncondensing** furnaces have generally less than 82% steady-state efficiency. This type of furnace has higher flue gas temperatures and requires either metal, masonry, or a combination of the two for venting materials.

Hydronic Heating Units and Radiators

Radiators, convectors, and baseboard and finned-tube units are heat-distributing devices used in hot-water and steam heating systems. They supply heat by a **combination of radiation and convection** and maintain the desired air temperature and/or mean radiant temperature in a space without fans. Figure 19.8 shows sections of typical heat-distributing units. In heating systems, radiant panels are also used. Units are inherently self-adjusting in the sense that heat output is based on temperature differentials; cold spaces receive more heat and warmer spaces receive less heat.

The following are the most common types of radiators:

- **Sectional radiators** are fabricated from welded sheet metal sections (generally two, three, or four tubes wide), and resemble freestanding cast-iron radiators.
- **Panel radiators** consist of fabricated flat panels (generally one, two, or three deep), with or without an exposed extended fin surface attached to the rear for increased output. These radiators are most common in Europe.
- **Tubular steel radiators** consist of supply and return headers with interconnecting parallel steel tubes in a wide variety of lengths and heights. They may be specially shaped to coincide with the building structure. Some are used to heat bathroom towel racks.
- **Specialty radiators** are fabricated of welded steel or extruded aluminum and are designed for installation in ceiling grids or floor-mounting. Various unconventional shapes are available.

Pipe coils have largely been replaced by finned tubes. See Table 5 in Chapter 28 of the 1988 *ASHRAE Handbook—Equipment* for the heat emission of such pipe coils.

A **convector** is a heat-distributing unit that operates with gravity-circulated air (natural convection). It has a heating element with a large amount of secondary surface and contains two or more tubes with headers at both ends. The heating element is surrounded by an enclosure with an air inlet below and an air outlet above the heating element.

Baseboard (or baseboard radiation) units are designed for installation along the bottom of walls in place of the conventional baseboard. They may be made of cast iron, with a substantial portion of the front face directly exposed to the room, or with a finned-tube element in a sheet metal enclosure. They use gravity-circulated room air.

Baseboard heat-distributing units are divided into three types: radiant, radiant convector, and finned tube. The **radiant** unit, which is made of aluminum, has no openings for air to pass over the wall side of the unit. Most of this unit's heat output is by radiation.

- The **radiant-convector** baseboard is made of cast iron or steel. The units have air openings at the top and bottom to allow circulation of room air over the wall side of the unit, which has extended surface to provide increased heat output. A large portion of the heat emitted is transferred by convection.
- The **finned-tube** baseboard has a finned-tube heating element concealed by a long, low sheet metal enclosure or cover. A major portion of the heat is transferred to the room by convection. The output varies over a wide range, depending on the physical dimensions and the materials used. A unit with a high relative output per unit length compared to overall heat loss (which would result in a concentration of the heating element over a relatively small area) should be avoided. Optimum comfort for room occupants is obtained when units are installed along as much of the exposed wall as possible.

Finned-tube (or fin-tube) units are fabricated from metallic tubing, with metallic fins bonded to the tube. They operate with gravity-circulated room air. Finned-tube elements are available in several tube sizes, in either steel or copper (1 to 2 in. nominal steel or 3/4 to 1 1/4 in. nominal copper) with various fin sizes, spacings, and materials. Resistance to steam or water flow is the same as that through standard distribution piping of equal size and type.

Finned-tube elements installed in occupied spaces generally have covers or enclosures in a variety of designs. When human contact is unlikely, they are sometimes installed bare or provided with an expanded metal grille for minimum protection.

The heat output ratings of heat-distributing units are expressed in Btu/h or in square feet equivalent direct radiation (EDR). By definition, 240 Btu/h = 1 ft^2 EDR with 1 psig steam.

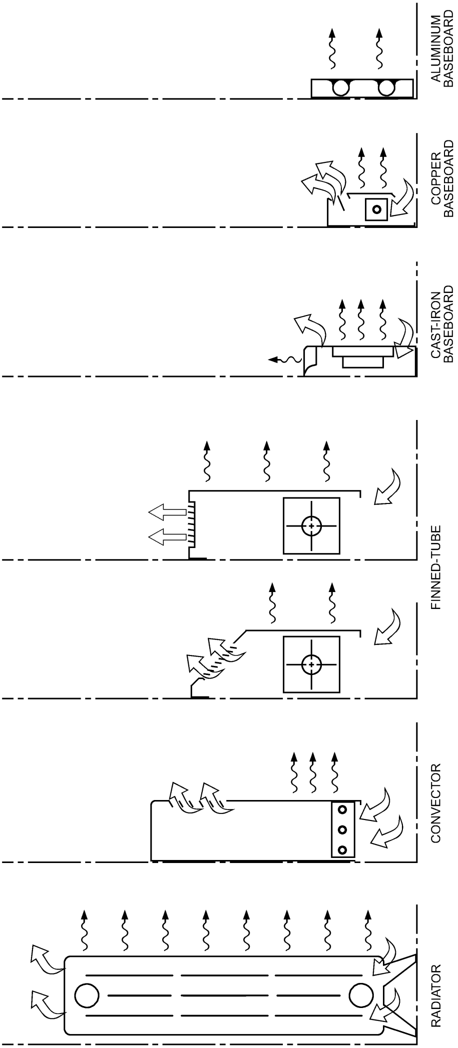


Figure 19.8 Typical Heating Units

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Table 19.1 Small-Tube Cast-Iron Radiators [2016S, Ch 36, Tbl 1]

Number of Tubes per Section	Catalog Rating per Section,[a] ft²	Catalog Rating per Section,[a] Btu/h	Section Dimensions: A Height, in.[b]	B Width, in. Min.	B Width, in. Max.	C Spacing, in.[c]	D Leg Height, in.[b]
3	1.6	384	25	3.25	3.50	1.75	2.50
4	1.6	384	19	4.44	4.81	1.75	2.50
	1.8	432	22	4.44	4.81	1.75	2.50
	2.0	480	25	4.44	4.81	1.75	2.50
5	2.1	504	22	5.63	6.31	1.75	2.50
	2.4	576	25	5.63	6.31	1.75	2.50
6	2.3	552	19	6.81	8	1.75	2.50
	3.0	720	25	6.81	8	1.75	2.50
	3.7	888	32	6.81	8	1.75	2.50

[a] Ratings based on steam at 215°F and air at 70°F. They apply only to installed radiators exposed in a normal manner, not to radiators installed behind enclosures, behind grilles, or under shelves. For Btu/h ratings at other temperatures, multiply table values by factors found in 2016S, Ch 36, Tbl 2.

[b] Overall height and leg height, as produced by some manufacturers, are 1 in. greater than shown in columns A and D. Radiators may be furnished without legs. Where greater than standard leg heights are required, leg height should be 4.5 in.

[c] Length equals number of sections multiplied by 1.75 in.

Table 19.2 Correction Factors *c* for Various Types of Heating Units [2016S, Ch 36, Tbl 2]

Steam Pressure (Approx.)		Steam or Water Temp., °F	Cast-Iron Radiator Room Temp., °F					Convector Inlet Air Temp., °F					Finned-Tube Inlet Air Temp., °F					Baseboard Inlet Air Temp., °F				
			80	75	70	65	60	75	70	65	60	55	75	70	65	60	55	75	70	65	60	55
		100											0.10	0.12	0.15	0.17	0.20	0.08	0.10	0.13	0.15	0.18
		110											0.15	0.17	0.20	0.23	0.26	0.13	0.15	0.18	0.21	0.25
		120											0.20	0.23	0.26	0.29	0.33	0.18	0.21	0.25	0.28	0.31
		130											0.26	0.29	0.33	0.36	0.40	0.25	0.28	0.31	0.34	0.38
		140											0.33	0.36	0.40	0.42	0.45	0.31	0.34	0.38	0.42	0.45
in. Hg Vac.	**psia**																					
22.4	3.7	150	0.39	0.42	0.46	0.50	0.54	0.35	0.39	0.43	0.46	0.50	0.40	0.42	0.45	0.49	0.53	0.38	0.42	0.45	0.49	0.53
20.3	4.7	160	0.46	0.50	0.54	0.58	0.62	0.43	0.47	0.51	0.54	0.58	0.45	0.49	0.53	0.57	0.61	0.45	0.49	0.53	0.57	0.61
17.7	6.0	170	0.54	0.58	0.62	0.66	0.69	0.51	0.54	0.58	0.63	0.67	0.53	0.57	0.61	0.65	0.69	0.53	0.57	0.61	0.65	0.69
14.6	7.5	180	0.62	0.66	0.69	0.74	0.78	0.58	0.63	0.67	0.71	0.76	0.61	0.65	0.69	0.73	0.78	0.61	0.65	0.69	0.72	0.78
10.9	9.3	190	0.69	0.74	0.78	0.83	0.87	0.67	0.71	0.76	0.81	0.85	0.69	0.73	0.78	0.81	0.86	0.69	0.73	0.78	0.82	0.86
6.5	11.5	200	0.78	0.83	0.87	0.91	0.95	0.76	0.81	0.85	0.90	0.95	0.77	0.81	0.86	0.90	0.95	0.81	0.86	0.92	0.95	1.00
psig	**psia**																					
1	15.6	215	0.91	0.95	1.00	1.04	1.09	0.90	0.95	1.00	1.05	1.10	0.91	0.94	1.00	1.06	1.11	0.91	0.95	1.00	1.05	1.09
6	21	230	1.04	1.09	1.14	1.18	1.23	1.05	1.10	1.15	1.20	1.26	1.03	1.08	1.14	1.19	1.24	1.04	1.09	1.14	1.19	1.25
15	30	250	1.23	1.28	1.32	1.37	1.43	1.27	1.32	1.37	1.43	1.47	1.20	1.26	1.31	1.37	1.43	1.22	1.27	1.32	1.37	1.43
27	42	270	1.43	1.47	1.52	1.56	1.61	1.47	1.54	1.59	1.67	1.72	1.38	1.44	1.50	1.56	1.62	1.43	1.47	1.52	1.59	1.64
52	67	300	1.72	1.75	1.82	1.89	1.92	1.85	1.89	1.96	2.04	2.08	1.67	1.73	1.79	1.86	1.92	1.75	1.82	1.89	1.92	1.96

Note: Use these correction factors to determine output ratings for radiators, convectors, and finned-tube and baseboard units at operating conditions other than standard. Standard conditions in the United States for a radiator are 215°F heating medium temperature and 70°F room temperature (at center of space and at 5 ft level).

Standard conditions for convectors and finned-tube and baseboard units are 215°F heating medium temperature and 65°F inlet air temperature at 29.92 in. Hg atmospheric pressure. Water flow is 3 fps for finned-tube units. Inlet air at 65°F for convectors and finned-tube or baseboard units represents the same room comfort conditions as 70°F room air temperature for a radiator. Standard conditions for radiant panels are 122° F heating medium temperature and 68° F for room air temperature; *c* depends on panel construction.

To determine output of a heating unit under nonstandard conditions, multiply standard heating capacity by appropriate factor for actual operating heating medium and room or inlet air temperatures.

Corrections for Nonstandard Conditions

The heating capacity of a radiator, convector, baseboard, finned-tube heat-distributing unit, or radiant panel is a power function of the temperature difference between the air in the room and the heating medium in the unit, shown as

$$q = c(t_s - t_a)^n \tag{19.1}$$

where

q = heating capacity, Btu/h

c = constant determined by test

t_s = average temperature of heating medium, °F. For hot water, the arithmetic average of the entering and leaving water temperatures is used.

t_a = room air temperature, °F. Air temperature 60 in. above the floor is generally used for radiators, whereas entering air temperature is used for convectors, baseboard units, and finned-tube units.

n = exponent that equals 1.3 for cast-iron radiators, 1.4 for baseboard radiation, 1.5 for convectors, 1.0 for ceiling heating and floor cooling panels, and 1.1 for floor heating and ceiling cooling panels. For finned-tube units, n varies with air and heating medium temperatures. Correction factors to convert heating capacities at standard rating conditions to heating capacities at other conditions are given in Table 19.2.

Equation 19.1 may also be used to calculate heating capacity at nonstandard conditions.

Designing for high temperature drops through the system (as much as 60 to 80°F in low-temperature water (LTW) systems and as much as 200°F in high-temperature systems) can result in low water velocities in the finned-tube or baseboard element. Applying very short runs designed for conventional temperature drops (i.e., 20°F) can also result in low velocities.

Figure 19.9 shows the effect of water velocity on the heat output of typical sizes of finned-tube elements. The figure is based on work done by Harris (1957) and Pierce (1963) and tests at the Hydronics Institute. The velocity correction factor F_v is

$$F_v = (V/3.0)^{0.04} \tag{19.2}$$

where V = water velocity, fps.

Heat output varies little over the range from 0.5 to 3 fps, where F_v ranges from 0.93 to 1.00. The factor drops rapidly below 0.5 fps because flow changes from turbulent to laminar at around 0.1 fps. Avoid such a low velocity because the output is difficult to predict accurately when designing a system. In addition, the curve is so steep in this region that small changes in actual flow have a significant effect on output. Not only does the heat transfer rate change, but the temperature drop and, therefore, the average water temperature change (assuming a constant inlet temperature).

The designer should check water velocity throughout the system and select finned-tube or baseboard elements on the basis of velocity as well as average temperature. Manufacturers of finned-tube and baseboard elements offer a variety of tube sizes, ranging from 0.5 in. copper tubes for small baseboard elements to 2 in. for large finned-tube units, to aid in maintenance of turbulent flow conditions over a wide range of flow.

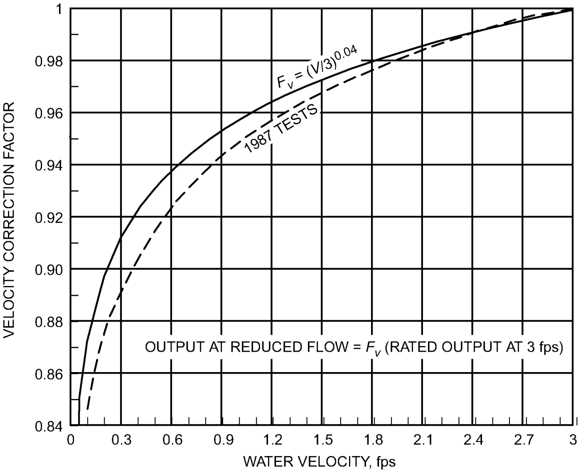


Figure 19.9 Water Velocity Correction Factor for Baseboard and Finned-Tube Radiators
[2016S, Ch 36, Fig 3]

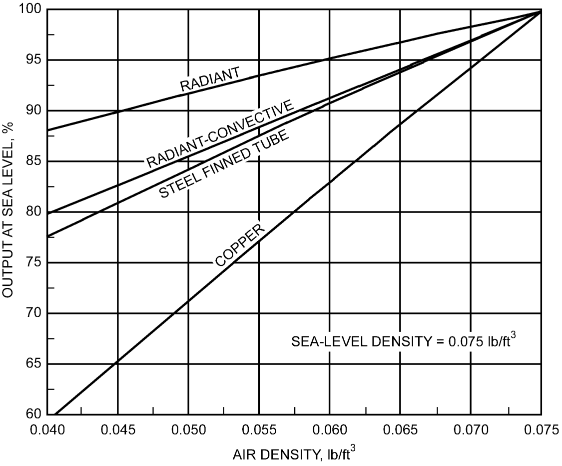


Figure 19.10 Effect of Air Density on Radiator Output [2016S, Ch 36, Fig 4]

Unit Ventilators, Unit Heaters, and Makeup Air Units

A heating unit ventilator is an assembly whose principal functions are to heat, ventilate, and cool a space by introducing outdoor air in quantities up to 100% of its rated capacity. The heating medium may be steam, hot water, gas, or electricity. The essential components of a heating unit ventilator are the fan, motor, heating element, damper, filter, automatic controls, and outlet grille, all of which are encased in a housing.

An **air-conditioning unit ventilator** is similar to a heating unit ventilator; however, in addition to the normal winter function of heating, ventilating, and cooling with outdoor air, it is also equipped to cool and dehumidify during the summer. It is usually arranged and controlled to introduce a fixed quantity of outdoor air for ventilation during cooling in mild weather. The air-conditioning unit ventilator may be provided with a various of combinations of heating and air-conditioning elements. Some of the more common arrangements include

- Combination hot- and chilled-water coil (two-pipe)
- Separate hot- and chilled-water coils (four-pipe)
- Hot-water or steam coil and direct-expansion coil
- Electric heating coil and chilled-water or direct-expansion coil
- Gas-fired furnace with direct-expansion coil

The typical unit ventilator has controls that allow heating, ventilating, and cooling to be varied while the fans operate continuously. In normal operation, the discharge air temperature from a unit is varied in accordance with the room requirements. The heating unit ventilator can provide **ventilation cooling** by bringing in outdoor air whenever the room temperature is above the room set point. Air-conditioning unit ventilators can provide refrigerated cooling when the outdoor air temperature is too high to be used effectively for ventilation cooling.

Unit ventilators are available for floor mounting, ceiling mounting, and recessed applications. They are available with various airflow and capacity ratings, and the fan can be arranged so that air is either blown through or drawn through the unit. With direct-expansion refrigerant cooling, the condensing unit can either be furnished as an integral part of the unit ventilator assembly or be remotely located.

Unit ventilators are used primarily in schools, meeting rooms, offices, and other areas where the density of occupancy requires controlled ventilation to meet local codes.

Floor-model unit ventilators are normally installed on an outer wall near the centerline of the room. Ceiling models are mounted against either the outer wall or one of the inside walls. Ceiling models discharge air horizontally. Best results are obtained if the unit can be placed so that the airflow is not interrupted by ceiling beams or surface-mounted lighting fixtures.

Example. A room has a heat loss of 24,000 Btu/h at a winter outdoor design condition of 0°F and an indoor design of 70°F, with 20% outdoor air. Minimum air discharge temperature from the unit is 60°F. To obtain the specified number of air changes, a 1250 cfm unit ventilator is required. Determine the ventilation heat requirement, the total heating requirement, and the ventilation cooling capacity of this unit with outdoor air temperature below 60°F.

Solution:

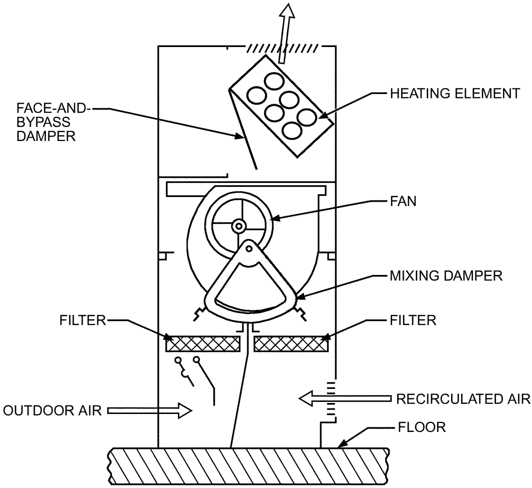
Ventilation heat requirement:

$$q_v = 60\rho c_p Q(t_i - t_o) \tag{19.3}$$

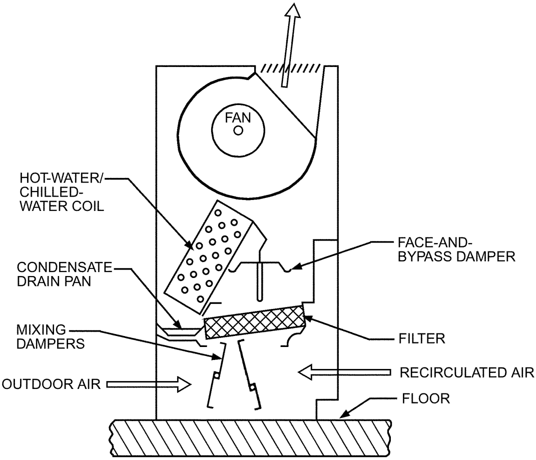
where

q_v = heat required to heat ventilating air, Btu/h
ρ = density of air at standard conditions = 0.075 lb/ft^3
c_p = air specific heat = 0.24 Btu/lb·°F
Q = ventilating airflow, cfm
t_i = required room air temperature, °F
t_o = outdoor air temperature, °F

$$q_v = 60 \times 0.075 \times 0.24 \times 1250(20/100)(70 - 0) = 18{,}900 \text{ Btu/h} \tag{19.4}$$

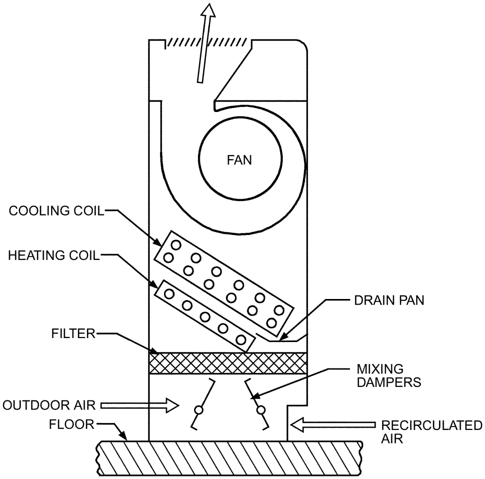


A. HEATING UNIT VENTILATOR

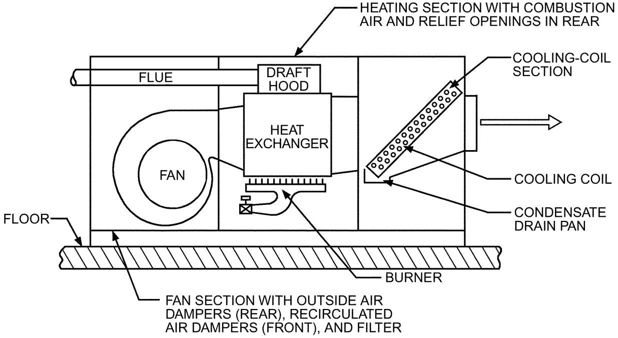


B. AIR-CONDITIONING UNIT VENTILATOR WITH COMBINED HOT- AND CHILLED-WATER COIL

Figure 19.11 Typical Unit Ventilators [2016S, Ch 28, Fig 1]



C. AIR-CONDITIONING UNIT VENTILATOR WITH SEPARATE COILS



D. GAS-FIRED AIR-CONDITIONING UNIT VENTILATOR

Figure 19.11 Typical Unit Ventilators [2016S, Ch 28, Fig 1]
(Continued)

Table 19.3 Typical Unit Ventilator Capacities [2016S, Ch 28, Tbl 1]

Airflow, cfm	Heating Unit Ventilator Total Heating Capacity, Btu/h	A/C Unit Ventilator Total Cooling Capacity, Btu/h
500	38,000	19,000
750	50,000	28,000
1000	72,000	38,000
1250	85,000	47,000
1500	100,000	56,000

Total heating requirement:

$$q_t = q_v + q_s \tag{19.5}$$

where

q_t = total heat requirement, Btu/h
q_s = heat required to make up heat losses, Btu/h

$$q_t = 18{,}900 + 24{,}000 = 42{,}900 \text{ Btu/h} \tag{19.6}$$

Ventilation cooling capacity:

$$q_c = 60\rho c_p Q(t_i - t_f) \tag{19.7}$$

where

q_c = ventilation cooling capacity of unit, Btu/h
t_f = unit discharge air temperature, °F

$$q_c = 60 \times 0.075 \times 0.24 \times 1250\,(70 - 60) = 13{,}500 \text{ Btu/h} \tag{19.8}$$

Unit Heaters

A unit heater is an assembly of elements with the main function of heating a space. The essential elements are a fan and motor, a heating element, and an enclosure. Filters, dampers, directional outlets, duct collars, combustion chambers, and flues may also be included. Some types of unit heaters are shown in Figure 19.12.

Unit heaters have the following principal characteristics:

- Relatively large heating capacities in compact casings
- Ability to project heated air in a controlled manner over a considerable distance
- Relatively low installed cost per unit of heat output
- Application where an elevated sound level is permissible

They are, therefore, usually placed in applications where the heating capacity requirements, physical volume of the heated space, or both, are too large to be handled adequately or economically by other means. By eliminating extensive duct installations, the space is freed for other use.

Unit heaters are mostly used for heating commercial and industrial structures such as garages, factories, warehouses, showrooms, stores, and laboratories, as well as corridors, lobbies, vestibules, and similar auxiliary spaces in all types of buildings. Unit heaters may often be used to advantage in specialized applications requiring spot or intermittent heating, such as at outer doors in industrial plants or in corridors and vestibules. Cabinet unit heaters may be used where heated air must be filtered.

The following factors should be considered when selecting a unit heater:

Heating Medium. The proper heating medium is usually determined by economics and requires examining initial cost, operating cost, and conditions of use.

Steam or **hot-water unit heaters** are relatively inexpensive but require a boiler and piping system. The unit cost of such a system generally decreases as the number of units increases. Therefore, steam or hot-water heating is most frequently used (1) in new installations involving a relatively large number of units, and (2) in existing systems that have sufficient capacity to handle the additional load. High-pressure steam or high-temperature hot-water units are normally used only in very large installations or when a high-temperature medium is required for process work. Low-pressure steam and conventional hot-water units are usually selected for smaller installations and for those concerned primarily with comfort heating.

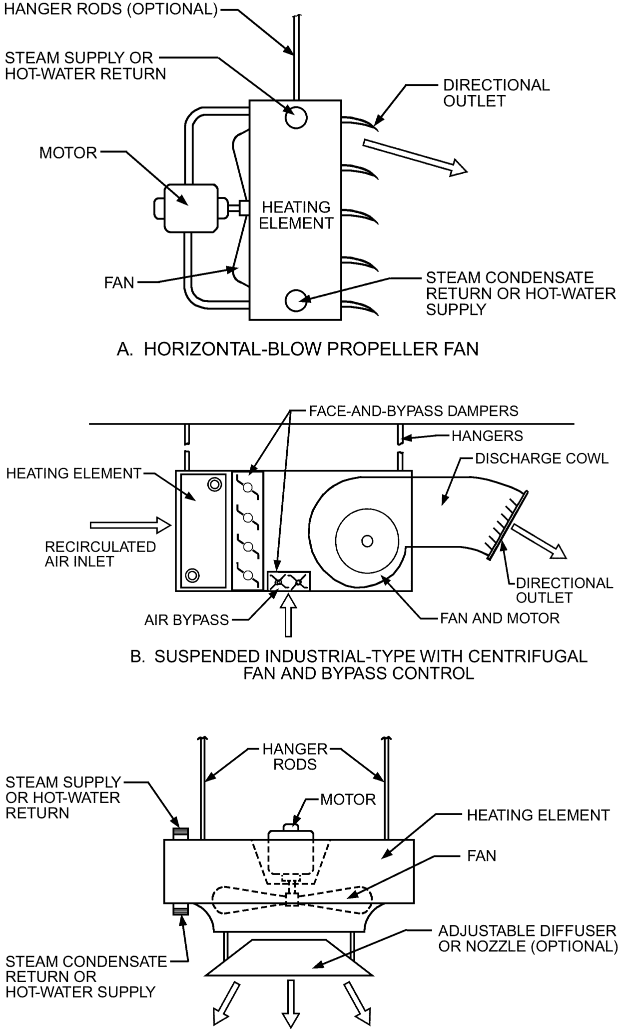
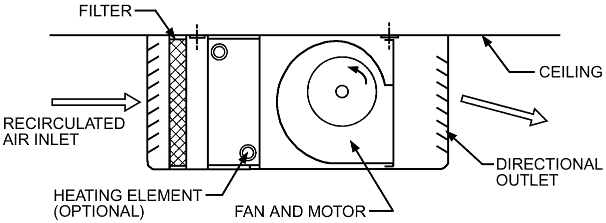
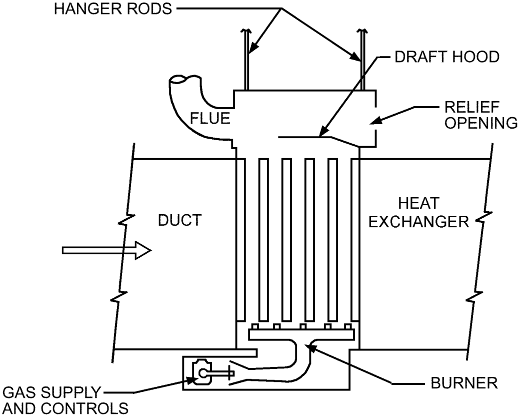


Figure 19.12 Typical Unit Heaters [2016S, Ch 28, Fig 2]

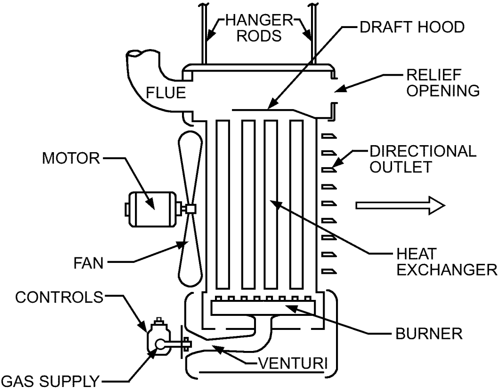


D. SUSPENDED CABINET WITH CENTRIFUGAL FAN

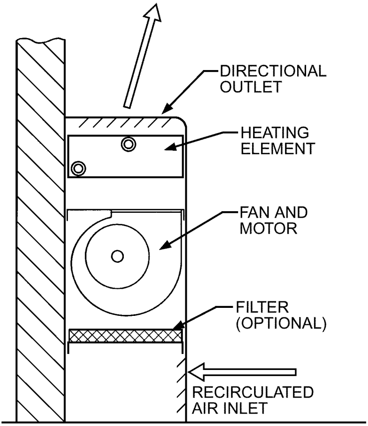


E. GAS INDIRECT-FIRED DUCT MOUNTED

Figure 19.12 Typical Unit Heaters [2016S, Ch 28, Fig 2]
(Continued)

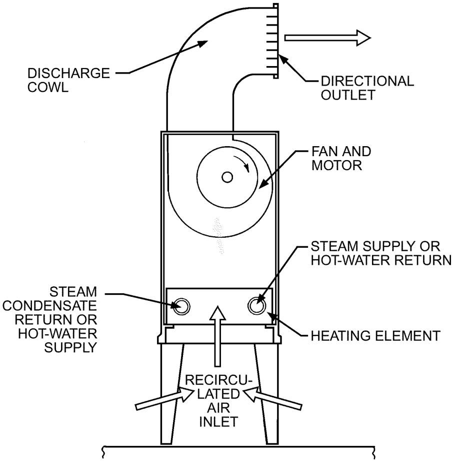


F. GAS INDIRECT-FIRED WITH PROPELLER FAN

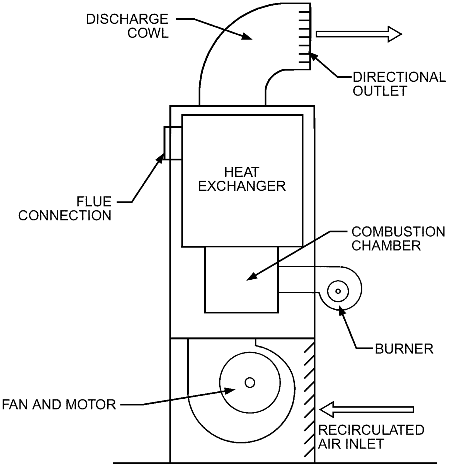


G. FLOOR-MOUNTED CABINET WITH CENTRIFUGAL FAN

Figure 19.12 Typical Unit Heaters [2016S, Ch 28, Fig 2]
(Continued)

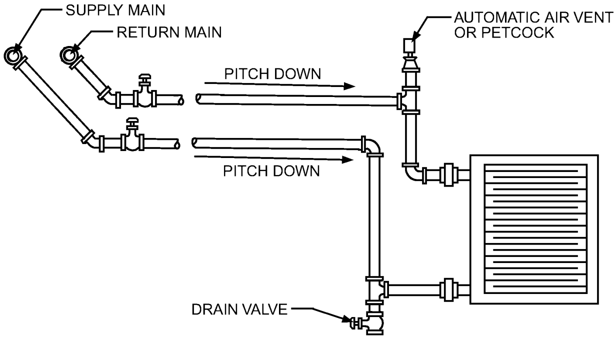


H. FLOOR-MOUNTED INDUSTRIAL-TYPE WITH CENTRIFUGAL FAN

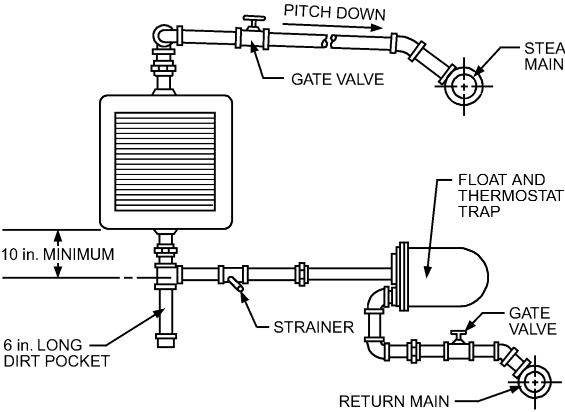


I. FLOOR-MOUNTED, INDUSTRIAL-TYPE, OIL OR GAS INDIRECT-FIRED WITH CENTRIFUGAL FAN

Figure 19.12 Typical Unit Heaters [2016S, Ch 28, Fig 2] ***(Continued)***

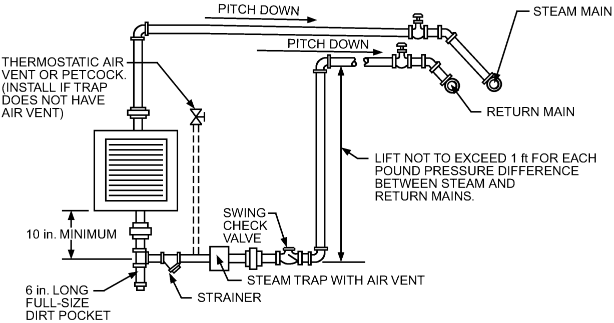


A. OVERHEAD HOT-WATER MAINS



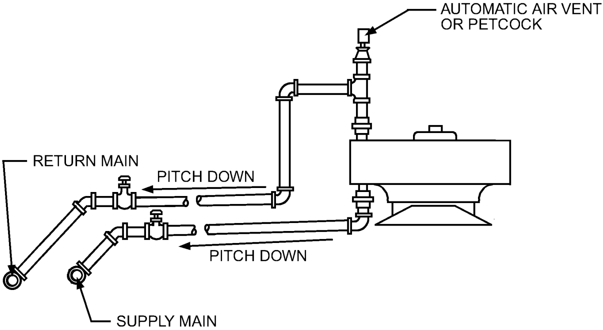
B. LOW-PRESSURE STEAM, OPEN GRAVITY, OR VACUUM RETURN

Figure 19.13 Hot Water and Steam Connections for Unit Heaters [2016S, Ch 28, Fig 4]



NOTE: This piping arrangement is only for two-position control. Modulating steam control may not provide sufficient pressure to lift condensate to return main in throttled position.

C. OVERHEAD STEAM AND RETURN MAINS



D. LOWER HOT-WATER MAINS

Figure 19.13 Hot Water and Steam Connections for Unit Heaters [2016S, Ch 28, Fig 4]
(Continued)

Gas and **oil indirect-fired unit heaters** are frequently preferred in small installations where the number of units does not justify the expense and space requirements of a new boiler system or where individual metering of the fuel supply is required, as in a shopping center. Gas indirect-fired units usually have either horizontal propeller fans or industrial centrifugal fans. Oil indirect-fired units largely have industrial centrifugal fans. Some codes limit the use of indirect-fired unit heaters in some applications. Indirect-fired oil and gas units are of blow-through design to mitigate the possibility of combustion products entering the occupied space.

Electric unit heaters are used when the cost of available electric power is lower than that of alternative fuel sources and for isolated locations, intermittent use, supplementary heating, or temporary service. Typical applications are ticket booths, security offices, factory offices, locker rooms, and other isolated rooms scattered over large areas. Electric units are particularly useful in isolated and untended pumping stations or pits, where they may be thermostatically controlled to prevent freezing.

Type of Unit. Propeller fan units are generally used in non-ducted applications where the heating capacity and distribution requirements can best be met by units of moderate output and where heated air does not need to be filtered. Horizontal-blow units are usually installed in buildings with low to moderate ceiling heights. Downblow units are used in spaces with high ceilings and where floor and wall space limitations dictate that heating equipment be kept out of the way. Downblow units may have an adjustable diffuser to vary the discharge pattern from a high-velocity vertical jet (to achieve the maximum distance of downward throw) to a horizontal discharge of lower velocity (to prevent excessive air motion in the zone of occupancy). Revolving diffusers are also available.

Cabinet unit heaters are used when a more attractive appearance is desired. They are suitable for free-air delivery or low static pressure duct applications. They may be equipped with filters, and they can be arranged to discharge either horizontally or vertically up or down.

Industrial centrifugal fan units are applied where heating capacities and space volumes are large or where filtration of the heated air or operation against static resistance is required. Downblow or horizontal-blow units may be used, depending on the requirements.

Duct unit heaters are used where the air handler is remote from the heater. These heaters sometimes provide an economical means of adding heating to existing cooling or ventilating systems with ductwork. They require flow and temperature limit controls.

Location for Proper Heat Distribution. Units must be selected, located, and arranged to provide complete heat coverage while maintaining acceptable air motion and temperature at an acceptable sound level in the working or occupied zone. Proper application depends on size, number, and type of units; direction of airflow and type of directional outlet used; mounting height; outlet velocity and temperature; and air volumetric flow. Many of these factors are interrelated.

The mounting height may be governed by space limitations or by the presence of equipment such as display cases or machinery. The higher a downblow heater is mounted, the lower the temperature of air leaving the heater must be to force the heated air into the occupied zone. Also, the distance that air leaving the heater travels depends largely on the air temperature and initial velocity. A high discharge temperature reduces the area of effective heat coverage because of its buoyancy.

For area heating, place horizontal-blow unit heaters in exterior zones such that they blow either along the exposure or toward it at a slight angle. When possible, arrange multiple units so that the discharge airstreams support each other and create a general circulatory motion in the space. Interior zones under exposed roofs or skylights should be completely blanketed. Arrange downblow units so that the heated areas from adjacent units overlap slightly to provide complete coverage.

For spot heating of individual spaces in larger unheated areas, single unit heaters may be used, but allowance must be made for the inflow of unheated air from adjacent spaces and the consequent reduction in heat coverage. Such spaces should be isolated by partitions or enclosures, if possible.

Horizontal unit heaters should have discharge outlets located well above head level. Both horizontal and vertical units should be placed so that the heated airstream is delivered to the occupied zone at acceptable temperature and velocity. Outlet air temperature of free-air delivery unit heaters used for comfort heating should be 50 to 60°F higher than the design room temperature. When possible, locate units so that they discharge into open spaces, such as aisles, and not directly on the occupants. For further information on air distribution, see Chapter 20 of the 2013 *ASHRAE Handbook—Fundamentals*.

Manufacturers' catalogs usually include suggestions for the best arrangements of various unit heaters, recommended mounting heights, heat coverage for various outlet velocities, final temperatures, directional outlets, and sound level ratings.

Steam or Hot Water. Heating capacity must be determined at a standard condition. Variations in entering steam or water temperature, entering air temperature, and steam or water flow affect capacity. Typical standard conditions for rating steam unit heaters are dry saturated steam at 2 psig pressure at the heater coil, air at 60°F (29.92 in. Hg barometric pressure) entering the heater, and the heater operating free of external resistance to airflow. Standard conditions for rating hot-water unit heaters are entering water at 200°F, water temperature drop of 20°F, entering air at 60°F and 29.92 in. Hg barometric pressure, and the heater operating free of external resistance to airflow.

Gas-Fired. Gas-fired unit heaters are rated in terms of both input and output, in accordance with the approval requirements of the American Gas Association.

Oil-Fired. Ratings of oil-fired unit heaters are based on heat delivered at the heater outlet.

Electric. Electric unit heaters are rated based on the energy input to the heating element.

Effect of Airflow Resistance on Capacity. Unit heaters are customarily rated at free-air delivery. Airflow and heating capacity decrease if outdoor air intakes, air filters, or ducts on the inlet or discharge are used. The manufacturer should have information on the heat output to be expected at other than free-air delivery.

Effect of Inlet Temperature. Changes in entering air temperature influence the total heating capacity in most unit heaters and the final temperature in all units. Because many unit heaters are located some distance from the occupied zone, possible differences between the temperature of the air actually entering the unit and that of air being maintained in the heated area should be considered, particularly with downblow unit heaters.

Filters. Air from propeller unit heaters cannot be filtered because the heaters are designed to operate with heater friction loss only. If dust in the building must be filtered, centrifugal fan units or cabinet units should be used.

The controls for a steam or hot water unit heater can provide either (1) on/off operation of the unit fan, or (2) continuous fan operation with modulation of heat output. For on/off operation, a room thermostat is used to start and stop the fan motor or group of fan motors. A limit thermostat, often strapped to the supply or return pipe, prevents fan operation in the event that heat is not being supplied to the unit. An auxiliary switch that energizes the fan only when power is applied to open the motorized supply valve may also be used to prevent undesirable cool air from being discharged by the unit.

Continuous fan operation eliminates both the intermittent blasts of hot air resulting from on/off operation and the stratification of temperature from floor to ceiling that often occurs during off periods. In this arrangement, a proportional room thermostat controls a valve modulating the heat supply to the coil or a bypass around the heating element. A limit thermostat or auxiliary switch stops the fan when heat is no longer available.

One type of control used with downblow unit heaters is designed to automatically return the warm air, which would normally stratify at the higher level, down to the zone of occupancy. Two thermostats and an auxiliary switch are required. The lower thermostat is placed in the zone of occupancy and is used to control a two-position supply valve to the heater. An auxiliary switch is used to stop the fan when the supply valve is closed. The higher thermostat is placed near the unit heater at the ceiling or roof level where the warm air tends to stratify. The lower thermostat automatically closes the steam valve when its setting is satisfied, but the higher thermostat overrides the auxiliary switch so that the fan continues to run until the temperature at the higher level falls below a point sufficiently high to produce a heating effect.

Indirect-fired and electric units are usually controlled by intermittent operation of the heat source under control of the room thermostat, with a separate fan switch to run the fan when heat is being supplied.

Unit heaters can be used to circulate air in summer. In such cases, the heat is shut off and the thermostat has a bypass switch, which allows the fan to run independently of the controls.

Makeup Air Units

Makeup air units are designed to condition ventilation air introduced into a space or to replace air exhausted from a building. The air exhausted may be from a process or general area exhaust, through either powered exhaust fans or gravity ventilators. The units may be used to prevent negative pressure within buildings or to reduce airborne contaminants in a space. The units may heat, cool, humidify, dehumidify, and/or filter incoming air. They may be used to replace air in the conditioned space or to supplement or accomplish all or part of the airflow needed to satisfy the heating, ventilating, or cooling airflow requirements.

Makeup air systems used for ventilation may be (1) sized to balance air exhaust volumes or (2) sized in excess of the exhaust volume to dilute contaminants. In applications where contaminant levels vary, variable-flow units should be considered so that the supply air varies for contaminant control and the exhaust volume varies to track supply volume. In critical spaces, the exhaust volume may be based on requirements to control pressure in the space.

Location. Makeup air units are defined by their location or the use of a key component. Examples are rooftop makeup air units, truss- or floor-mounted units, and sidewall units. Some manufacturers differentiate their units by heating mode, such as steam or direct gas-fired makeup air units.

Rooftop units are commonly used for large single-story industrial buildings to simplify air distribution. Access (via roof walks) is more convenient than access to equipment mounted in the truss; truss units are only accessible by installing a catwalk adjacent to the air units. Disadvantages of rooftop units are (1) they increase foot traffic on the roof, thus reducing its life and increasing the likelihood of leaks; (2) inclement weather reduces equipment accessibility; and (3) units are exposed to weather.

Makeup air units can also be placed around the perimeter of a building with air ducted through the sidewall. This approach limits future building expansion, and the effectiveness of ventilating internal spaces decreases as the building gets larger. However, access to the units is good, and minimum support is required because the units are mounted on the ground.

Heaters in makeup air systems may be direct gas-fired burners, electric resistance heating coils, indirect gas-fired heaters, steam coils, or hot-water heating coils. Air distribution systems are often required to direct heat to spaces requiring it.

Mechanical refrigeration with direct-expansion or chilled-water cooling coils, direct or indirect evaporative cooling sections, or well water coils may be used. Air distribution systems are often required to direct cooling to specific spaces that experience or create heat gain.

If direct-expansion coils are used in conjunction with direct-fired gas coils, the cooling coils' headers must be isolated from the airstream and directly vented outdoors.

High-efficiency filters (approximately MERV 16 for near-HEPA performance) are not normally used in a makeup air unit because of their relatively high cost. HVAC prefilters are generally in the MERV 6 to 13 range, depending on particulate removal needs.

Follow AMCA Standard 205-12 for fan selection. Fans should have variable-speed drives for possible energy savings or for use in variable-airflow systems.

Fans should have variable-speed drives for possible energy savings or for use in variable-airflow systems.

Controls for a makeup air unit fall into the following categories: (1) local temperature controls, (2) airflow controls, (3) plant-wide controls for proper equipment operation and efficient performance, (4) safety controls for burner gas, and (5) building smoke control systems.

Safety controls for gas-fired units include components to properly light the burner and to provide a safeguard against flame failure. The heater and all attached inlet ducting must be purged with at least four air changes before initiating an ignition sequence and before reignition after a malfunction. A flame monitor and control system must be used to automatically shut off gas to the burner upon burner ignition or flame failure. Critical malfunctions include flame failure, supply fan failure, combustion air depletion, power failure, control signal failure, excessive or inadequate inlet gas supply pressure, excess air temperature, and gas leaks in motorized valves or inlet gas supply piping.

Makeup air units should be interlocked with exhaust units to avoid overpressurization, and should include shutoff dampers with limit switches for when not in use. Damper leakage rates should be within limits set in ASHRAE Standard 90.1. These units should also be interlocked to the building's fire alarm system to shut down in the case of a fire, where required by applicable codes.

Consider using automatic safety shutoff valves on interconnecting piping systems where there are risks of overtemperature, overpressure, or gas leaks.

Small Forced-Air Heating and Cooling Systems

Forced-air systems are heating and/or cooling systems that use motor-driven blowers to distribute heated, cooled, and otherwise treated air to multiple outlets for the comfort of individuals in confined spaces. A typical residential or small commercial system includes (1) a heating and/or cooling unit, (2) supply and return ductwork (including registers and grilles), (3) accessory equipment, and (4) controls (see Figure 19.14).

Three types of forced-air heating and cooling devices are (1) furnaces, (2) air conditioners, and (3) heat pumps.

Furnaces are the basic component of most forced-air heating systems (see the section in this chapter). They are manufactured to use specific fuels such as oil, natural gas, or liquefied petroleum gas, and are augmented with an air-conditioning coil when cooling is included. The fuel used dictates installation requirements and safety considerations.

Common **air-conditioning** systems use a split configuration with an air-handling unit, such as a furnace. The air-conditioning evaporator coil (indoor unit) is installed on the discharge air side of the air handler. The compressor and condensing coil (outdoor unit) are located outside the structure, and refrigerant lines connect the outdoor and indoor units.

Self-contained air conditioners contain all necessary air-conditioning components, including circulating air blowers, and may or may not include fuel-fired heat exchangers or electric heating elements.

Heat pumps cool and heat using the refrigeration cycle. They are available in split and packaged (self-contained) configurations. Generally, air-source heat pumps require supplemental heating; therefore, electric heating elements are usually included with the heat pump as part of the forced-air system. Heat pumps offer high efficiency at mild temperatures, but may be combined with fossil-fuel furnaces to minimize heating cost. Heat pump supplemental heating also may be provided by thermostat-controlled gas heating appliances (e.g., fireplaces, free-standing stoves).

Forced-air systems may be equipped with accessories that further condition the air. They may modify humidity, remove contaminants, mix outdoor air with the recirculating air, or transfer energy in other ways. Disposable air filters on the return side of forced-air systems are so common that they are not considered accessories.

A simple thermostat controlling on/off cycling of central equipment may be all that is used or needed for temperature control. Such thermostats typically have a switch for automatic or continuous fan operation, and another to choose heating, cooling, or neither.

More complex systems may provide control features for timed variations (from simple night setback to a weekly schedule of temperatures); multiple independent zones, power stages, or fan speeds; influence of outdoor sensors; humidity; automatic switching between heating and cooling modes, etc.

The overall system design should proceed as follows:

1. Estimate heating and cooling loads, including target values for duct losses.
2. Determine preliminary ductwork location and materials of ductwork and outlets.
3. Determine heating and cooling unit location.
4. Select accessory equipment. Accessory equipment is not generally provided with initial construction; however, the system may be designed for later addition of these components.
5. Select control components.
6. Select heating/cooling equipment.
7. Determine maximum airflow (cooling or heating) for each supply and return location.
8. Determine airflow at reduced heating and cooling loads (two-speed and variable-speed fans).
9. Finalize heating/cooling equipment.
10. Finalize control system.
11. Finalize duct design and size.
12. Select supply and return grilles.
13. When the duct system is in place, measure duct leakage and compare results with target values used in step 1.

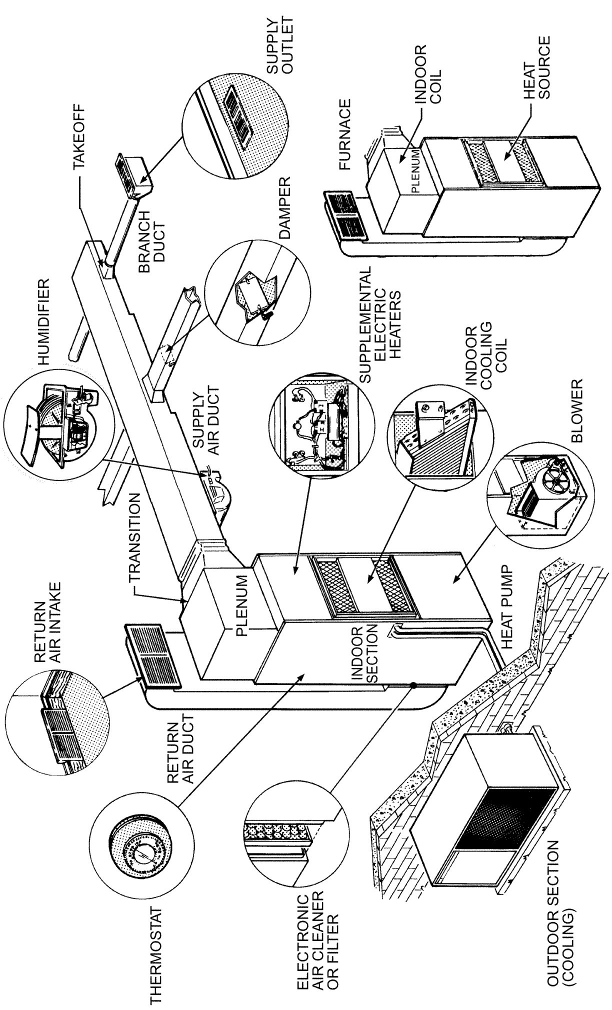


Figure 19.14 Heating and Cooling Components [2016S, Ch 10, Fig 1]

This procedure requires certain preliminary information such as location, weather conditions, and architectural considerations. See applicable sections of this guide for details on each step.

For maximum energy efficiency, ductwork and equipment should be installed in the conditioned space. The next best location is in a full basement. If a structure has an insulated, unvented, and sealed crawlspace, the ductwork and equipment can be located there (with appropriate provision for combustion air, if applicable), or the equipment can be placed in a closet or utility room. Vented attics and vented crawlspaces are the least preferred locations for ductwork and HVAC equipment.

Furnace heating output should match or slightly exceed the estimated design load. The Air Conditioning Contractors of America (ACCA *Manual* S) recommends a 40% limit on oversizing for fossil fuel furnaces. This limit minimizes venting problems associated with oversized equipment and improves part-load performance. Note that the calculated load must include duct loss, humidification load, and night setback recovery load, as well as building conduction and infiltration heat losses.

To help conserve energy, manufacturers have added features to improve furnace efficiency. Electric ignition has replaced the standing pilot; vent dampers and more efficient motors are also available. Furnaces with fan-assisted combustion systems (FACSs) and condensing furnaces also improve efficiency. Two-stage heating and cooling, variable-speed heat pumps, and two-speed and variable-speed blowers are also available.

Research on the effect of blower performance on residential forced-air heating system performance suggested reductions of 180 to 250 kWh/ yr for automatic furnace fan operation and 2600 kWh/ yr for continuous fan operation by changing from permanent split capacitor (PSC) blower motors to brushless permanent electronically commutated magnet motors (ECMs) (Phillips 1998).

A system designed to both heat and cool and that cycles cooling equipment on and off by sensing dry-bulb temperature alone should be sized to match the design heat gain as closely as possible. Oversizing under this control strategy could lead to higher-than-desired indoor humidity levels. Chapter 17 of the 2013 *ASHRAE Handbook—Fundamentals* recommends that cooling units not be oversized. Other sources suggest limiting oversizing to 15% of the sensible load. A heat pump should be sized for the cooling load with supplemental heat provided to meet heating requirements. Size air-source heat pumps in accordance with the equipment manufacturer recommendations. ACCA *Manual* S can also be used to assist in the selection and sizing of equipment.

The required airflow and the blower's static pressure limitation are the parameters around which the duct system is designed. The heat loss or gain for each space determines the proportion of the total airflow supplied to each space. Static pressure drop in supply registers should be limited to about 0.03 in. of water. The required pressure drop must be deducted from the static pressure available for duct design.

The flow delivered by a single supply outlet should be determined by considering the (1) space limitations on the number of registers that can be installed, (2) pressure drop for the register at the flow rate selected, (3) adequacy of air delivery patterns for offsetting heat loss or gain, and (4) space use pattern.

Manufacturers' specifications include blower airflow for each blower speed and external static pressure combination. Determining static pressure available for duct design should include the possibility of adding accessories in the future (e.g., electronic air cleaners or humidifiers). Therefore, the highest available fan speed should not be used for design.

For systems that heat only, the blower rate may be determined from the manufacturer's data. The temperature rise of air passing through the heat exchanger of a fossil-fuel furnace must be within the manufacturer's recommended range (usually 40 to 80°F). The possible later addition of cooling should also be considered by selecting a blower that operates in the midrange of the fan speed and settings.

For cooling only, or for heating and cooling, the design flow can be estimated by Equation 19.9:

$$Q = \frac{q_s U}{\rho c_p \Delta t} \tag{19.9}$$

where

Q = flow rate, cfm
q_s = sensible load, Btu/ h
ρ = air density assumed to equal 0.075 lb/ ft^3
c_p = specific heat of air = 0.24 Btu/ lb·°F
Δt = dry-bulb temperature difference between air entering and leaving equipment, °F
U = unit conversion factor, 1 h/60 min

Replacing all constant values gives the simplified equation in the given units.

$$Q = \frac{1}{1.08} \times \frac{q_s}{\Delta t} = \frac{q_s}{1.08 \Delta t} \tag{19.10}$$

For preliminary design, an approximate Δt is as follows:

Sensible Heat Ratio (SHR)	**Δt, °F**
0.75 to 0.79	21
0.80 to 0.85	19
0.85 to 0.90	17

SHR = Calculated sensible load/Calculated total load

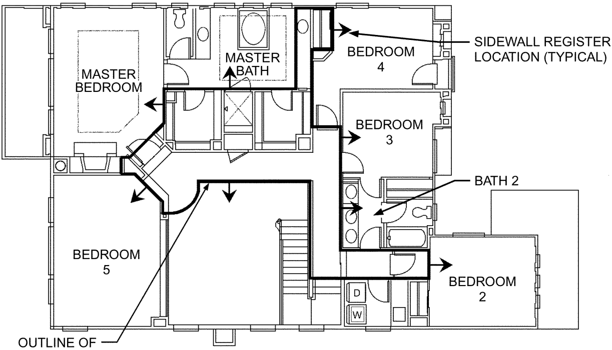
For example, if calculation indicates the sensible load is 23,000 Btu/ h and the latent load is 4900 Btu/ h, the SHR is calculated as follows:

$$\text{SHR} = \frac{23{,}000}{23{,}000 + 4900} = 0.82 \tag{19.11}$$

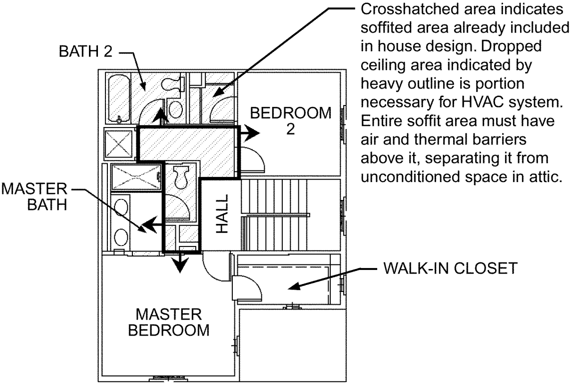
and

$$Q = \frac{23{,}000}{1.08 + 19} = 1121 \text{ cfm} \tag{19.12}$$

This value is the estimated design flow. The exact design flow can only be determined after the cooling unit is selected. The unit that is ultimately selected should supply an airflow in the range of the estimated flow, and must also have adequate sensible and latent cooling capacity when operating at design conditions.



(A)



(B)

Figure 19.15 Sample Floor Plans for Locating Ductwork in Second Floor of (A) Two-Story House and (B) Townhouse [2016S, Ch 10, Fig 2]
(Hedrick 2002)

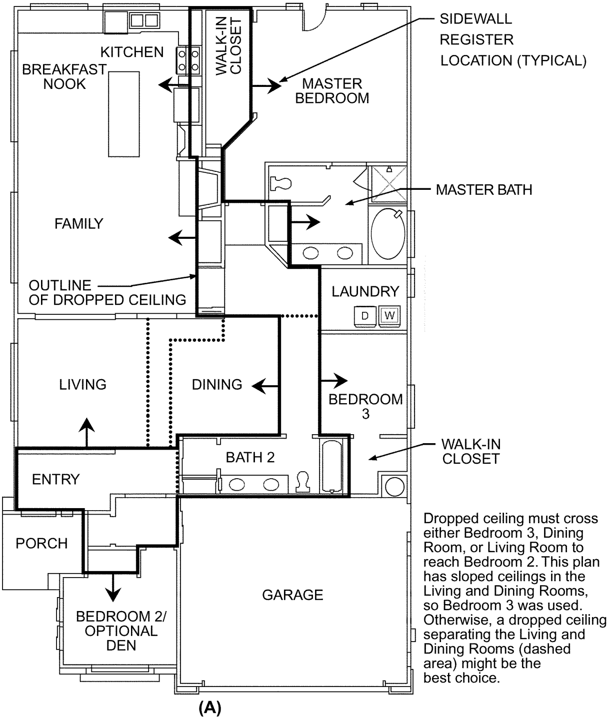
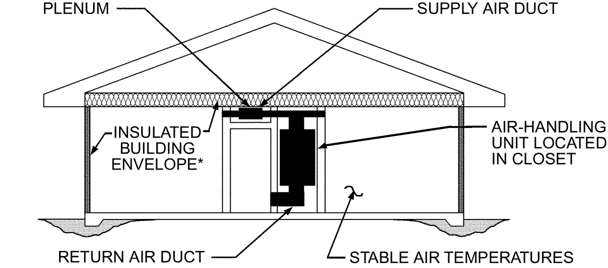
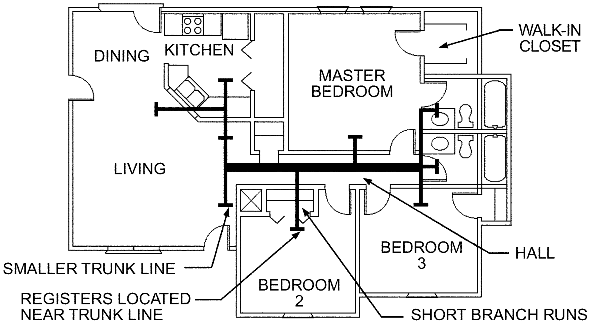


Figure 19.16 Sample Floor Plans for One-Story House with (A) Dropped Ceilings, (B) Ducts in Conditioned Spaces, and (C) Right-Sized Air Distribution in Conditioned Spaces
[2016S, Ch 10, Fig 3]
(EPA 2000)



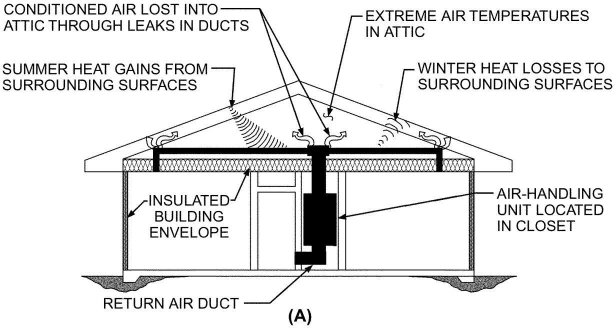
*To minimize duct leakage to attic, continuous drywall or other air barrier is needed at ceiling below insulation in all plenum areas.

(B)



(C)

Figure 19.16 Sample Floor Plans for One-Story House with (A) Dropped Ceilings, (B) Ducts in Conditioned Spaces, and (C) Right-Sized Air Distribution in Conditioned Spaces
[2016S, Ch 10, Fig 3] ***(Continued)***
(EPA 2000)



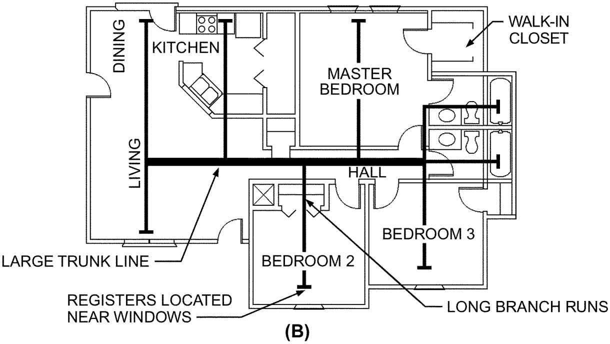


Figure 19.17 (A) Ducts in Unconditioned Spaces and (B) Standard Air Distribution System in Unconditioned Spaces [2016S, Ch 10, Fig 4]
(EPA 2000)

Unitary Air Conditioners and Heat Pumps

Unitary air conditioners are factory-made assemblies that nor-mally include an evaporator or cooling coil and a compressor/ condenser combination, and possibly provide heating as well. An **air-source unitary heat pump** normally includes an indoor conditioning coil, compressor(s), and an outdoor coil. It must provide heating and possibly cooling as well. A **water-source heat pump** rejects or extracts heat to and from a water loop instead of from ambient air. A unitary air conditioner or heat pump with more than one factory-made assembly (e.g., indoor and outdoor units) is commonly called a **split system**.

Unitary equipment is divided into three general categories: residential, light commercial, and commercial. Residential equipment is single-phase unitary equipment with a cooling capacity of 65,000 Btu/h or less and is designed specifically for residential application. Light commercial equipment is generally three-phase, with cooling capacity up to 135,000 Btu/h, and is designed for small businesses and commercial properties. Commercial unitary equipment has cooling capacity higher than 135,000 Btu/h and is designed for large commercial buildings.

Unitary equipment is available in many configurations, such as

- **Single-zone, constant-volume**, which consists of one controlled space with one thermostat that controls to maintain a set point. This equipment may be single stage, multistage, or variable capacity.
- **Multizone, constant-volume**, which has several controlled spaces served by one unit that supplies air of different temperatures to different zones as demanded (Figure 19.18).
- **Single-package, variable-volume**, which consists of several controlled spaces served by one unit. Supply air from the unit is at a constant temperature, with air volume to each space varied to satisfy space demands (Figure 19.19).
- **Multisplit,** which consists of several controlled spaces, each served by a separate indoor unit. All indoor units are connected to an outdoor condensing unit (Figure 19.20). When each indoor unit varies its refrigerant flow in response to heating or cooling load demand, the system is called **variable refrigerant flow (VRF)**.

In general, roof-mounted single-package unitary equipment is limited to five or six stories because duct space and available blower power become excessive in taller buildings. Split units are limited by the maximum distance allowed between the indoor and outdoor sections because of losses in refrigerant piping, compressor capability, and refrigerant oil management. Indoor, single-zone equipment is generally less expensive to maintain and service than multizone or multisplit units.

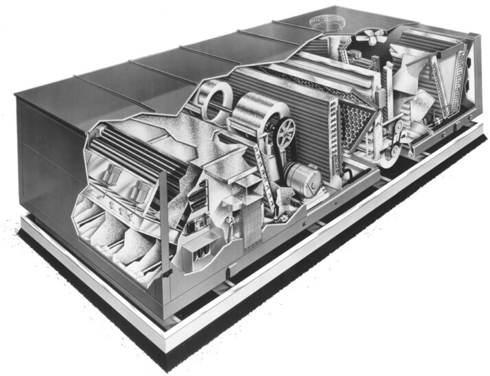


Figure 19.18 Typical Rooftop Air-Cooled Single-Package Air Conditioner [2016S, Ch 49, Fig 1]

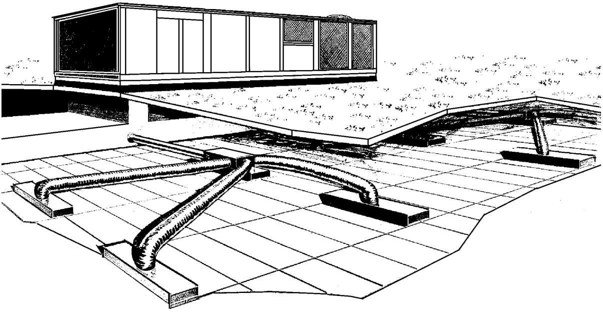


Figure 19.19 Single-Package Air Equipment with Variable Air Volume [2016S, Ch 49, Fig 2]

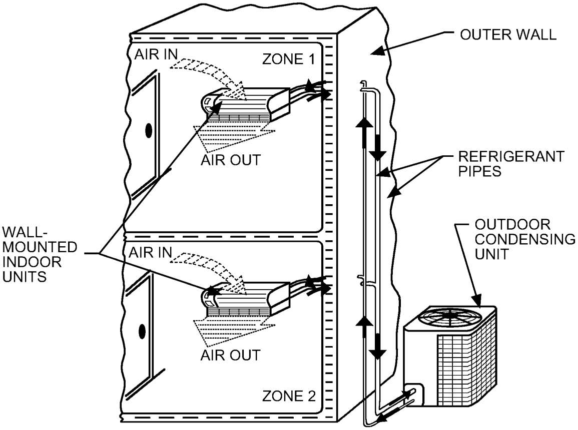


Figure 19.20 Example of Two-Zone Ductless Multisplit System in Typical Residential Installation [2016S, Ch 49, Fig 3]

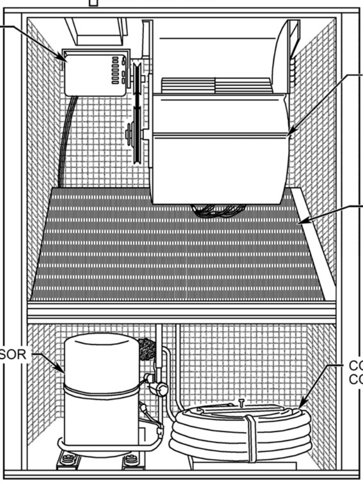


Figure 19.21 Water-Cooled Single-Package Air Conditioner [2016S, Ch 49, Fig 4]

Manufacturers' literature has detailed information about geometry, performance, electrical characteristics, application, and operating limits. The system designer selects suitable equipment with the capacity for the application.

Unitary equipment is designed to keep installation costs low. Adequate planning is important for installing large, roof-mounted equipment because special rigging equipment is frequently required. ACCA Standard 5 describes minimum criteria for the proper installation of HVAC systems in residential and commercial installations.

An advantage of packaged unitary equipment is that proper installation minimizes the risk of field contamination of the circuit. Care must be taken to properly install split-system interconnecting tubing (e.g., proper cleanliness, brazing, and evacuation to remove moisture and other noncondensables). Split systems should be charged according to the manufacturer's instructions. Filter-driers are necessary; if they are not installed at the factory, they should be field installed. When installing split, multisplit, and VRF systems, lines must be properly routed and sized to ensure proper oil return to the compressor.

Unitary equipment must be located to avoid noise and vibration problems. Single-package equipment of over 20 ton capacity should be mounted on concrete pads if vibration control is a concern. Large-capacity equipment should be roof mounted only after the roof's structural adequacy has been evaluated. If they are located over occupied space, roof-mounted units with return fans that use ceiling space for the return plenum should have a lined return plenum according to the manufacturer's recommendations. Use duct silencers where low sound levels are desired. Weight and sound data are available from many manufacturers. Additional installation guidelines include the following:

- In general, install products containing compressors on solid, level surfaces.
- Avoid mounting products containing compressors (such as remote units) on or touching the foundation of a house or building, or outside bedroom windows. A separate pad that does not touch the foundation is recommended to reduce any noise and vibration transmission through the slab.

- Do not box in outdoor air-cooled units with fences, walls, overhangs, or bushes. Doing so reduces the air-moving capability of the unit, reducing efficiency. Manufacturers include minimum clearances in literature.
- For a split-system remote unit, choose an installation site that is close to the indoor part of the system to minimize pressure drop in the connecting refrigerant tubing. Comply with manufacturers' refrigerant line length limits and required accessories listed in their literature.
- For VRF units, locate the refrigerant pipes' headers so that the length of refrigerant pipes is minimized.
- Contact the manufacturer or consult installation instructions for further information on installation procedures.

Unitary equipment should be listed or certified by nationally recognized testing laboratories to ensure safe operation and compliance with government and utility regulations. Equipment should also be installed to comply with agency standards' rating and application requirements to ensure that it performs according to industry criteria. Larger and more specialized equipment often does not carry agency labeling. However, power and control wiring practices should comply with the *National Electrical Code*® (NFPA Standard 70). Consult local codes before the installation is designed, and consult local inspectors before installation.

Unitary air conditioners have factory-matched refrigerant circuit components that are applied in the field to fulfill the user's requirements. The manufacturer often incorporates a heating function compatible with the cooling system and a control system that requires minimal field wiring.

Products are available to meet the objectives of nearly any system. Many different heating sections (gas- or oil-fired, electric, or condenser reheat), air filters, and heat pumps, which are a specialized form of unitary product, are available. Such matched equipment, selected with compatible accessory items, requires little field design or field installation work.

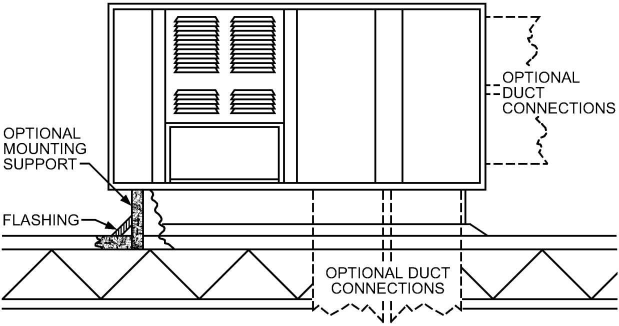


Figure 19.22 Rooftop Installation of Air-Cooled Single-Package Unit [2016S, Ch 49, Fig 5]

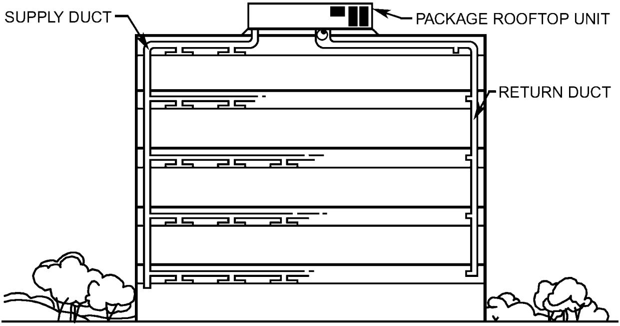


Figure 19.23 Multistory Rooftop Installation of Single-Package Unit [2016S, Ch 49, Fig 6]

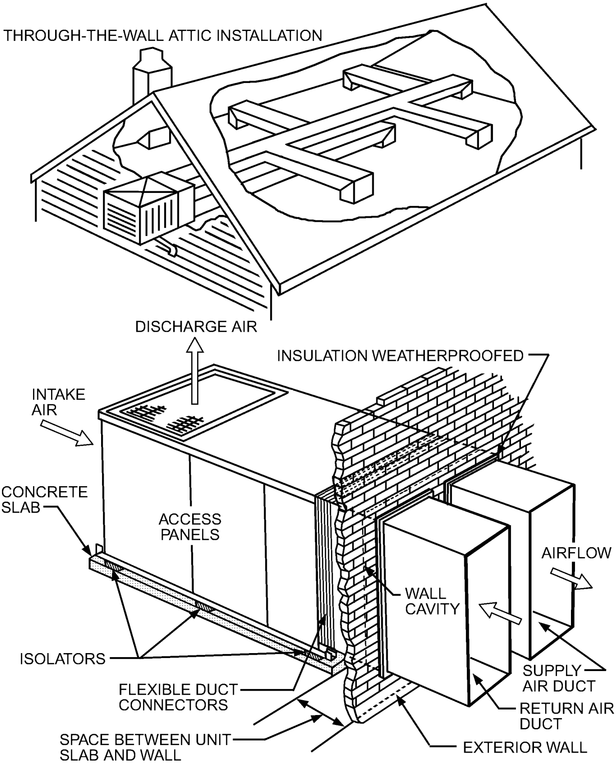


Figure 19.24 Through-the-Wall Installation of Air-Cooled Single-Package Unit
[2016S, Ch 49, Fig 7]

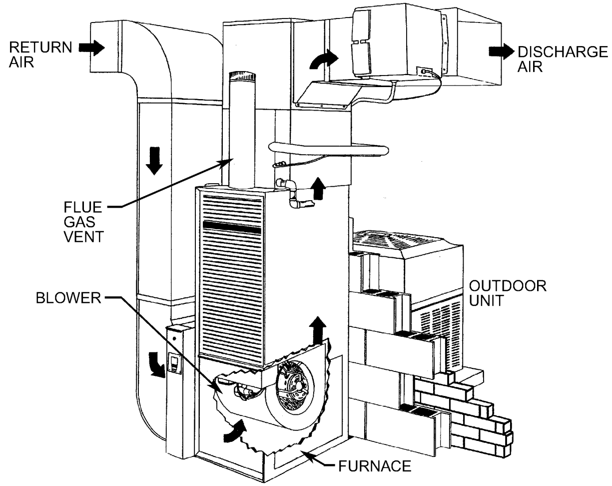


Figure 19.25 Residential Installation of Split-System Air-Cooled Condensing Unit with Coil and Upflow Furnace [2016S, Ch 49, Fig 8]

Evaporative Cooling

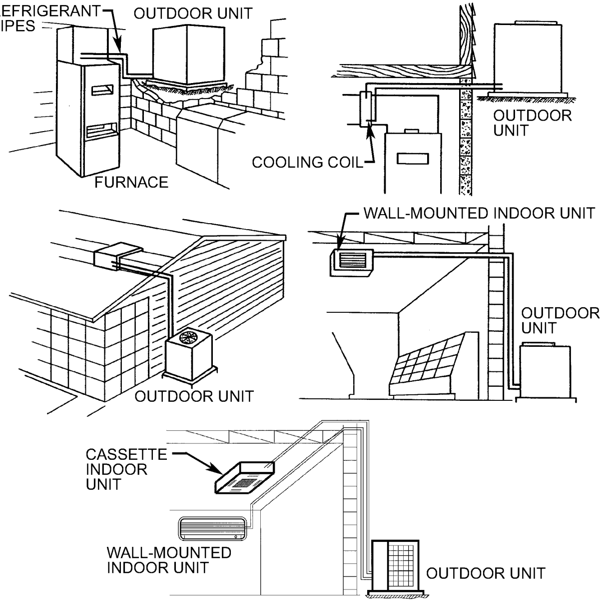


Figure 19.26 Outdoor Installations of Split-System Air-Cooled Condensing Units with Coil and Upflow Furnace or with Indoor Blower-Coils [2016S, Ch 49, Fig 9]

Evaporative Cooling

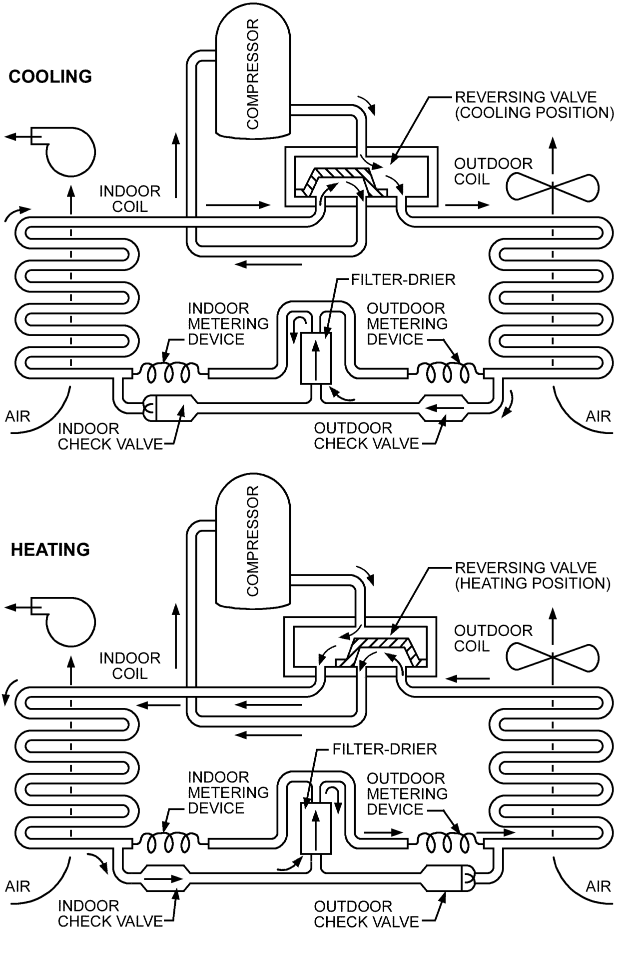


Figure 19.27 Schematic Typical of Air-to-Air Heat Pump System [2016S, Ch 49, Fig 12]

Evaporative Cooling

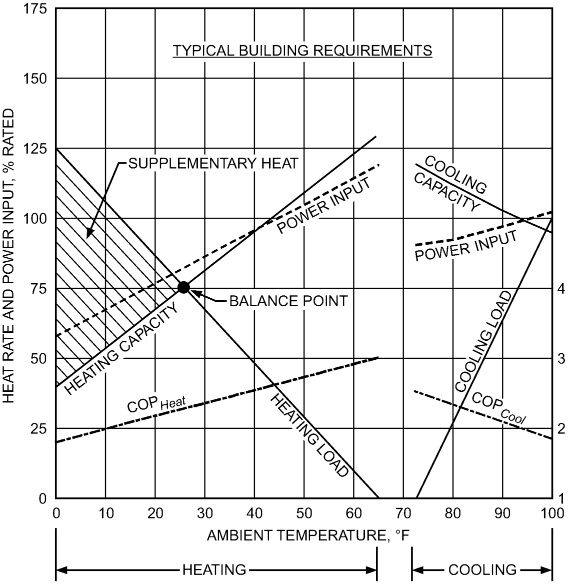


Figure 19.28 Operating Characteristics of Single-Stage Unmodulated Heat Pump
[2016S, Ch 49, Fig 13]

Evaporative Cooling

Water-Source Heat Pumps

A water-source heat pump (WSHP) is a single-package reverse-cycle heat pump that uses water as the heat source for heating and as the heat sink for cooling. The water supply may be a recirculating closed loop, a well, a lake, or a stream. Water for closed-loop heat pumps is usually circulated at 2 to 3 gpm per ton of cooling capacity.

WSHPs are used in a variety of systems, such as the following:

A **water-loop heat pump (WLHP)** uses a circulating water loop as the heat source and heat sink. When loop water temperature exceeds a certain level during cooling, a cooling tower dissipates heat from the water loop into the atmosphere. When loop water temperature drops below a prescribed level during heating, heat is added to the circulating loop water, usually with a boiler. In multiple-unit installations, some heat pumps may operate in cooling mode while others operate in heating, and controls are needed to keep loop water temperature within the prescribed limits. Chapter 9 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment* has more information on water-loop heat pumps.

A **groundwater heat pump (GWHP)** passes groundwater from a nearby well through the heat pump's water-to-refrigerant heat exchanger, where it is warmed or cooled, depending on the operating mode. It is then discharged to a drain, stream, or lake, or is returned to the ground through a reinjection well.

Many state and local jurisdictions have ordinances about use and discharge of groundwater. Because aquifers, the water table, and groundwater availability vary from region to region, these regulations cover a wide spectrum.

A **surface-water heat pump (SWHP)** uses water from a nearby lake, stream, or canal. After passing through the heat pump heat exchanger, it is returned to the source or a drain several degrees warmer or cooler, depending on the operating mode of the heat pump. **Closed-loop** surface water heat pumps use a closed water or brine loop that includes pipes or tubing submerged in the surface water (river, lake, or large pond) that serves as the heat exchanger. The adequacy of the total thermal capacity of the body of water must be considered.

A **ground-coupled heat pump (GCHP), ground-source heat pump (GSHP),** or **geothermal heat pump (GHP)** system uses the earth as a heat source and sink. Usually, plastic piping is installed in either a shallow horizontal or deep vertical array to form the heat exchanger. The massive thermal capacity of the earth provides a temperature-stabilizing effect on the circulating loop water or brine. Installing this type of system requires detailed knowledge of the climate; site; soil temperature, moisture content, and thermal characteristics; and performance, design,

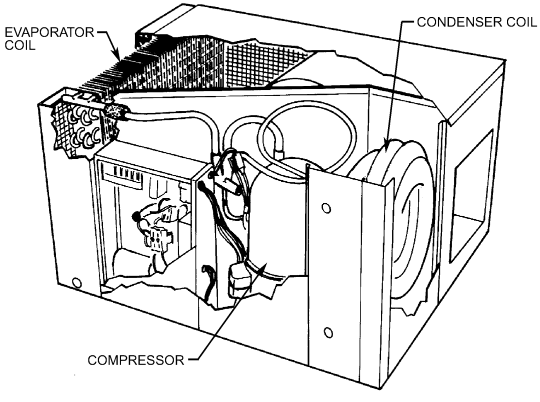


Figure 19.29 Typical Horizontal Water-Source Heat Pump [2016S, Ch 49, Fig 15]

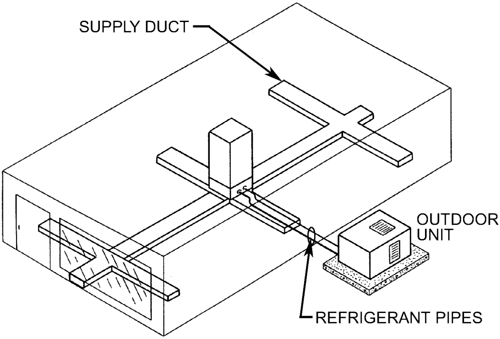


Figure 19.30 Outdoor Installation of Split-System Air-Cooled Condensing Unit with Indoor Coil and Downflow Furnace [2016S, Ch 49, Fig 10]

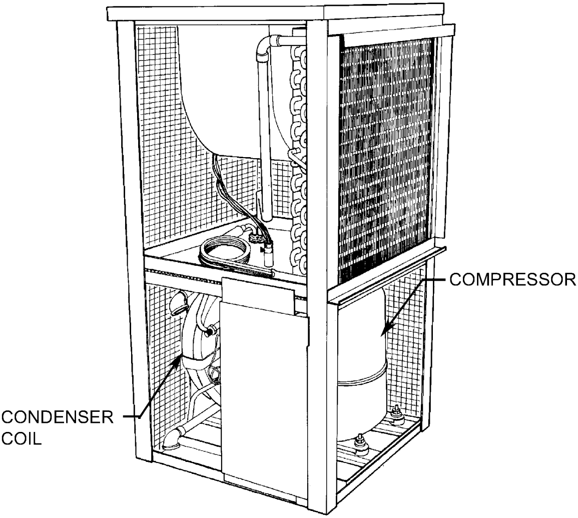


Figure 19.31 Typical Vertical Water-Source Heat Pump [2016S, Ch 49, Fig 16]

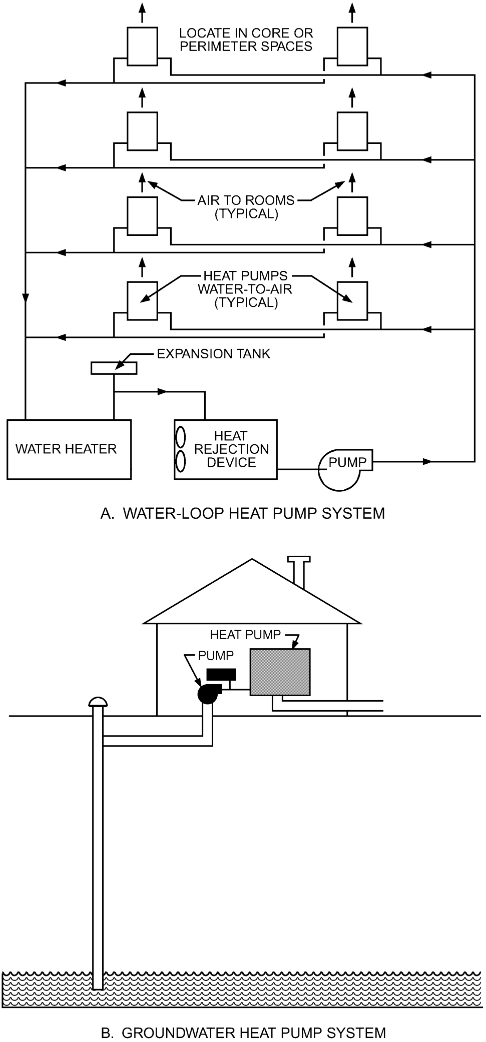
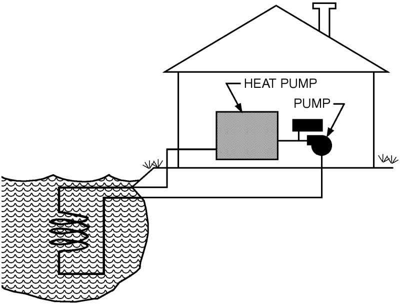
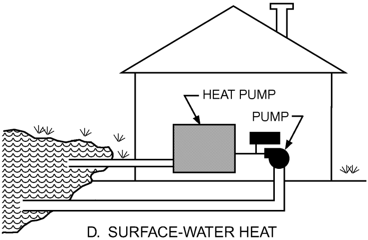


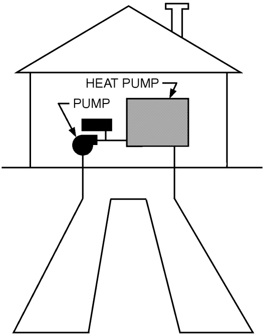
Figure 19.32 Water-Source Heat Pump Systems [2016S, Ch 49, Fig 17]



C. CLOSED-LOOP SURFACE-WATER
HEAT PUMP SYSTEM



D. SURFACE-WATER HEAT
PUMP SYSTEM



E. GROUND-COUPLED HEAT
PUMP SYSTEM

Figure 19.32 Water-Source Heat Pump Systems [2016S, Ch 49, Fig 17]
(Continued)

and installation of water-to-earth heat exchangers. Additional information on GCHP systems is presented in Chapter 34 of the 2015 *ASHRAE Handbook—HVAC Applications*.

Entering Water Temperatures. These various water sources provide a wide range of entering water temperatures to WSHPs. Entering water temperatures vary not only by water source, but also by climate and time of year. Because of the wide range of entering water or brine temperatures encountered, it is not feasible to design a universal packaged product that can handle the full range of possibilities effectively. Therefore, WSHPs are rated for performance at a number of standard rating conditions.

Compressors. WSHPs usually have single-speed compressors, although some high-efficiency models use multispeed compressors. Higher-capacity equipment may use multiple compressors. Compressors may be reciprocating, rotary, or scroll. Single-phase units are available at voltages of 115, 208, 230, and 265. All larger equipment is for three-phase power supplies with voltages of 208, 230, 460, or 575. Compressors usually have electromechanical protective devices.

Indoor Air System. Console WSHP models are designed for free delivery of conditioned air. Other models have ducting capability. Smaller WSHPs have multispeed, direct-drive centrifugal blower wheel fan systems. Large-capacity equipment has belt-drive systems. All units have provisions for fiberglass, metal, or plastic foam air filters.

Indoor Air Heat Exchanger. The indoor air heat exchanger of WSHP units is usually a conventional plate-fin coil of copper tubes and aluminum fins. Microchannel evaporators are also used in some products. The indoor air heat exchanger must be circuited so that it can function effectively as an evaporator with refrigerant flow in one direction and as a condenser when refrigerant flow is reversed.

Refrigerant-to-Water Heat Exchanger. The heat exchanger, which couples the heat pump to source/sink water, is tube-in-tube, tube-in-shell, or brazed-plate. It must function in either condensing or evaporating mode, so special attention is given to refrigerant-side circuitry. Heat exchanger construction is usually of copper and steel, and the source/sink water is exposed only to the copper portions. Cupronickel options to replace the copper are usually available for use with brackish or corrosive water. Brazed-plate heat exchangers are usually constructed of stainless steel, which reduces the need for special materials.

Refrigerant Expansion Devices. These WSHPs operate over a narrow range of entering water temperatures and typically use simple capillaries as expansion devices. However, units may also use thermostatic expansion valves for improved performance over a broader range of inlet fluid temperatures.

Refrigerant-Reversing Valve. The refrigerant-reversing valves in WSHPs are identical to those used in air-source heat pumps.

Condensate Disposal. Condensate, which forms on the indoor coil when cooling, is collected and conveyed to a drain system.

Controls. Console WSHP units have built-in operating mode selector and thermostatic controls. Ducted units use low-voltage remote heat/cool thermostats.

Size. Typical space requirements and weights of WSHPs are presented in Table 19.4.

Special Features. Some WSHPs include the following:

Desuperheater. Uses discharge gas in a special water/refrigerant heat exchanger to heat water for a building.

Capacity modulation. May use multiple compressors, multispeed compressors, or hot-gas bypass.

Variable air volume (VAV). Reduces fan energy usage and requires some form of capacity modulation.

Automatic water valve. Closes off water flow through the unit when the compressor is off and allows variable water volume in the loop, which reduces pumping energy.

Outdoor air economizer. Cools directly with outdoor air to reduce or eliminate the need for mechanical refrigeration during mild or cold weather when outdoor humidity levels and air quality are appropriate.

Water-side economizer. Cools with loop water to reduce or eliminate the need for mechanical refrigeration during cold weather; requires a hydronic coil in the indoor air circuit that is valved into the circulating loop when loop temperatures are relatively low and cooling is required.

Electric heaters. Used in WLHP systems that do not have a boiler as a source for loop heating.

Table 19.4 Space Requirements for Typical Packaged Water-Source Heat Pumps
[2016S, Ch 49, Tbl 3]

Water-to-Air Heat Pump	Length × Width × Height, ft	Weight, lb
1.5 ton vertical unit	2.0 × 2.0 × 3.0	180
3 ton vertical unit	2.5 × 2.5 × 4.0	250
3 ton horizontal unit	3.5 × 2.0 × 2.0	250
5 ton vertical unit	3.0 × 2.5 × 4.0	330
11 ton vertical unit	3.5 × 3.0 × 6.0	720
26 ton vertical unit	3.5 × 5.0 × 6.0	1550

Note: See manufacturers' specification sheets for actual values.

Variable-Refrigerant-Flow Heat Pumps

A variable-refrigerant-flow (VRF) system typically consists of a condensing section housing compressor(s) and condenser heat exchanger interconnected by a single set of refrigerant piping to multiple indoor direct-expansion (DX) evaporator fan-coil units. Thirty or more DX fan-coil units can be connected to a single condensing section, depending on system design, and with capacity ranging from 0.5 to 8 tons.

The DX fan-coils are constant air volume, but use variable refrigerant flow through an electronic expansion valve. The electronic expansion valve reacts to several temperature-sensing devices such as return air, inlet and outlet refrigerant temperatures, or suction pressure. The electronic expansion valve modulates to maintain the desired set point.

Application. VRF systems are most commonly air-to-air, but are also available in a water-source (water-to-refrigerant) configuration. They can be configured for simultaneous heating and cooling operation (some indoor fan-coil units operating in heating and some in cooling, depending on requirements of each building zone).

Indoor units are typically direct-expansion evaporators using individual electronic expansion devices and dedicated microprocessor controls for individual control. Each indoor unit can be controlled by an individual thermostat. The outdoor unit may connect several indoor evaporator units with capacities 130% or more than the outdoor condensing unit capacity.

Categories. VRF equipment is divided into three general categories: residential, light commercial, and applied. Residential equipment is single-phase unitary equipment with a cooling capacity of 65,000 Btu/h or less. Light commercial equipment is generally three-phase, with cooling capacity greater than 65,000 Btu/h, and is designed for small businesses and commercial properties. Applied equipment has cooling capacity higher than 135,000 Btu/h and is designed for large commercial buildings.

Refrigerant Circuit and Components. VRF heat pump systems use a two-pipe (liquid and suction gas) system; simultaneous heat and cool systems use the same system, as well as a gas flow device that determines the proper routing of refrigerant gas to a particular indoor unit.

VRF systems use a sophisticated refrigerant circuit that monitors mass flow, oil flow, and balance to ensure optimum performance. This is accomplished in unison with variable-speed compressors and condenser fan motors. Both of these components adjust their frequency in reaction to changing mass flow conditions and refrigerant operating pressures and temperatures. A dedicated microprocessor continuously monitors and controls these key components to ensure proper refrigerant is delivered to each indoor unit in cooling or heating.

Heating and Defrost Operation. In heating mode, VRF systems typically must defrost like any mechanical heat pump, using reverse-cycle valves to temporarily operate the outdoor coil in cooling mode. Oil return and balance with the refrigerant circuit is managed by the microprocessor to ensure that any oil entrained in the low side of the system is brought back to the high side by increasing the refrigerant velocity using a high-frequency operation performed automatically based on hours of operation.

More information on VRF heat pumps can be found in Chapter 18 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*, and information on performance-rating these systems can be found in AHRI Standard 1230-2010.

Room Air Conditioners and Packaged Terminal Air Conditioners

Room Air Conditioners

Room air conditioners are encased assemblies designed prima-rily for mounting in a window or through a wall. They are designed to deliver cool or warm conditioned air to the room, either without ducts or with very short ducts (up to a maximum of about 48 in.). Each unit includes a prime source of refrigeration and dehumidification and a means for circulating and filtering air; it may also include a means for ventilating and/or exhausting and heating.

The basic function of a room air conditioner is to provide comfort by cooling, dehumidifying, filtering or cleaning, and circulating the room air. It may also provide ventilation by introducing outdoor air into the room and/or exhausting room air to the outdoors. Room temperature may be controlled by an integral thermostat. The conditioner may provide heating by heat pump operation, electric resistance elements, or a combination of the two.

Figure 19.33 shows a typical room air conditioner in cooling mode. Warm room air passes over the cooling coil and transfers sensible and latent heat. The conditioned air is then recirculated in the room by a fan or blower.

Heat from the warm room air vaporizes the cold (low-pressure) liquid refrigerant flowing through the evaporator. The vapor then carries the heat to the compressor, which compresses the vapor and increases its temperature above that of the outdoor air. In the condenser, the hot (high-pressure) refrigerant vapor liquefies, transferring the heat from the room air to outdoor air. Next, the high-pressure liquid refrigerant passes through a restrictor, which reduces its pressure and temperature. The cold (low-pressure) liquid refrigerant then enters the evaporator to repeat the refrigeration cycle.

Room air conditioners have line cords, which may be plugged into standard or special electric circuits. Most units in the United States are designed to operate at 115, 208, or 230 V; single-phase; 60 Hz power. Some units are rated at 265 V or 277 V, for which the chassis or chassis assembly must provide permanent electrical connection. The maximum amperage of 115 V units is generally 12 A, which is the maximum current permitted by NFPA Standard 70 [the *National Electrical Code*® (NEC)] for a single-outlet, 15 A circuit. Models designed for countries other

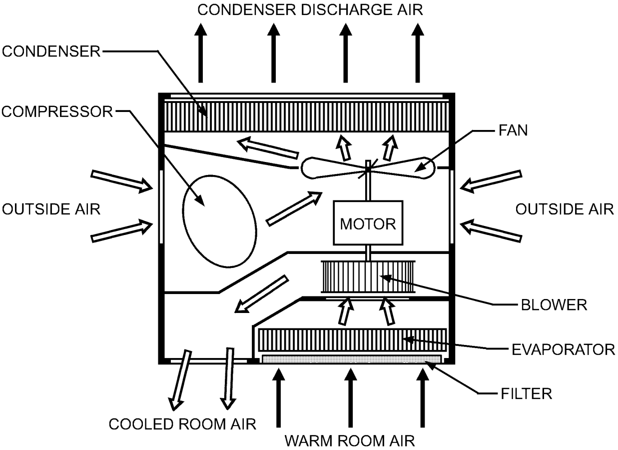


Figure 19.33 Schematic View of Typical Room Air Conditioner [2016S, Ch 50, Fig 1]

than the United States are generally for 50 or 60 Hz systems, with typical design voltage ranges of 100 to 120 and 200 to 240 V, single-phase.

Popular 115 V models have capacities in the range of 5000 to 8000 Btu/h, and are typically used in single-room applications. Larger-capacity 115 V units are in the 12,000 to 15,000 Btu/h range. Capacities for 230, 208, or 230/208 V units range from 8000 to 36,000 Btu/h. These higher-voltage units are typically used in multiple-room installations.

Heat pump models are also available, usually for 208 or 230 V applications. These units are generally designed for reversed-refrigerant-cycle operation as the normal means of supplying heat, but may incorporate electrical-resistance heat either to supplement heat pump capacity or to provide the total heating capacity when outdoor temperatures drop below a set value.

Another type of heating model incorporates electrical heating elements in regular cooling units so that heating is provided entirely by electrical resistance heat.

Installation procedures vary because units can be mounted in various ways. It is important to select the mounting for each installation that best satisfies the user and complies with applicable building codes. Common mounting methods include the following:

- **Indoor flush mounting.** Interior face of conditioner is approximately flush with inside wall.
- **Balance mounting.** Unit is approximately half inside and half outside window.
- **Outdoor flush mounting.** Outer face of unit is flush with or slightly beyond outside wall.
- **Special mounting.** Examples include casement windows, horizontal sliding windows, and office windows with swinging units (or swinging windows) to allow window washing, and transoms over doorways.
- **Through-the-wall mounts or sleeves.** This mounting is used for installing window-type chassis, complete units, or consoles in walls of apartment buildings, hotels, motels, and residences. Although very similar to window-mounted units, through-the-wall models do not have side louvers for condenser air; air comes from the outdoor end of the unit.

Room air conditioners have become more compact to minimize both loss of window light and projection inside and outside the structure. Several types of expandable mounts are now available for fast, dependable installation in single- and double-hung windows, as well as in horizontal sliding windows. Window air conditioners should fit in the opening without too much space around the sides; most units come with accordion-style flaps attached to the sides, used to fill the open space around the unit. Installation kits include all parts needed for structural mounting, such as gaskets, panels, and seals for weathertight assembly. Sealing the space around the unit (e.g., with caulk or foam seals) is important to prevent air from escaping and limit seeping of hot outdoor air into the conditioned space, and to achieve energy savings during operation.

Adequate wiring and proper breakers or fuses must be provided for the service outlet. Necessary information is usually given on instruction sheets or stamped on the air conditioner near the service cord or on the serial plate. It is important to follow the manufacturer's recommendation for size and type of breaker or fuse. All units are equipped by the manufacturer with grounding plug caps on the service cord. Receptacles with grounding contacts correctly designed to fit these plug caps should be used when units are installed.

Units rated 265 or 277 V must provide for permanent electrical connection with armored cable or conduit to the chassis or chassis assembly. Manufacturers usually provide an adequate cord and plug cap in the chassis assembly to facilitate installation and service.

One type of room air conditioner is the integral chassis design, with the outer cabinet fastened permanently to the chassis. Most electrical components can be serviced by partially dismantling the control area without removing the unit from the installation. Another type is the **slide-out chassis** design, which allows the outer cabinet to remain in place while the chassis is removed for service.

Effective condensate management should be considered during the installation of the air conditioner. Many window air conditioners have connection points for condensate drains. Follow plumbing and building code requirements for handling discharge of the condensate produced when the air conditioner is operating.

Packaged Terminal Air Conditioners

A packaged terminal air conditioner (PTAC) includes a wall sleeve and a separate unencased combination of heating and cooling assemblies intended for mounting through the wall. A PTAC includes refrigeration components, separable outdoor louvers, forced ventilation, and heating by hot water, steam, or electric resistance. PTAC units with indirect-fired gas heaters are also available from some manufacturers. A packaged terminal heat pump (PTHP) is a heat pump version of a PTAC that provides heat with a reverse-cycle operating mode. A PTHP should provide a supplementary heat source, which can be hot water, steam, electric resistance, or another source.

PTACs are designed primarily for commercial installations to provide the total heating and cooling functions for a room or zone and are specifically for through-the-wall installation. The units are mostly used in relatively small zones on the perimeter of buildings such as hotels and motels, apartments, hospitals, nursing homes, and office buildings. In larger buildings, they may be combined with nearly any system selected for environmental control of the building core.

PTACs and PTHPs are similar in design and construction. The most apparent difference is the addition of a refrigerant-reversing valve in the PTHP. Optional components that control the heating functions of the heat pump include an outdoor thermostat to signal the need for changes in heating operating modes, and, in more complex designs, frost sensors, defrost termination devices, and base pan heaters.

PTACs/PTHPs are available in a wide range of rated cooling capacities, typically 6000 to 18,000 Btu/h, with comparable levels of heating output. Units are available as sectional types or integrated types. A sectional-type unit (Figure 19.34) has a separate cooling chassis; an integrated-type unit (Figure 19.35) has an electric or a gas heating option added to the chassis. Hot-water or steam heating options are usually part of the cabinet or wall box. Both types include the following:

- Heating elements available in hot water, steam, electric, or gas heat
- Integral or remote temperature and operating controls
- Wall sleeve or box
- Removable (or separable) outdoor louvers
- Room cabinet
- Means for controlled forced ventilation
- Means for filtering air delivered to the room

PTAC assemblies are intended for use in free conditioned-air distribution, but a particular application may require minimal ductwork with a total external static resistance up to 0.1 in. of water.

Packaged terminal air conditioners and packaged terminal heat pumps allow the HVAC designer to integrate the exposed outdoor louver or grille with the building design. Various grilles are available to blend with or accent most construction materials. Because the product becomes part of the building's facade, the architect must consider the product during the conception of the building. Wall sleeve installation is usually done by ironworkers, masons, or carpenters. All-electric units dominate the market.

All the energy of all-electric versions is dispersed through the building via electrical wiring, so the electric designer and electrical contractor play a major role. Final installation is reduced to sliding in the chassis and plugging the unit into an adjacent receptacle. For these all-electric units, the traditional HVAC contractor's work involving ducting, piping, and refrigeration systems is bypassed. This results in a low-cost installation and allows installation of the PTAC/PTHP chassis to be deferred until just before occupancy.

When comparing a gas-fired PTAC to a PTAC with electric resistance heat or a PTHP, evaluate both operating and installation costs. Generally, a gas-fired PTAC is more expensive to install but less expensive to operate in heating mode. A life-cycle cost comparison is recommended (see Chapter 37 of the 2015 *ASHRAE Handbook—HVAC Applications*).

One main advantage of the PTAC/PTHP concept is that it provides excellent zoning capability. Units can be shut down or operated in a holding condition during unoccupied periods. Present equipment efficiency-rating criteria are based on full-load operation, so an efficiency comparison to other approaches may suffer.

The designer must also consider that total capacity is the sum of the peak loads of each zone rather than the peak load of the building. Therefore, total cooling capacity of the zonal system will exceed that of a central system.

Because PTAC units are located in the conditioned space, both appearance and sound level of the equipment are important considerations. Sound attenuation in ducting is not available with the free-discharge PTAC units.

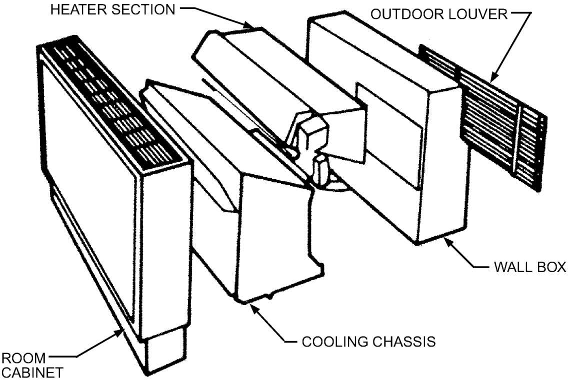


Figure 19.34 Sectional Packaged Terminal Air Conditioner [2016S, Ch 50, Fig 2]

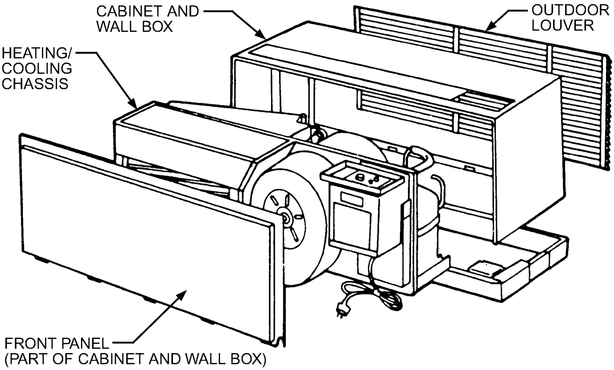


Figure 19.35 Integrated Packaged Terminal Air Conditioner [2016S, Ch 50, Fig 3]

The designer must also consider the added infiltration and thermal leakage load resulting from perimeter wall penetrations. These losses are accounted for during the *on* cycle in equipment cooling ratings and PTHP heating ratings, but during the *off* cycle or with other forms of heating, they could be significant.

Most packaged terminal equipment is designed to fit into a wall aperture approximately 42 in. wide and 16 in. high. Although unitary products can increase in size with increasing cooling capacity, PTAC/PTHP units, regardless of cooling capacity, are usually constrained to a few cabinet sizes. The exterior of the equipment must be essentially flush with the exterior wall to meet most building codes. In addition, cabinet structural requirements and the slide-in chassis reduce the available area for outdoor air inlet and relief to less than a total of 3.5 ft^2. Manufacturers' specification sheets should be consulted for more accurate and detailed information.

Basic PTHP units can operate in heat pump mode until outdoor temperature is just above the point at which the outdoor heat exchanger would frost. At that point, heat pump mode is locked out, and other forms of heating are required. Some units include two-stage indoor thermostats and automatically switch from heat pump mode to an alternative heat source if space temperature drops too far below the first-stage set point. Some PTHPs use control schemes that extend heat pump operation to lower temperatures. One approach allows heat pump operation down to outdoor temperatures just above freezing. If the outdoor coil frosts, it is defrosted by shutting down the compressor and allowing the outdoor fan to continue circulating outdoor air over the coil. Another approach allows heat pump operation to even lower outdoor temperatures by using a reverse-cycle defrost sequence. In those cases, the heat pump mode is usually locked out for outdoor temperatures below 10°F.

EVAPORATIVE COOLING

Direct Evaporative Air Coolers

Air is drawn through porous wetted pads or a spray, or rigid media; and its sensible heat energy evaporates some water. The heat and mass transfer between the air and water lowers the air dry-bulb temperature and increases the humidity at a constant wet-bulb temperature. The dry-bulb temperature of the nearly saturated air approaches the ambient air's wet-bulb temperature. The process is adiabatic, so no sensible cooling occurs.

The extent to which the leaving air temperature from a direct evaporative cooler approaches the thermodynamic wet-bulb temperature of the entering air or the extent to which complete saturation is approached is expressed as the **direct saturation efficiency**, defined as

$$\varepsilon_e = 100 \frac{t_1 - t_2}{t_1 - t_s'} \tag{19.13}$$

where

ε_e	=	direct evaporative cooling or saturation efficiency, %
t_1	=	dry-bulb temperature of entering air, °F
t_2	=	dry-bulb temperature of leaving air, °F
t_s'	=	thermodynamic wet-bulb temperature of entering air, °F

An efficient wetted pad can reduce the air dry-bulb temperature by as much as 95% of the wet-bulb depression (ambient dry-bulb temperature less wet-bulb temperature), while an inefficient and poorly designed pad may only reduce this by 50% or less.

Direct evaporative cooling, though simple and inexpensive, has the disadvantage that if the ambient wet-bulb temperature is higher than about 70°F, the cooling effect is not sufficient for indoor comfort but still may be sufficient for relief cooling applications. Direct evaporative coolers should not recirculate indoor air.

Two-inch pad coolers, usually small capacity, operate at 100 to 250 fpm face velocity. Twelve-inch-deep rigid media larger coolers operate at 400 to 600 fpm face velocity and have higher saturation efficiencies.

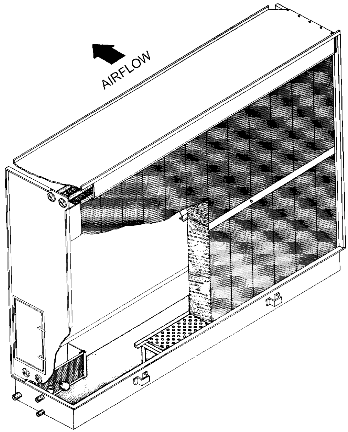


Figure 19.36 Rigid Media Direct Evaporative Cooler [2016S, Ch 41, Fig 2]

Indirect Evaporative Air Coolers

In indirect evaporative air coolers, outdoor air or exhaust air from the conditioned space passes through one side of a heat exchanger. This secondary airstream is cooled by evaporation by direct wetting of the heat exchanger surface, or passing through evaporative cooling media, atomizing spray, or disk evaporator. The surfaces of the heat exchanger are cooled by the secondary airstream. On the other side of the heat exchanger surface, the primary airstream (conditioned air to be supplied to the space) is sensibly cooled.

Although the primary air is cooled by secondary air, no moisture is added to the primary air. Because the enthalpy of the primary air decreases, the leaving dry-bulb temperature of the primary air must always be above the entering wet-bulb temperature of the secondary airstream. Dehumidifying in the primary airstream can occur only when the dew point of the primary airstream is several degrees higher than the wet-bulb temperature of the secondary air-stream. This condition exists only when the secondary airstream is drier than the primary airstream, such as when building exhaust air is used for the secondary air.

Indirect evaporative cooling efficiency, or **wet-bulb depression efficiency (WBDE),** is defined as

$$\text{WBDE} = 100\frac{t_1 - t_2}{t_1 - t_s'} \tag{19.14}$$

where

WBDE	=	indirect evaporative cooling efficiency,%
t_1	=	dry-bulb temperature of entering primary air, °F
t_2	=	dry-bulb temperature of leaving primary air, °F
t_s'	=	wet-bulb temperature of entering secondary air, °F

In a two-stage indirect/direct evaporative cooler, a first-stage indirect evaporative cooler lowers both the dry- and wet-bulb temperature of the incoming air. After leaving the indirect stage, the supply air passes through a second-stage direct evaporative cooler.

This method can lower the supply air dry-bulb temperature by 10°F or more below the secondary air wet-bulb temperature.

In areas with a higher wet-bulb design temperature or where the design requires a supply air temperature lower than that attainable using indirect/direct evaporative cooling, a third cooling stage may be required. This stage may be a direct-expansion refrigeration unit or a chilled-water coil located either upstream or downstream from the direct evaporative cooling stage, but always downstream from the indirect evaporative stage.

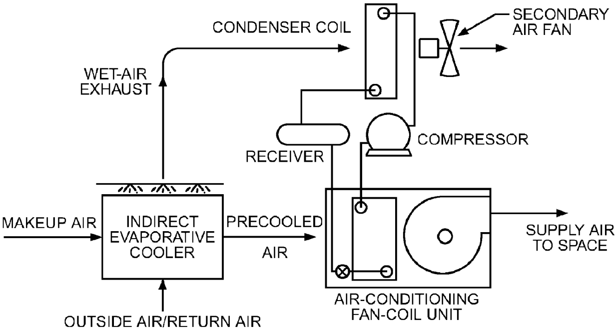


Figure 19.37 Indirect Evaporative Cooler Used as Precooler [2016S, Ch 41, Fig 4]

Table 19.5 Indirect Evaporative Cooling Systems Comparison

System Type[a]	WBDE,[b] %	Heat Recovery Efficiency, %	Wet-Side Air ΔP, in. of water	Dry-Side Air ΔP, in. of water	Pump hp per 10,000 cfm	Parasitic Loss Range,[e] kW/ton of Cooling	Notes
Cooling tower to coil	40 to 60	NA	NA	0.4 to 0.7	Varies	Varies	Best for serving multiple AHUs from a single cooling tower. No winter heat recovery.
Crossflow plate	60 to 85	40 to 50	0.7 to 1.0	0.4 to 0.7	0.1 to 0.2	0.12 to 0.20	Most cost-effective for lower airflows. Some cross contamination possible. Low winter heat recovery.
Heat pipe[c]	65 to 75	50 to 60	0.7 to 1.0	0.5 to 0.7	0.2 to 0.4	0.15 to 0.25	Most cost-effective for large airflows. Some cross contamination possible. Medium winter heat recovery.
Heat wheel[d]	60 to 70	70 to 80	0.6 to 0.9	0.4 to 0.65	0.1 to 0.2	0.2 to 0.3	Best for high airflows. Some cross contamination. Highest winter heat recovery rates.
Runaround coil[δ]	35 to 50	40 to 60	0.6 to 0.8	0.4 to 0.6	Varies	> 0.35	Best for applications where supply and return air ducts are separated. Lowest summer WBDE.

WBDE = wet-bulb depression efficiency

Notes:
[a]All air-to-air heat exchangers have equal mass flow on supply and exhaust sides.
[b]Plate and heat pipe are direct spray on exhaust side. Heat wheel and runaround coil systems use 90% WBDE direct evaporative cooling media on exhaust air side.
[c]Assumes six-row heat pipe, 11 fpi, with 500 fpm face velocity on both sides.
[d]Assumes 500 fpm face velocity. Parasitic loss includes wheel rotational power.
[e]Includes air-side static pressure and pumping penalty.

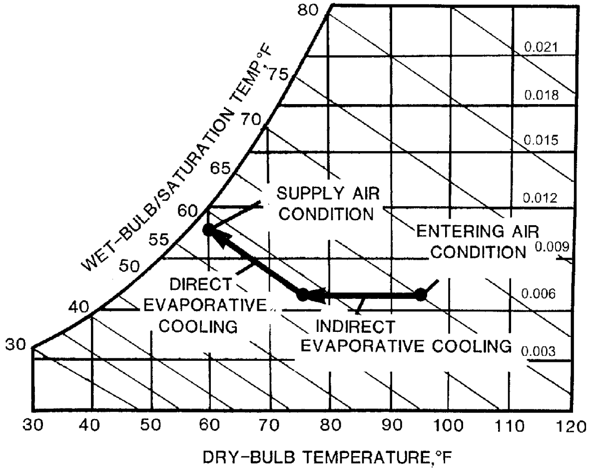


Figure 19.38 Two-Stage Indirect/Direct Evaporative Cooling Process [2016S, Ch 41, Fig 6]

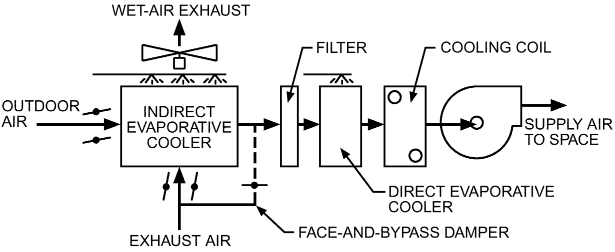


Figure 19.39 Three-Stage Indirect/Direct Evaporative Cooler [2016S, Ch 41, Fig 8]

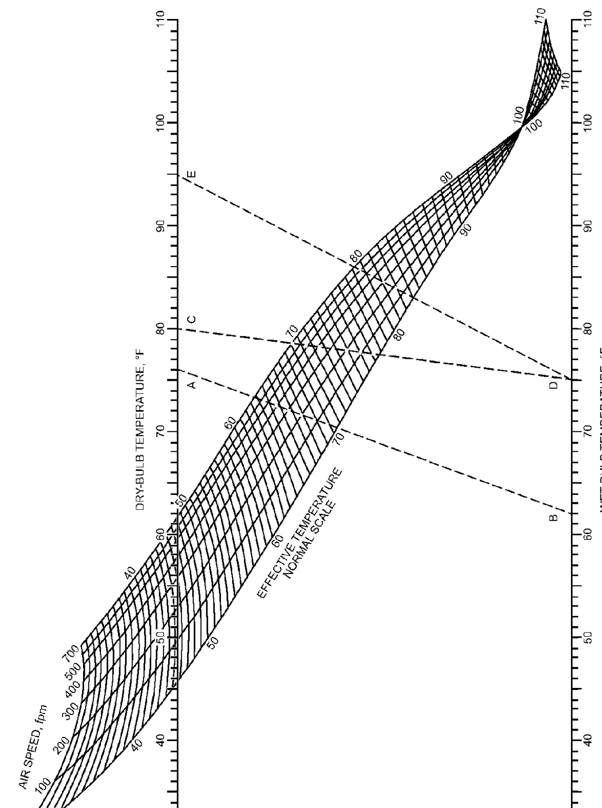


Figure 19.40 Effective Temperature Chart [2015A, Ch 52, Fig 14]

20. AUTOMATIC CONTROLS

HVAC System Components

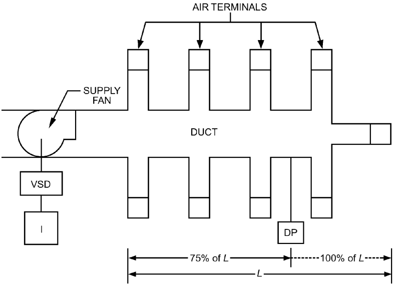


Figure 20.1 Duct Static Pressure Control [2015A, Ch 47, Fig 15]

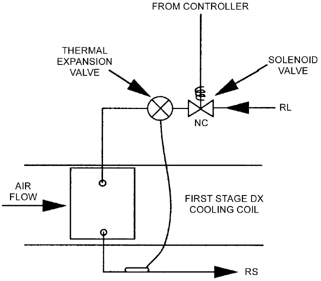


Figure 20.2 Direct Expansion—Two-Position Control

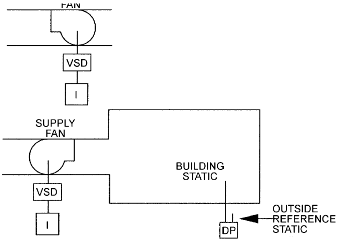


Figure 20.3 Duct Static Control of Return Fan

Automatic Controls

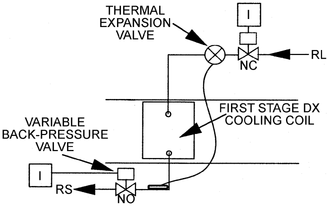


Figure 20.4 Modulating Direct-Expansion Cooling

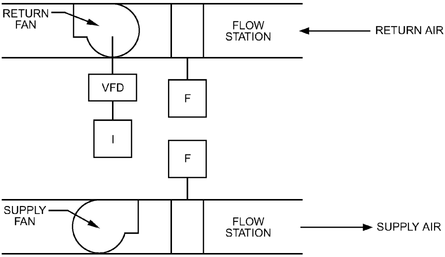


Figure 20.5 Airflow Tracking Control [2015A, Ch 47, Fig 17]

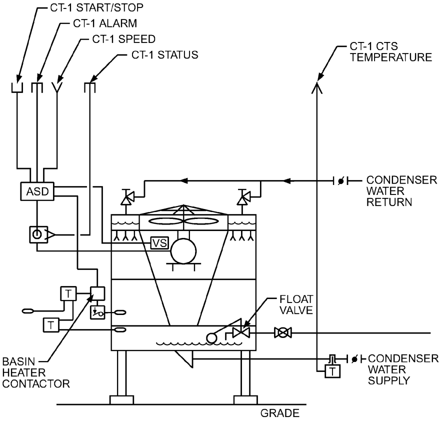


Figure 20.6 Cooling Tower [2015A, Ch 47, Fig 13]

Automatic Controls

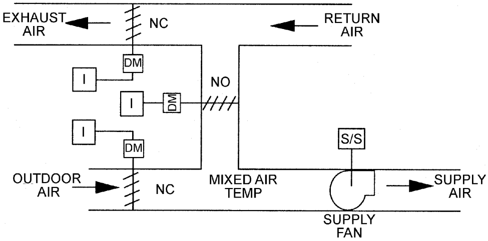


Figure 20.7 Economizer Cycle Control

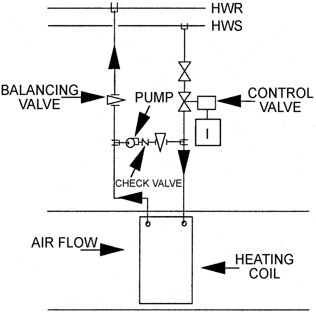


Figure 20.8 Preheat with Secondary Pump and Two-Way Valve

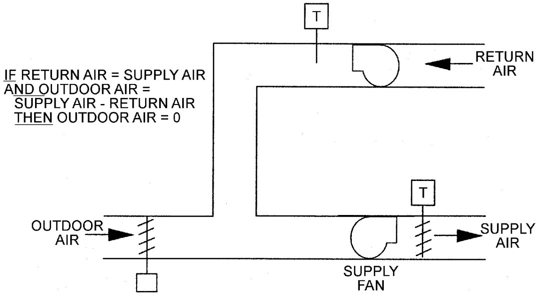


Figure 20.9 Warm-Up Control

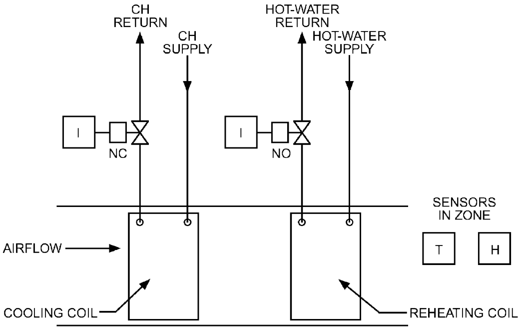


Figure 20.10 Cooling and Dehumidifying with Reheat [2015A, Ch 47, Fig 32]

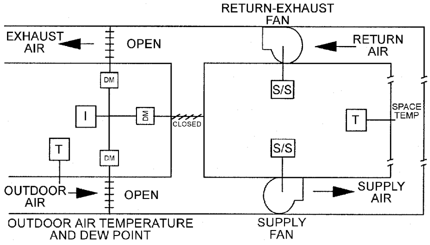


Figure 20.11 Night Cooldown Control

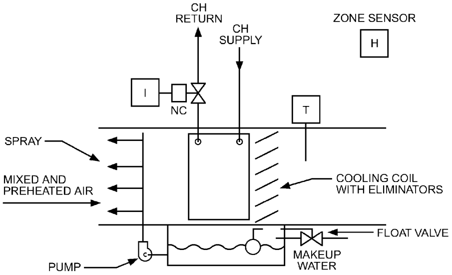


Figure 20.12 Sprayed Coil Dehumidifier [2015A, Ch 47, Fig 33]

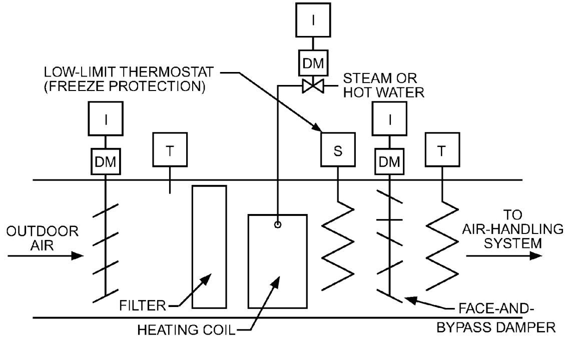


Figure 20.13 Preheat with Face and Bypass Dampers [2015A, Ch 47, Fig 5]

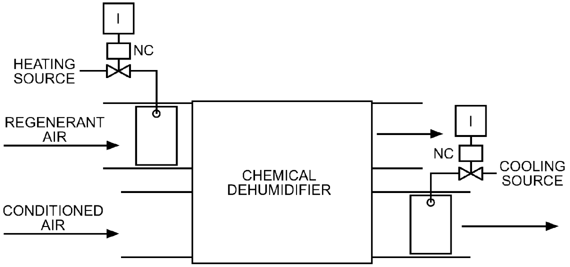


Figure 20.14 Chemical Dehumidifier [2015A, Ch 47, Fig 35]

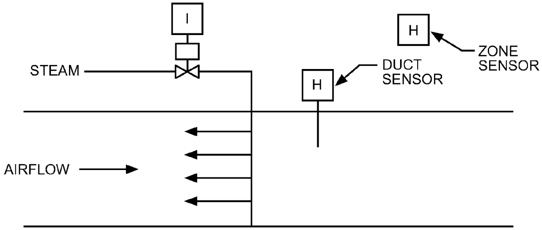


Figure 20.15 Steam Jet Humidifier [2015A, Ch 47, Fig 36]

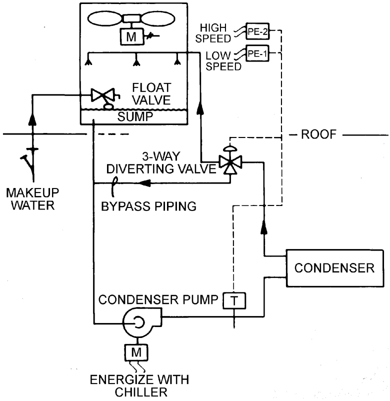


Figure 20.16 Condenser Water Temperature Control

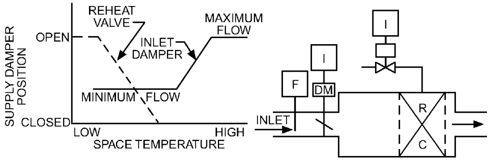


Figure 20.17 Throttling VAV Terminal Unit [2015A, Ch 47, Fig 25]

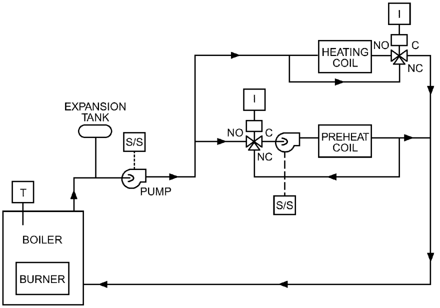


Figure 20.18 Load and Zone Control in Simple Hydronic System [2015A, Ch 47, Fig 3]

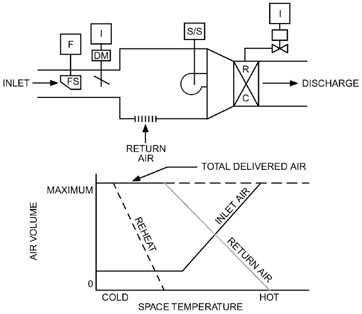


Figure 20.19 Fan-Powered VAV Terminal Unit [2015A, Ch 47, Fig 28]

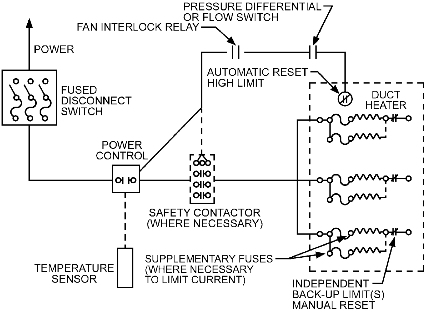


Figure 20.20 Duct Heater Control [2015A, Ch 47, Fig 9]

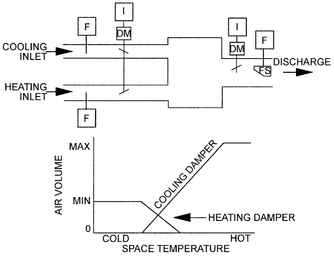


Figure 20.21 Pressure-Independent Dual-Duct VAV Terminal Unit

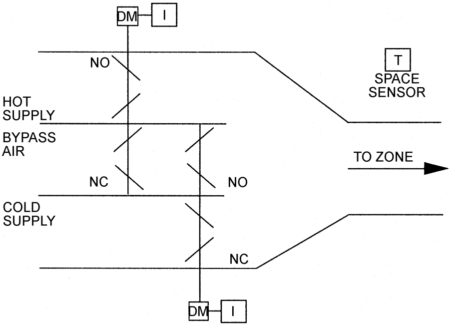


Figure 20.22 Zone Mixing Dampers—Three-Deck Multizone System

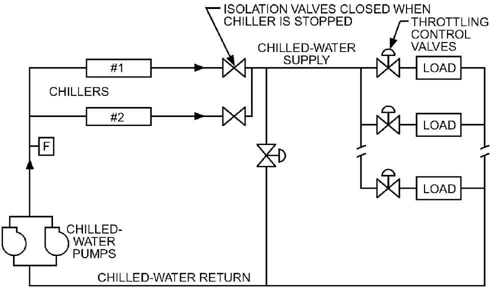


Figure 20.23 Variable-Flow Chilled-Water System (Primary Only) [2015A, Ch 47, Fig 10]

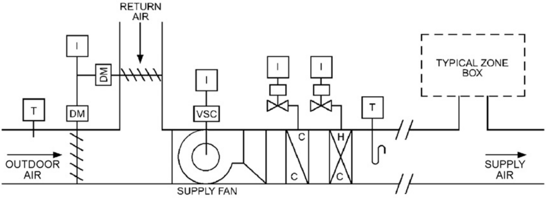


Figure 20.24 Multizone Single-Duct System [2015A, Ch 47, Fig 42]

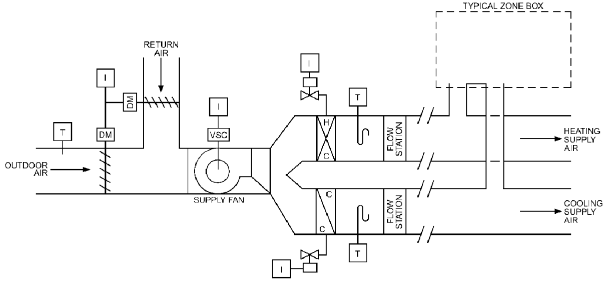


Figure 20.25 Dual-Duct Single Supply Fan System [2015A, Ch 47, Fig 43]

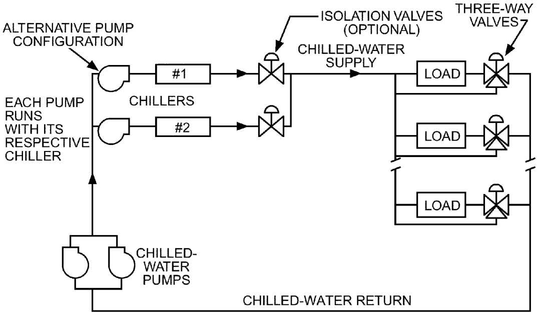
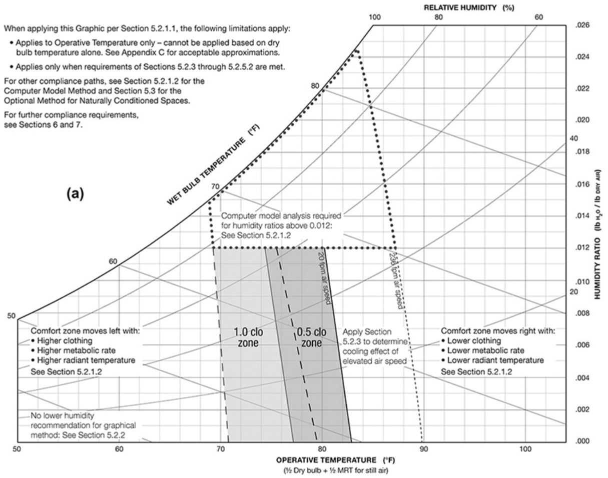


Figure 20.26 Constant-Flow Chilled-Water System (Primary/Only) [2015A, Ch 47, Fig 12]

21. OCCUPANT COMFORT

ASHRAE Standard 55-2013, *Thermal Environmental Conditions for Human Occupancy*

(*See complete standard for detailed guidance.*)



Acceptable ranges of operative temperature and humidity for people in 0.5 to 1.0 clo clothing, activity between 1.0 met and 1.3 met. The operative temperature ranges are based on a 80% satisfaction criterion; 10% general dissatisfaction and 10% partial (local) dissatisfaction.

Figure 21.1 Graphic Comfort Zone Method [Std 55-2013, Fig 5.3.1]

Table 21.1 Acceptable Thermal Environment for General Comfort
[Std 55-2013, Tbl F3]

PPD	PMV Range
< 10	–0.5 < PMV < + 0.5

temperature, operative **(t_o):** the uniform temperature of an imaginary black enclosure in which an occupant would exchange the same amount of heat by radiation plus convection as in the actual nonuniform environment. An acceptable approximation that operative temperature equals air temperature exists when there is no radiant or radiant panel heating or cooling system; there is no major heat generating equipment in the space; the wall/window $U_w < 15.8/(t_{di} - t_{de})$, where t_{di} is the inside design temperature and t_{de} is the outside design temperature; and window solar heat gain coefficient (SHGC) < 0.48. Where air speed is low and t_{air} is closer than 7°F to $t_{mean\ radiant}$, the t_{op} is their mean value.

A computer program is presented in Appendix B of Standard 55-2013 to calculate predicted mean vote (PMV). The PPD (predicted percentage of people dissatisfied) is a function of the PMV.

Table 21.2 Typical Insulation and Permeation Efficiency Values for Western Clothing Ensembles [2017F, Ch 9, Tbl 7]

Ensemble Description[a]	I_{cl}, clo	I_t,[b] clo	f_{cl}	i_{cl}	i_m[b]
Walking shorts, short-sleeved shirt	0.36	1.02	1.10	0.34	0.42
Trousers, short-sleeved shirt	0.57	1.20	1.15	0.36	0.43
Trousers, long-sleeved shirt	0.61	1.21	1.20	0.41	0.45
Same as above, plus suit jacket	0.96	1.54	1.23		
Same as above, plus vest and T-shirt	1.14	1.69	1.32	0.32	0.37
Trousers, long-sleeved shirt, long-sleeved sweater, T-shirt	1.01	1.56	1.28		
Same as above, plus suit jacket and long underwear bottoms	1.30	1.83	1.33		
Sweat pants, sweat shirt	0.74	1.35	1.19	0.41	0.45
Long-sleeved pajama top, long pajama trousers, short 3/4 sleeved robe, slippers (no socks)	0.96	1.50	1.32	0.37	0.41
Knee-length skirt, short-sleeved shirt, panty hose, sandals	0.54	1.10	1.26		
Knee-length skirt, long-sleeved shirt, full slip, panty hose	0.67	1.22	1.29		
Knee-length skirt, long-sleeved shirt, half slip, panty hose, long-sleeved sweater	1.10	1.59	1.46		
Same as above, replace sweater with suit jacket	1.04	1.60	1.30	0.35	0.40
Ankle-length skirt, long-sleeved shirt, suit jacket, panty hose	1.10	1.59	1.46		
Long-sleeved coveralls, T-shirt	0.72	1.30	1.23		
Overalls, long-sleeved shirt, T-shirt	0.89	1.46	1.27	0.35	0.40
Insulated coveralls, long-sleeved thermal underwear, long underwear bottoms	1.37	1.94	1.26	0.35	0.39

Sources: McCullough and Jones (1984) and McCullough et al. (1989).

[a] All ensembles include shoes and briefs or panties. All ensembles except those with panty hose include socks unless otherwise noted.

[b] For $t_r = t_a$ and air velocity less than 40 fpm (I_a = 0.72 clo and i_m = 0.48 when nude).

Table 21.3 Insulation and Permeability Values for a Selection of Non-Western Clothing Ensembles [2017F, Ch 9, Tbl 8]

Ensemble Description[a]	Country	I_{cl} clo	I_t, clo	f_{cl}	i_m
Shalwar (pants), kameez (shirt), scarf, sandals (f)	Pakistan	0.69	1.1	1.41	0.32
Shalwar (pants), kameez (shirt), socks, athletic shoes (m)	Pakistan	0.86	1.3	1.36	0.35
Dishdasha (thowb or caftan), short-sleeved t-shirt, long serwal (pants), tagiya (hat), iqal (cord), ghutra (headdress), socks, athletic shoes (m)	Kuwait	1.36	1.7	1.66	0.30
Full slip, double-layer abaya (dress), anta (head cover), hijab (headscarf), sandals (f)	Kuwait	1.27	1.7	1.65	0.33
Underskirt, blouse, sari, sandals (f)	India	0.74	1.2	1.46	0.33
Churidhar pants, churidhar dress, shawl, sandals (f)	India	0.58	1.1	1.28	0.36
Short shirt with long sleeves, long pants, boubou (wide-sleeved robe), kufi (hat), sandals (m)	Nigeria/ Ghana	1.40	1.7	1.96	0.42
Short shirt with long sleeves, long pants, sandals (f)	Nigeria/ Ghana	0.78	1.3	1.35	0.40
Long-sleeved shirt, skirt, headscarf, socks, athletic shoes (f)	Indonesia	0.97	1.4	1.43	0.31
Camisole, short-sleeved qipao (dress), sandals (f)	China	0.42	0.9	1.31	0.40

(f) = clothing traditionally worn by women
(m) = clothing traditionally worn by men
Source: Havenith et al. (2015). Values are the means of manikin-based measurements conducted in three laboratories. All ensembles include bra and panties (female) and briefs (male). For all women's ensembles, I_a = 0.64 clo; for all men's ensembles, I_a = 0.63 clo.

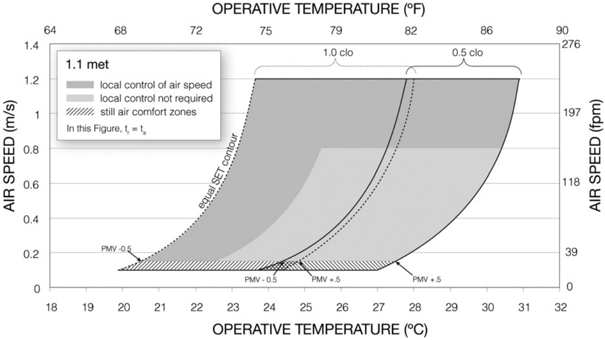


Figure 21.2 Acceptable Range of Operative Temperature and Air Speeds for the Comfort Zone Shown in Figure 21.1, at Humidity Ratio 0.010 [Std 55-2013, Fig 5.3.3A]

Table 21.4 Percentage Dissatisfied Due to Local Discomfort from Draft (DR) or Other Sources (PD) [Std 55-2013, Tbl H1]

DR Due to Draft	PD Due to Vertical Air Temperature Difference	PD Due to Warm or Cool Floors	PD Due to Radiant Asymmetry
< 20%	< 5%	< 10%	< 5%

Table 21.5 Allowable Radiant Temperature Asymmetry [Std 55-2013, Tbl 5.3.4.2]

Radiant Temperature Asymmetry °F			
Warm Ceiling	**Cool Wall**	**Cool Ceiling**	**Warm Wall**
9.0	18.0	25.2	41.4

Table 21.6 Limits on Temperature Drifts and Ramps [Std 55-2013, Tbl 5.3.5.3]

Time Period	0.25 h	0.5 h	1 h	2 h	4 h
Maximum Operative Temperature Change Allowed	2.0°F	3.0°F	4.0°F	5.0°F	6.0°F

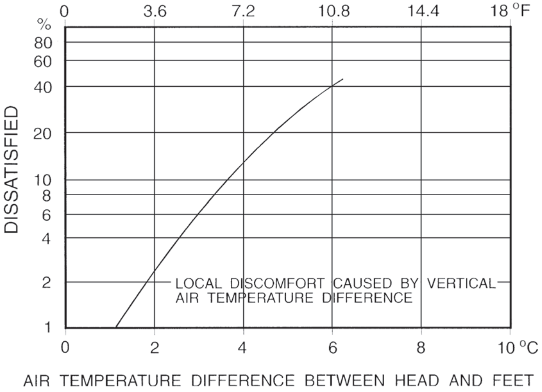


Figure 21.3 Local Thermal Discomfort caused by Vertical Temperature Differences
[Std 55-2013, Fig H4]

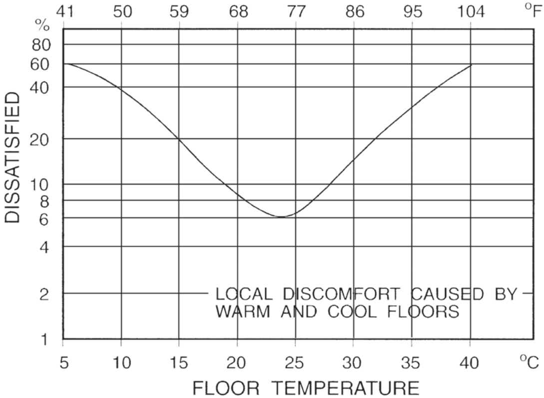


Figure 21.4 Local Discomfort caused by Warm and Cool Floors [Std 55-2013, Fig H5]

Table 21.7 Increases in Acceptable Operative Temperature Limits (Δt_0) in Occupant-Controlled, Naturally Conditioned Spaces (Figure 21.6) Resulting from Increasing Air Speed above 59 fpm [Std 55-2013, Tbl 5.4.2.4]

Average Air Speed (V_a) 118 fpm	Average Air Speed (V_a) 177 fpm	Average Air Speed (V_a) 236 fpm
2.2°F	3.2°F	4.0°F

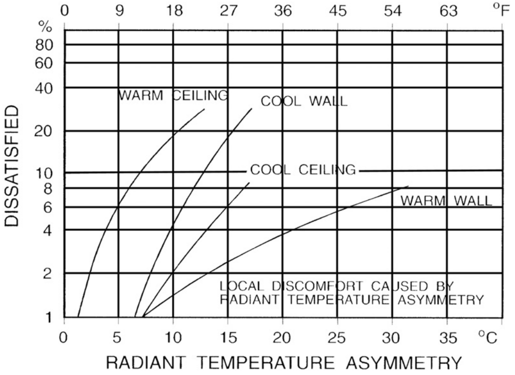


Figure 21.5 Local Thermal Discomfort caused by Radiant Asymmetry [Std 55-2013, Fig H2]

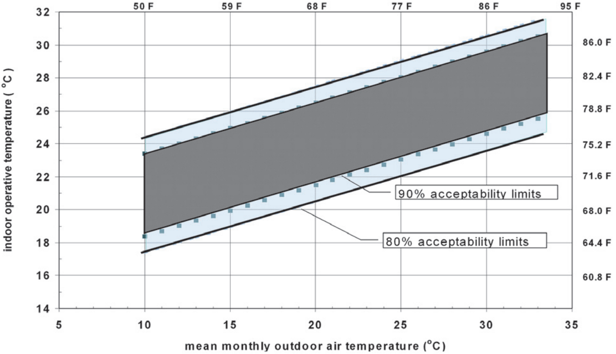


Figure 21.6 Thermal Comfort in Naturally Ventilated Buildings [Std 55-2013, Fig 5.4.2]

Use Figure 21.6 to calculate the average of the mean minimum and maximum air temperatures for a given month, and then use the chart to determine the acceptable range of indoor operative temperatures for a naturally ventilated building. During the design phase of a building, these numbers could be compared to the output of a thermal simulation model of the proposed building to determine whether the predicted indoor temperatures are likely to be comfortable using natural ventilation, or if air conditioning would be required. Figure 21.6 also could be used to evaluate the acceptability of thermal conditions in an existing building by comparing the acceptable temperature range obtained from the chart to indoor temperatures measured in the building.

Figure 21.6 is applicable where occupants control operable windows, where activity levels are between 1.0 and 1.3 met, and where occupants may freely adapt their clothing to the indoor and/or outdoor thermal conditions.

22. GENERAL

Table 22.1 General Design Criteria[a, b] [2007A, Ch 3, Tbl 1]

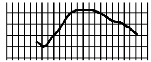
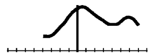
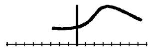
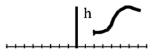
General Category	Specific Category	Inside Design Conditions: Winter	Inside Design Conditions: Summer	Air Movement	Circulation, ach	Noise[c]	Filtering Efficiencies (ASHRAE Std. 52.1)	Load Profile	Comments
Dining and Entertainment Centers	Cafeterias and Luncheonettes	70 to 74°F 20 to 30% rh	78°F[d] 50% rh	50 fpm at 6 ft above floor	12 to 15	NC 40 to 50[e]	35% or better	Peak at 1 to 2 PM 	Prevent draft discomfort for patrons waiting in serving lines
	Restaurants	70 to 74°F 20 to 30% rh	74 to 78°F 55 to 60% rh	25 to 30 fpm	8 to 12	NC 35 to 40	35% or better	Peak at 1 to 2 PM 	
	Bars	70 to 74°F 20 to 30% rh	74 to 78°F 50 to 60% rh	30 fpm at 6 ft above floor	15 to 20	NC 35 to 50	Use charcoal for odor control with manual purge control for 100% outside air to exhaust ±35% prefilters	Peak at 5 to 7 PM 	
	Nightclubs and Casinos	70 to 74°F 20 to 30% rh	74 to 78°F 50 to 60% rh	below 25 fpm at 5 ft above floor	20 to 30	NC 35 to 45[f]	Use charcoal for odor control with manual purge control for 100% outside air to exhaust ±35% prefilters	Nightclubs peak at 8 PM to 2 AM; Casinos peak at 4 PM to 2 AM; Equipment, 24 h/day	Provide good air movement but prevent cold draft discomfort for patrons
	Kitchens	70 to 74°F	85 to 88°F	30 to 50 fpm	12 to 15[g]	NC 40 to 50	10 to 15% or better		Negative air pressure required for odor control (also see 2015A, Ch 33)

Table 22.1 General Design Criteria[a, b] [2007A, Ch 3, Tbl 1] (*Continued*)

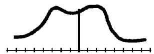
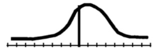
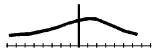
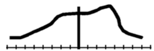
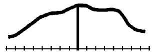
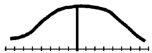
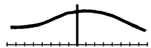
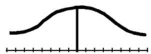
General Category	Specific Category	Inside Design Conditions: Winter	Inside Design Conditions: Summer	Air Movement	Circulation, ach	Noise[c]	Filtering Efficiencies (ASHRAE Std. 52.1)	Load Profile	Comments
Office Buildings		70 to 74°F 20 to 30% rh	74 to 78°F 50 to 60% rh	25 to 45 fpm 0.75 to 2 cfm/ft^2	4 to 10	NC 30 to 40	35 to 60% or better	Peak at 4 PM 	
Museums, Galleries, Libraries, and Archives (also see Chapter 21)	Average	68 to 72°F 40 to 55% rh		below 25 fpm	8 to 12	NC 35 to 40	35 to 60% or better	Peak at 3 PM 	
	Archival	See 2015A, Ch 23		below 25 fpm	8 to 12	NC 35	35% prefilters plus charcoal filters 85 to 95% final[i]	Peak at 3 PM 	
Bowling Centers		70 to 74°F 20 to 30% rh	75 to 78°F 50 to 55% rh	50 fpm at 6 ft above floor	10 to 15	NC 40 to 50	10 to 15%	Peak at 6 to 8 PM 	
Communication Centers	Telephone Terminal Rooms	72 to 78°F 40 to 50% rh	72 to 78°F 40 to 50% rh	25 to 30 fpm	8 to 20	to NC 60	85% or better	Varies with location and use	Constant temperature and humidity required
	Radio and Television Studios	74 to 78°F 30 to 40% rh	74 to 78°F 40 to 55% rh	below 25 fpm at 12 ft above floor	15 to 40	NC 15 to 25	35% or better	Varies widely because of changes in lighting and people	Constant temperature and humidity required

Table 22.1 General Design Criteria[a, b] [2007A, Ch 3, Tbl 1] (*Continued*)

General Category	Specific Category	Inside Design Conditions		Air Movement	Circulation, ach	Noise[c]	Filtering Efficiencies (ASHRAE Std. 52.1)	Load Profile	Comments
		Winter	Summer						
Transportation Centers (also see 2015A, Ch 15)	Airport Terminals	70 to 74°F 20 to 30% rh	74 to 78°F 50 to 60% rh	25 to 30 fpm at 6 ft above floor	8 to 12	NC 35 to 50	35% or better and charcoal filters	Peak at 10 AM to 9 PM 	Positive air pressure required in terminal
	Ship Docks	70 to 74°F 20 to 30% rh	74 to 78°F 50 to 60% rh	25 to 30 fpm at 6 ft above floor	8 to 12	NC 35 to 50	10 to 15%	Peak at 10 AM to 5 PM 	Positive air pressure required in waiting area
	Bus Terminals	70 to 74°F 20 to 30% rh	74 to 78°F 50 to 60% rh	25 to 30 fpm at 6 ft above floor	8 to 12	NC 35 to 50	35% with exfiltration	Peak at 10 AM to 5 PM 	Positive air pressure required in terminal
	Garages[j]	40 to 55°F	80 to 100°F	30 to 75 fpm	4 to 6	NC 35 to 50	10 to 15%	Peak at 10 AM to 5 PM 	Negative air pressure required to remove fumes; positive air in pressure adjacent occupied spaces
Warehouses		Inside design temperatures for warehouses often depend on the materials stored			1 to 4	to NC 75	10 to 35%	Peak at 10 AM to 3 PM 	

Notes to General Design Criteria

[a]This table shows design criteria differences between various commercial and public buildings. It should not be used as sole source for design criteria. Each type of data contained here can be determined from *ASHRAE Handbook* and standards.

[b]Consult governing codes to determine minimum allowable requirements. Outside air requirements may be reduced if high-efficiency adsorption equipment or other odor- or gas-removal equipment is used. See ASHRAE Standard 62.1 for calculation procedures.

[c]Refer to Chapter 48 of the 2011 *ASHRAE Handbook—HVAC Applications*.

[d]Food in these areas is often eaten more quickly than in a restaurant; therefore, turnover of diners is much faster. Because diners seldom remain for long periods, they do not require the degree of comfort necessary in restaurants. Thus, it may be possible to lower design criteria standards and still provide reasonably comfortable conditions. Although space conditions of 80°F and 50% rh may be satisfactory for patrons when it is 95°F and 50% rh outside, inside conditions of 78°F and 40% rh are better.

[e]Cafeterias and luncheonettes usually have some or all food preparation equipment and trays in the same room with diners. These establishments are generally noisier than restaurants, so noise transmission from air-conditioning equipment is not as critical.

[f]In some nightclubs, air-conditioning system noise must be kept low so patrons can hear the entertainment.

[g]Usually determined by kitchen hood requirements.

[h]Peak kitchen heat load does not generally occur at peak dining load, although in luncheonettes and some cafeterias where cooking is done in dining areas, peaks may be simultaneous.

[i]Methods for removing chemical pollutants must also be considered.

[j]Also includes service stations.

Air-Conditioning Formulas

1 Btu = amount of heat required to raise (or lower) temperature of one pound of water 1°F

1 ton refrigeration = 12,000 Btu/h = 200 Btu/min

1 watt = 3.412 Btu/h

1 horsepower = 2545 Btu/h

1 lb = 7000 grains

1 ft (head) = 0.433 psi

1 square foot EDR (equivalent direct radiation) = 240 Btu

1 boiler horsepower = 33,479 Btu/h

No. of air changes (N) = 60 (cfm)/ft^3

Sensible heat (Btu/h) = 1.08 $Q\Delta t$

where Δt = difference between entering and leaving dry-bulb temperature and Q = airflow rate in cubic feet per minute

Latent heat (Btu/h) = 0.68 $Q\Delta g$

where Δg = difference in moisture content of entering and leaving air, grains per pound of dry air

Water quantity (gpm) required for heating and cooling =

$q/500\ \Delta t_{water}$

where q = load in Btu/h

Chiller capacity (tons) = gpm (chilled water) × Δt (water)/24

For Air:

1 lb/h = 4.5 Q

1 ton = $Q\Delta h/2670$

$$\text{Fan hp} = \frac{\text{cfm} \times \text{static pressure (in. w.g.)}}{6356 \times \text{Efficiency}} \times \frac{\text{Density of air}}{\text{Density of standard air}}$$

For water:

1 lb/h = 500 gpm

1 ton = (gpm) $\Delta t/24$

$$\text{Pump hp} = \frac{\text{gpm} \times \text{ft head}}{3960 \times \text{Efficiency}} \times \text{Specific Gravity}$$

small pumps 0.40 – 0.60 efficiency

large pumps 0.70 – 0.85 efficiency

Control Valves (C_v):

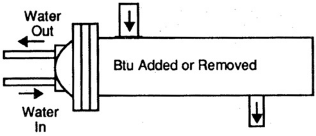
$$\text{Liquid: } = \frac{gpm\sqrt{sp\ gr}}{\sqrt{\Delta p_{psi}}}$$

$$\text{Steam: } = \frac{(lbsteam/hr)\sqrt{spec\ vol}}{63.5\sqrt{\Delta p_{psi}}}$$

(at 5 psi; specific volume = 20.4, at 30 psi; specific volume = 9.46)

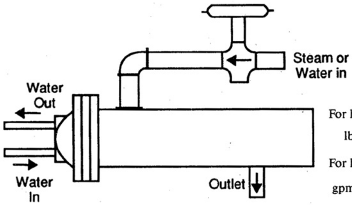
General

Sizing Formulas



For heating or cooling water:

$$\text{gpm} = \frac{\text{Btuh}}{500 \times (\text{water temp. rise or drop, °F})}$$

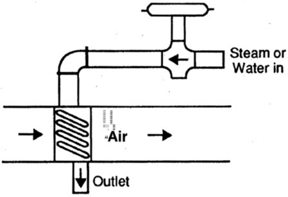


For heating water with steam:

$$\text{lb steam/h} = 0.5\ \text{gpm (water temp. rise, °F)}$$

For heating or cooling water with water:

$$\text{gpm}_1 = \text{gpm}_2 \left(\frac{\text{water}_2 \text{ temp. rise or drop, °F}}{\text{water}_1 \text{ temp. rise or drop, °F}} \right)$$



For heating air with steam coils:

$$\text{lb steam/h} = \left(\frac{\text{CFM}}{1000} \right) (\text{air temp. rise, °F})$$

For heating air with water coils:

$$\text{gpm} = 2.16 \left(\frac{\text{CFM}}{1000} \right) \left(\frac{\text{air temp. rise, °F}}{\text{water temp. drop, °F}} \right)$$

For radiation:

$$\text{lb steam/h} = 0.25\ (\text{sq ft EDR})$$

$$(1\ \text{sq ft EDR} = 240\ \text{Btu/h})$$

Figure 22.1 Sizing Formulas for Heating/Cooling

Table 22.2 Conversions to I-P and SI Units [2017F, Ch 38, Tbl 1]
(Multiply I-P values by conversion factors to obtain SI; divide SI values by conversion factors to obtain I-P)

Multiply I-P	By	To Obtain SI
acre (43,560 ft^2)	0.4047	ha
	4046.873	m^2
atmosphere (standard)	*101.325	kPa
bar	*100	kPa
barrel (42 U.S. gal, petroleum)	159.0	L
	0.1580987	m^3
Btu (International Table)	1055.056	J
Btu (thermochemical)	1054.350	J
Btu/ft^2 (International Table)	11,356.53	J/m^2
Btu/ft^3 (International Table)	37,258.951	J/m^3
Btu/gal	278,717.1765	J/m^3
Btu·ft/h·ft^2·°F	1.730735	W/(m·K)
Btu·in/h·ft^2·°F (thermal conductivity *k*).	0.1442279	W/(m·K)
Btu/h	0.2930711	W
Btu/h·ft^2	3.154591	W/m^2
Btu/h·ft^2·°F (overall heat transfer coefficient *U*)	5.678263	W/(m^2·K)
Btu/lb	*2.326	kJ/kg
Btu/lb·°F (specific heat c_p)	*4.1868	kJ/(kg·K)
bushel (dry, U.S.)	0.0352394	m^3
calorie (thermochemical)	*4.184	J
centipoise (dynamic viscosity μ)	*1.00	mPa·s
centistokes (kinematic viscosity ν)	*1.00	mm^2/s
clo	0.155	(m^2·K)/W
dyne	1.0×10^{-5}	N
dyne/cm^2	*0.100	Pa
EDR hot water (150 Btu/h)	43.9606	W
EDR steam (240 Btu/h)	70.33706	W
EER	0.293	COP
ft	*0.3048	m
	*304.8	mm
ft/min, fpm	*0.00508	m/s
ft/s, fps	*0.3048	m/s
ft of water	2989	Pa
ft of water per 100 ft pipe	98.1	Pa/m
ft^2	0.092903	m^2
ft^2·h·°F/Btu (thermal resistance *R*)	0.176110	(m^2·K)/W
ft^2/s (kinematic viscosity ν)	92,900	mm^2/s
ft^3	28.316846	L
	0.02832	m^3
ft^3/min, cfm	0.471947	L/s
ft^3/s, cfs	28.316845	L/s
ft·lb_f (torque or moment)	1.355818	N·m
ft·lb_f (work)	1.356	J
ft·lb_f/lb (specific energy)	2.99	J/kg
ft·lb_f/min (power)	0.0226	W
footcandle	10.76391	lx
gallon (U.S., *231 in^3)	3.785412	L
gph	1.05	mL/s
gpm	0.0631	L/s
gpm/ft^2	0.6791	L/(s·m^2)
gpm/ton refrigeration	0.0179	mL/J
grain (1/7000 lb)	0.0648	g
gr/gal	17.1	g/m^3
gr/lb	0.143	g/kg
horsepower (boiler) (33,470 Btu/h)	9.81	kW
horsepower (550 ft·lb_f/s)	0.7457	kW
inch	*25.4	mm
in. of mercury (60°F)	3.3864	kPa
in. of water (60°F)	248.84	Pa
in/100 ft, thermal expansion coefficient	0.833	mm/m

Table 22.2 Conversions to I-P and SI Units [2017F, Ch 38, Tbl 1] **(*Continued*)**
(Multiply I-P values by conversion factors to obtain SI; divide SI values by conversion factors to obtain I-P)

Multiply I-P	By	To Obtain SI
$in \cdot lb_f$ (torque or moment)	113	mN·m
in^2	645.16	mm^2
in^3 (volume)	16.3874	mL
in^3/min (SCIM)	0.273117	mL/s
in^3 (section modulus)	16,387	mm^3
in^4 (section moment)	416,231	mm^4
kWh	*3.60	MJ
kW/1000 cfm	2.118880	kJ/m^3
kilopond (kg force)	9.81	N
kip (1000 lb_f)	4.45	kN
kip/in^2 (ksi)	6.895	MPa
litre	*0.001	m^3
met	58.15	W/m^2
micron (μm) of mercury (60°F)	133	mPa
mile	1.609	km
mile, nautical	*1.852	km
mile per hour (mph)	1.609344	km/h
	0.447	m/s
millibar	*0.100	kPa
mm of mercury (60°F)	0.133	kPa
mm of water (60°F)	9.80	Pa
ounce (mass, avoirdupois)	28.35	g
ounce (force or thrust)	0.278	N
ounce (liquid, U.S.)	29.6	mL
ounce inch (torque, moment)	7.06	mN·m
ounce (avoirdupois) per gallon	7.489152	kg/m^3
perm (permeance at 32°F)	5.72135×10^{-11}	$kg/(Pa \cdot s \cdot m^2)$
perm inch (permeability at 32°F)	1.45362×10^{-12}	$kg/(Pa \cdot s \cdot m)$
pint (liquid, U.S.)	4.73176×10^{-4}	m^3
pound		
lb (avoirdupois, mass)	0.453592	kg
	453.592	g
lb_f (force or thrust)	4.448222	N
lb_f/ft (uniform load)	14.59390	N/m
lb/ft·h (dynamic viscosity μ)	0.4134	mPa·s
lb/ft·s (dynamic viscosity μ)	1490	mPa·s
$lb_f \cdot s/ft^2$ (dynamic viscosity μ)	47.88026	Pa·s
lb/h	0.000126	kg/s
lb/min	0.007559	kg/s
lb/h [steam at 212°F (100°C)]	0.2843	kW
lb_f/ft^2	47.9	Pa
lb/ft^2	4.88	kg/m^2
lb/ft^3 (density ρ)	16.0	kg/m^3
lb/gallon	120	kg/m^3
ppm (by mass)	*1.00	mg/kg
psi	6.895	kPa
quad (10^{15} Btu)	1.055	EJ
quart (liquid, U.S.)	0.9463	L
square (100 ft^2)	9.2903	m^2
tablespoon (approximately)	15	mL
teaspoon (approximately)	5	mL
therm (U.S.)	105.5	MJ
ton, long (2240 lb)	1.016046	Mg
ton, short (2000 lb)	0.907184	Mg; t (tonne)
ton, refrigeration (12,000 Btu/h)	3.517	kW
torr (1 mm Hg at 0°C)	133	Pa
watt per square foot	10.76	W/m^2
yd	*0.9144	m
yd^2	0.8361	m^2
yd^3	0.7646	m^3

*Conversion factor is exact.

Notes: 1. Units are U.S. values unless noted otherwise.

2. Litre is a special name for the cubic decimetre. 1 L = 1 dm^3 and 1 mL = 1 cm^3.

DB: Dry-bulb temperature, °F *WB: Wet-bulb temperature, °F* *DP: Dew-point temperature, °F*
MCWB: Mean coincident wet-bulb temperature, °F *HDD and CDD 65: Annual heating and cooling degree-days, base 65°F, °F-day*

Appendix: Climatic Design Conditions for Selected Locations

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
United States of America										
Alabama										
AUBURN UNIVERSITY REGIONAL	23.2	27.5	93.4	73.5	91.2	73.9	90.1	73.6	2416	1916
BIRMINGHAM SHUTTLESWORTH INTL	20.8	25.1	95.6	74.6	93.2	74.5	91.1	74.2	2610	2054
CAIRNS AAF	26.8	29.8	95.6	76.2	93.4	76.0	91.4	75.6	1797	2378
DOTHAN REGIONAL	27.4	30.9	96.4	75.7	93.9	75.3	91.9	75.0	1735	2521
HUNTSVILLE INTL	18.7	22.9	95.3	75.0	92.9	74.6	90.7	74.0	3052	1846
MAXWELL AFB	25.6	29.6	97.5	76.1	95.5	76.3	93.5	76.3	1957	2546
MOBILE REGIONAL	27.7	31.1	94.1	76.8	92.2	76.5	90.5	76.1	1642	2516
MONTGOMERY REGIONAL	24.3	27.9	96.8	76.0	94.7	75.9	92.7	75.6	2109	2352
NORTHEAST ALABAMA AP	18.8	22.6	93.4	74.4	91.2	74.4	90.0	74.2	3179	1590
NORTHWEST ALABAMA REGIONAL	19.8	23.8	96.2	75.4	93.7	75.1	91.4	74.7	2997	1906
TUSCALOOSA REGIONAL	22.5	26.6	97.3	75.4	94.6	75.7	92.4	75.4	2450	2200
Alaska										
BRYANT AAF	–19.6	–14.0	74.8	60.3	71.6	58.9	68.3	57.1	10677	5
ELMENDORF AFB	–15.4	–10.6	74.2	58.7	71.6	57.8	68.2	56.4	10324	12
FAIRBANKS INTL	–42.8	–38.6	81.2	60.9	78.1	59.8	74.8	58.5	13577	69
JUNEAU INTL	4.7	8.9	73.9	59.5	70.1	58.0	66.5	56.5	8377	3
LAKE HOOD SEAPLANE BASE	–8.2	–3.7	74.0	59.5	70.7	58.1	67.9	56.7	9741	15
MERRILL FIELD	–10.7	–7.0	73.2	59.3	70.6	58.2	68.1	56.9	9961	13
TED STEVENS ANCHORAGE INTL	–8.7	–4.5	71.5	58.9	68.4	57.4	66.0	56.2	10115	5
Arizona										
CASA GRANDE MUNICIPAL	31.7	34.9	108.5	69.5	106.5	69.1	104.5	68.6	1498	3575
DAVIS-MONTHAN AFB	32.4	35.5	105.2	65.2	102.8	64.9	100.4	64.6	1457	3222

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
FLAGSTAFF PULLIAM AP	3.9	9.4	85.8	55.3	83.5	54.9	81.3	54.7	6801	129
LUKE AFB	34.8	37.3	110.9	70.0	108.6	69.9	106.4	69.6	1222	4041
PHOENIX SKY HARBOR INTL	39.2	41.8	110.3	69.5	108.3	69.3	106.4	69.0	912	4636
PRESCOTT MUNICIPAL	17.7	20.8	94.5	60.6	92.1	60.1	90.1	59.7	4145	1021
TUCSON INTL	31.8	34.4	105.8	66.0	103.6	65.7	101.4	65.4	1401	3310
WINDOW ROCK AP	0.1	5.2	89.8	56.2	87.7	55.7	85.1	55.0	6395	306
YUMA INTL AP	42.3	44.8	111.0	72.6	108.9	72.3	107.5	71.7	657	4749
Arkansas										
BENTONVILLE MUNICIPAL	10.1	15.9	96.6	74.5	92.6	74.4	90.2	74.1	4043	1437
CLINTON NATL	20.6	24.5	98.8	77.0	96.0	77.0	93.4	76.6	2897	2207
DRAKE FIELD	10.4	16.3	95.9	74.3	92.8	74.4	90.4	74.3	3962	1426
FORT SMITH REGIONAL	17.9	22.4	100.3	75.8	97.3	76.2	94.5	75.9	3122	2131
GRIDER FIELD	22.4	25.8	97.5	77.5	95.2	77.3	93.0	76.9	2745	2223
JONESBORO MUNICIPAL	17.1	20.7	97.1	76.9	94.7	76.2	92.6	75.8	3504	1958
LITTLE ROCK AFB	18.1	22.2	99.8	77.3	97.0	77.5	94.3	77.2	3135	2098
NORTH LITTLE ROCK MUNICIPAL	18.4	23.2	95.4	76.6	93.0	76.3	90.9	75.6	3175	1937
ROGERS MUNICIPAL	9.9	15.7	95.1	73.3	91.9	73.7	89.8	73.2	4046	1429
SMITH FIELD	10.2	16.1	97.2	74.3	93.1	74.3	90.5	74.1	3954	1489
TEXARKANA REGIONAL	23.8	27.2	99.3	75.7	96.8	75.9	94.2	75.8	2448	2369
California										
ALAMEDA	40.4	42.4	81.1	64.0	77.2	62.9	73.9	62.0	2508	160
BEALE AFB	31.1	33.9	101.1	70.3	98.2	69.1	95.2	67.9	2398	1548
BOB HOPE AP	38.8	41.1	97.6	67.9	94.1	66.8	91.1	66.4	1381	1449
BROWN FIELD MUNICIPAL	39.0	41.5	89.5	64.1	85.2	64.6	82.0	64.6	1647	667
CAMARILLO AP	37.5	39.7	86.2	62.2	82.4	63.1	79.9	63.1	1819	421
CAMP PENDLETON MCAS	31.9	34.7	92.0	65.8	87.9	65.3	84.3	65.2	1818	679

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
CASTLE AFB	30.0	31.9	102.5	69.6	99.9	68.2	97.7	67.3	2384	1762
DESERT RESORTS REGIONAL	31.1	34.4	111.5	72.2	109.1	71.7	107.1	71.2	1092	3915
EL TORO MCAS	43.2	45.3	91.9	67.8	88.9	67.5	86.0	66.8	1111	1172
FRESNO YOSEMITE INTL	31.9	34.1	103.7	70.3	101.0	69.0	98.5	68.0	2216	2166
FULLERTON MUNICIPAL	39.8	42.8	95.0	66.9	91.3	66.8	88.3	66.3	1113	1352
HAWTHORN MUNICIPAL	44.5	45.8	87.9	62.8	84.0	63.0	81.1	63.5	1105	819
HAYWARD EXECUTIVE	36.8	38.9	87.1	65.2	82.4	64.1	78.5	63.3	2542	293
IMPERIAL COUNTY AP	35.7	37.9	111.3	72.8	109.1	72.4	107.5	72.1	937	4164
LEMOORE NAS	28.0	31.0	103.4	70.7	100.6	69.7	98.4	68.7	2296	1836
LIVERMORE MUNICIPAL	30.6	33.1	98.7	67.6	94.8	66.0	90.9	64.8	2671	836
LOMPOC AP	32.4	35.5	81.7	61.1	77.4	60.9	74.1	60.4	2809	60
LONG BEACH AP	41.6	43.7	91.4	66.2	87.8	66.3	84.5	65.6	1188	1098
LOS ANGELES INTL	44.9	46.8	83.8	63.1	80.4	63.9	77.8	64.1	1287	614
LOS ANGELES ONTARIO INTL	38.4	40.6	100.2	69.7	97.5	68.4	94.6	67.8	1381	1797
MARCH AFB	32.1	35.0	101.3	67.9	98.8	66.9	95.9	66.2	1895	1584
MCCLELLAN-PALOMAR AP	42.9	44.8	83.8	62.0	80.7	63.8	77.9	64.0	1623	536
MEADOWS FIELD	32.6	35.3	102.9	70.4	100.4	69.2	98.1	68.2	2050	2319
MINETA SAN JOSE INTL	35.5	37.7	91.4	66.1	87.8	65.3	83.9	64.3	2143	624
MIRAMAR MCAS	39.1	41.4	90.8	65.9	87.5	65.9	84.1	65.5	1489	845
MODESTO CITY-COUNTY AP	30.7	33.3	101.4	69.9	98.2	68.4	95.4	67.1	2362	1631
MOFFETT FEDERAL AIRFIELD	36.0	38.5	88.1	65.5	83.7	64.5	80.5	64.1	2190	467
MONTEREY REGIONAL	36.6	38.8	78.7	59.9	73.5	59.2	71.1	58.9	3231	50
MONTGOMERY FIELD	40.7	43.0	90.3	65.4	86.5	64.9	83.3	64.9	1465	854
NAPA COUNTY AP	29.2	31.8	91.1	65.8	86.5	64.8	82.4	63.9	3182	242
NORTH ISLAND NAS	44.8	46.2	84.6	64.0	81.3	65.2	78.9	65.8	1118	783
OAKLAND INTL	36.6	38.9	83.4	64.2	79.1	63.0	75.0	62.2	2692	164

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
PALM SPRINGS INTL	41.1	43.5	111.4	70.7	109.2	70.4	107.6	69.9	757	4394
POINT ARGUELLO	45.7	47.6	71.6	N/A	67.9	N/A	65.5	N/A	3405	23
POINT MUGU NAS	38.9	41.0	81.5	61.2	78.4	62.4	75.4	62.9	2078	213
PORTERVILLE MUNICIPAL	30.1	33.5	100.5	70.0	99.2	69.3	97.1	68.1	2520	1704
REDDING MUNICIPAL	28.5	30.9	105.3	68.4	102.0	67.1	99.1	66.0	2702	1889
RIVERSIDE MUNICIPAL	36.4	38.7	100.2	69.3	97.7	68.4	94.8	67.4	1448	1718
SACRAMENTO EXECUTIVE	30.9	33.6	100.1	69.8	96.9	68.4	93.6	67.2	2491	1222
SACRAMENTO INTL	30.3	33.2	100.5	70.4	97.6	69.3	94.6	68.0	2496	1364
SACRAMENTO MATHER AP	27.7	30.1	101.2	68.8	98.2	67.4	94.8	66.5	2771	1190
SACRAMENTO MCCLELLAN AFB	30.1	33.0	102.1	70.0	99.3	68.7	95.8	67.4	2323	1588
SALINAS MUNICIPAL	33.9	36.4	83.2	62.1	78.6	61.0	75.0	60.7	2707	112
SAN BERNARDINO INTL	33.9	36.5	102.9	69.7	100.2	69.5	97.4	68.8	1652	1811
SAN DIEGO INTL	45.0	46.9	83.7	64.7	80.6	65.5	78.3	65.6	1171	715
SAN FRANCISCO INTL	39.5	41.6	82.6	62.7	77.9	62.0	74.3	61.4	2679	157
SAN LUIS OBISPO CO REGIONAL	34.0	36.3	89.1	64.0	84.3	63.2	81.2	62.8	2223	294
SANTA BARBARA MUNICIPAL	35.0	37.0	82.9	63.3	79.7	63.7	77.0	63.1	2224	218
SANTA MARIA PUBLIC AP	33.2	35.5	84.2	61.9	80.0	61.3	76.6	60.8	2668	115
SONOMA COUNTY AP	28.9	31.2	94.7	66.4	90.9	65.7	87.2	64.4	2986	366
SOUTHERN CALIFORNIA LOGISTICS	27.6	30.5	100.7	65.3	98.4	64.7	96.0	63.9	2661	1911
STOCKTON METROPOLITAN	30.3	32.7	101.1	70.0	97.9	68.8	94.8	67.9	2448	1393
TRAVIS AFB	30.0	33.0	99.0	67.7	95.1	66.6	91.1	65.6	2516	993
VISALIA MUNICIPAL	29.8	32.4	100.0	71.8	98.7	70.9	96.5	69.9	2509	1641
WILLIAM J FOX AP	21.3	24.7	102.8	65.8	100.3	64.4	98.0	63.5	2937	1890
Colorado										
BUCKLEY AFB	0.4	7.2	93.2	58.4	90.6	58.4	88.0	58.3	5798	687
CENTENNIAL AP	–0.1	5.8	91.9	59.5	89.9	59.2	87.2	58.8	6052	628

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
COLORADO SPRINGS MUNICIPAL	1.4	7.0	90.9	58.7	88.3	58.4	85.7	58.3	6117	505
DENVER INTL	–0.8	5.3	94.8	59.7	92.2	59.7	89.5	59.5	5935	812
DENVER STAPLETON	–1.4	5.1	93.9	60.7	91.2	60.0	88.5	59.6	5667	721
FORT COLLINS DOWNTOWN	–2.6	4.8	90.1	60.9	87.2	60.4	84.4	60.1	6096	462
FORT COLLINS LOVELAND MUNI	–0.2	5.5	93.5	60.9	90.9	60.8	88.3	60.5	6196	624
GRAND JUNCTION REGIONAL	3.7	9.3	97.7	61.2	95.2	60.3	92.7	59.7	5481	1228
GREELEY-WELD COUNTY AP	–7.8	0.0	96.6	62.6	92.6	62.2	90.0	62.1	6579	643
PUEBLO MEMORIAL	0.0	6.5	98.5	62.2	95.8	61.9	93.1	61.6	5488	951
Connecticut										
BRADLEY INTL	4.8	9.8	91.4	73.2	88.4	71.8	85.6	70.3	5843	789
HARTFORD-BRAINARD AP	8.2	12.2	90.9	73.1	88.2	72.2	85.1	70.7	5500	863
IGOR SIKORSKY MEMORIAL	11.4	15.6	87.9	73.1	84.9	71.7	82.5	70.6	5230	858
WATERBURY-OXFORD AP	3.2	8.8	87.6	72.9	83.7	71.1	81.3	69.5	6416	468
WINDHAM AP	3.6	9.3	89.7	73.2	86.5	71.9	83.8	70.4	5950	642
Delaware										
DOVER AFB	14.7	18.7	92.5	75.7	90.0	74.8	87.3	74.0	4461	1194
NEW CASTLE AP	13.8	17.7	92.3	75.1	89.4	73.8	86.8	72.9	4684	1160
Florida										
CECIL FIELD	30.3	34.0	96.1	76.7	93.9	76.4	92.0	75.9	1164	2730
CRAIG MUNICIPAL	32.7	36.1	94.3	77.0	92.2	76.8	90.4	76.6	1158	2745
DAYTONA BEACH INTL	35.5	39.6	92.7	76.9	90.9	76.8	89.3	76.7	728	3020
FT LAUDERDALE HOLLYWOOD INTL	47.8	52.0	91.8	78.3	90.6	78.3	89.8	78.2	124	4609
GAINESVILLE REGIONAL	29.6	33.2	93.8	75.9	92.2	75.8	90.6	75.5	1148	2670
HOMESTEAD AFB	46.0	50.2	91.2	79.2	90.4	79.0	89.6	78.9	144	4228
JACKSONVILLE INTL	29.5	32.8	94.5	77.2	92.7	76.9	90.9	76.5	1303	2629
JACKSONVILLE NAS	34.2	37.5	95.8	76.8	93.7	76.3	91.9	76.0	946	3259

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
KENNEDY SPACE CENTER	39.2	43.1	91.7	78.1	90.4	78.0	89.3	77.8	525	3176
MACDILL AFB	38.8	43.0	93.3	78.4	92.2	78.1	91.0	77.8	491	3653
MAYPORT NAF	35.2	39.0	93.9	77.2	91.6	77.1	89.9	76.8	1007	2961
MELBOURNE INTL	38.6	43.0	91.9	77.5	90.4	77.5	89.5	77.5	462	3473
MIAMI EXECUTIVE	45.6	49.6	92.8	77.9	91.4	77.7	90.5	77.6	164	4147
MIAMI INTL	48.8	52.6	91.9	77.6	90.8	77.6	89.8	77.5	113	4578
NAPLES MUNICIPAL	43.6	47.5	91.6	77.7	90.5	77.8	89.7	77.8	273	3807
ORLANDO EXECUTIVE	38.9	43.2	93.5	76.1	92.4	75.9	91.0	75.7	505	3543
ORLANDO INTL	38.3	42.3	93.8	76.5	92.4	76.2	91.1	75.9	529	3405
ORLANDO SANFORD INTL	36.9	40.9	94.4	75.7	92.9	75.5	91.2	75.4	617	3342
PAGE FIELD	42.7	46.6	93.5	76.7	92.4	76.7	91.2	76.7	267	3943
PALM BEACH INTL	44.5	48.5	91.7	77.7	90.4	77.7	89.3	77.7	208	4138
PANAMA CITY BAY COUNTY INTL	31.8	35.7	92.8	76.8	91.1	76.9	90.1	76.7	1242	2847
PENSACOLA INTL	30.0	33.8	93.9	77.5	92.0	77.3	90.3	77.0	1422	2706
PENSACOLA NAS	29.6	33.1	93.1	78.7	91.3	78.5	90.0	78.1	1474	2618
SARASOTA BRADENTON INTL	39.6	44.1	92.4	78.5	91.1	78.4	90.3	78.2	456	3477
SOUTHWEST FLORIDA INTL	41.3	45.4	93.4	76.6	92.2	76.6	91.0	76.5	306	3743
ST PETE-CLEARWATER INTL	42.2	45.4	92.2	77.6	91.0	77.6	90.2	77.4	443	3672
TALLAHASSEE REGIONAL	26.2	29.7	96.2	76.1	94.2	75.6	92.4	75.3	1493	2662
TAMPA INTL	39.6	43.4	92.5	77.0	91.3	77.0	90.3	77.0	505	3609
TAYLOR FIELD	29.6	33.9	93.2	75.2	91.4	75.3	90.6	75.1	1043	2756
TYNDALL AFB	31.5	35.5	91.3	78.8	90.2	78.8	89.0	78.5	1324	2644
VENICE	41.8	45.8	88.1	76.4	86.9	77.0	86.2	77.2	477	3009
VERO BEACH REGIONAL	38.8	43.2	91.8	77.6	90.5	77.7	89.4	77.7	414	3460
Georgia										
ATHENS BEN EPPS AP	22.6	26.5	95.4	74.6	93.1	74.0	90.7	73.6	2755	1804

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
ATLANTA HARTSFIELD-JACKSON	21.9	26.5	94.0	74.2	91.6	73.8	89.5	73.3	2640	1901
ATLANTA REGIONAL	19.4	23.4	93.4	73.8	91.3	73.6	89.8	73.4	3002	1578
AUGUSTA REGIONAL	22.6	26.1	97.3	76.0	94.9	75.6	92.6	75.2	2384	2074
COLUMBUS AP	25.9	29.6	96.4	74.6	94.2	74.4	92.3	74.1	2047	2359
DANIEL FIELD	27.2	29.9	96.8	74.4	93.9	73.7	91.8	73.4	2106	2329
DEKALB-PEACHTREE AP	21.1	25.5	94.2	73.6	91.8	73.3	90.2	72.8	2867	1816
DOBBINS AFB	19.5	24.5	93.1	74.4	91.1	74.2	89.2	73.6	2924	1757
FULTON COUNTY AP	21.2	25.7	94.1	74.4	91.9	74.0	90.1	73.5	2807	1765
HUNTER AAF	27.9	31.7	95.6	77.3	93.3	76.9	91.2	76.6	1610	2579
LAWSON AAF	22.5	25.9	96.7	75.9	94.6	75.8	92.4	75.5	2312	2074
LEE GILMER MEMORIAL	21.3	26.1	92.5	73.3	90.5	73.1	88.4	72.5	3006	1643
MIDDLE GEORGIA REGIONAL	23.9	27.4	96.6	75.3	94.4	75.1	92.3	74.7	2265	2153
MOODY AFB	29.1	32.6	96.1	76.5	94.2	76.2	92.6	75.8	1433	2667
RICHARD B RUSSELL REGIONAL	19.4	23.4	95.8	74.5	93.2	73.9	91.1	73.8	3030	1777
ROBINS AFB	24.8	28.0	97.0	75.8	94.9	75.8	92.7	75.2	2135	2228
SAVANNAH HILTON HEAD INTL	27.6	30.8	95.5	77.1	93.3	76.8	91.3	76.2	1723	2465
SW GEORGIA REGIONAL	26.6	29.6	96.8	75.9	94.7	75.7	92.7	75.4	1746	2547
VALDOSTA REGIONAL	27.7	30.8	96.6	76.6	94.5	76.2	92.7	75.8	1477	2627
Hawaii										
HILO INTL	61.6	62.8	85.7	74.0	84.7	73.7	83.9	73.4	0	3245
HONOLULU INTL	62.5	64.5	89.4	73.8	88.5	73.4	87.7	73.0	0	4656
KALAELOA	60.4	62.5	90.1	73.4	88.8	73.2	87.9	73.0	0	4214
KANEOHE MCAS	64.0	65.9	84.9	74.3	84.1	74.1	83.4	73.8	0	4190
Idaho										
BOISE AP	9.4	15.9	98.6	63.8	95.7	62.8	92.8	61.9	5414	1007
CALDWELL INDUSTRIAL AP	9.6	15.7	97.0	66.2	93.1	64.7	90.6	63.8	5739	692

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
COEUR D'ALENE AP	5.8	10.4	91.3	63.1	88.5	62.6	84.2	61.1	6875	316
IDAHO FALLS REGIONAL	–6.6	–0.3	92.1	60.9	89.6	60.5	86.7	59.6	7672	288
LEWISTON-NEZ PERCE CO REGL	13.0	18.8	98.5	65.3	95.0	64.5	91.4	63.2	5044	868
MAGIC VALLEY REGIONAL	7.6	12.0	95.0	62.6	92.2	62.1	89.8	61.6	6029	775
POCATELLO REGIONAL	–2.0	3.8	94.9	61.4	91.8	60.8	88.8	60.0	6941	440
Illinois										
ABRAHAM LINCOLN CAPITAL	1.1	6.9	92.6	76.7	90.4	75.6	88.0	74.1	5328	1144
AURORA MUNICIPAL	–4.9	0.7	90.7	74.1	88.2	73.2	85.5	72.0	6537	729
CHICAGO MIDWAY INTL	0.5	6.1	91.9	74.7	89.4	73.1	86.6	72.0	5850	1057
CHICAGO O'HARE INTL	–1.0	4.4	91.3	74.2	88.5	72.9	85.9	71.6	6190	882
CHICAGO ROCKFORD INTL	–5.4	0.3	90.9	74.4	88.0	72.9	85.4	71.6	6589	786
DECATUR AP	1.9	7.3	93.1	76.6	90.7	75.5	88.3	74.3	5388	1105
DUPAGE COUNT AP	–2.5	2.6	90.2	74.6	87.8	73.5	84.6	71.8	6430	750
GLENVIEW NAS	–0.7	4.8	93.7	75.0	90.2	73.3	87.1	72.1	6104	909
GREATER PEORIA REGIONAL	–0.9	4.3	92.0	76.5	89.6	75.2	87.0	73.6	5733	1057
QUAD CITY INTL	–3.5	1.8	92.4	76.2	89.7	74.9	87.1	73.2	6091	988
QUINCY REGIONAL	0.4	5.6	93.1	76.5	90.3	75.4	87.7	74.1	5497	1119
SCOTT AFB	7.2	12.1	95.2	77.7	92.7	76.9	90.3	75.9	4617	1413
ST LOUIS DOWNTOWN AP	8.6	12.7	95.4	76.6	92.7	76.3	90.4	75.2	4569	1432
U OF ILLINOIS WILLARD AP	–0.4	4.9	91.4	75.5	89.6	74.8	86.9	73.4	5693	973
Indiana										
EVANSVILLE REGIONAL	9.0	14.6	94.0	76.1	91.7	75.6	89.5	74.7	4388	1452
FORT WAYNE INTL	–0.5	5.4	90.8	74.5	88.1	73.0	85.5	71.7	5978	826
GRISSOM AFB	–1.4	5.5	90.6	75.4	88.0	74.0	85.6	72.8	5868	899
INDIANAPOLIS INTL	2.6	8.6	91.3	75.0	88.9	74.0	86.5	72.8	5249	1106
MONROE COUNTY AP	3.9	9.8	91.4	74.9	89.6	74.9	87.1	73.4	5071	1026

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
PURDUE UNIVERSITY AP	0.6	6.3	91.6	75.6	89.5	74.3	86.9	72.8	5526	998
SOUTH BEND INTL	0.4	6.3	90.4	74.0	87.7	72.4	85.0	71.1	6160	799
TERRE HAUTE INTL	2.2	8.8	92.5	76.2	90.1	75.6	87.8	74.2	5170	1084
Iowa										
AMES MUNICIPAL	–6.4	–1.5	90.9	76.2	88.3	74.7	85.8	73.1	6625	808
ANKENY REGIONAL	–3.7	1.0	93.0	75.6	90.3	75.1	87.8	73.7	6125	975
BOONE MUNICIPAL	–5.7	0.2	91.2	76.5	89.5	75.8	86.1	73.5	6484	878
DAVENPORT MUNICIPAL	–6.9	–1.4	91.1	75.0	88.2	73.9	85.8	72.6	6474	878
DES MOINES INTL	–4.3	0.6	92.8	76.2	89.7	75.0	87.1	73.4	6147	1066
DUBUQUE REGIONAL	–8.1	–2.7	88.4	75.0	85.6	73.3	83.1	71.4	7042	641
SIOUX GATEWAY AP	–7.1	–2.5	92.9	75.2	90.1	74.3	87.4	73.0	6733	914
THE EASTERN IOWA AP	–8.2	–2.6	90.4	76.3	87.6	74.5	84.8	72.7	6755	774
WATERLOO MUNICIPAL	–9.5	–4.5	91.0	75.5	88.1	73.7	85.5	72.3	7016	769
Kansas										
COLONEL JAMES JABARA AP	6.7	11.0	100.0	73.6	97.2	73.9	93.2	73.6	4504	1632
FORBES FIELD	3.5	8.9	99.0	76.0	94.8	75.4	91.4	74.6	4977	1422
JOHNSON COUNTY EXECUTIVE	4.3	9.2	96.5	75.5	92.7	75.3	90.1	74.9	4837	1413
LAWRENCE MUNICIPAL	3.4	8.6	99.2	76.6	95.5	75.8	92.0	75.4	4991	1475
MANHATTAN REGIONAL	1.9	7.5	100.2	75.3	97.0	75.3	93.1	74.6	5145	1486
MARSHALL AIRFIELD	4.8	9.2	100.7	75.0	96.8	75.1	93.6	74.8	4940	1593
MCCONNELL AFB	8.3	12.8	100.2	72.7	97.2	73.3	93.8	73.4	4284	1745
PHILIP BILLARD MUNICIPAL	4.0	8.9	98.3	75.9	94.9	75.9	91.8	75.0	4885	1506
SALINA MUNICIPAL	4.3	9.2	101.9	73.5	98.8	73.7	95.3	73.2	4799	1706
WICHITA EISENHOWER NATL	7.6	12.3	101.1	73.2	97.8	73.6	94.4	73.5	4444	1743
Kentucky										
BLUE GRASS AP	8.8	14.1	91.5	74.0	89.2	73.6	87.0	72.7	4535	1197

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
BOWLING GREEN AP	12.4	17.6	94.1	75.1	91.6	75.0	89.6	74.5	3979	1475
BOWMAN FIELD	10.7	16.3	93.4	75.1	91.2	74.7	89.3	73.9	4156	1486
CAMPBELL AAF	12.0	17.7	94.6	76.1	92.1	75.8	90.2	75.3	3773	1615
CINCINNATI NORTHERN KY INTL	5.9	11.9	91.6	74.3	89.0	73.4	86.7	72.5	4918	1115
HENDERSON CITY-COUNTY AP	8.8	14.3	93.4	76.3	91.2	75.9	90.1	75.3	4473	1396
LAKE CUMBERLAND REGIONAL	12.2	18.0	94.1	74.6	91.3	74.0	90.0	73.5	3945	1388
LOUISVILLE INTL	11.0	16.6	94.2	75.3	91.8	75.0	89.5	74.2	4057	1604
Louisiana										
ALEXANDRIA ESLER REGIONAL	26.0	28.4	98.0	76.7	95.6	77.0	93.5	76.9	1988	2533
ALEXANDRIA INTL	27.0	29.9	97.4	76.9	95.1	77.0	93.0	76.7	1850	2659
BARKSDALE AFB	24.6	27.6	99.0	75.5	96.4	75.8	93.9	76.0	2238	2424
BATON ROUGE METROPOLITAN	28.4	31.6	94.8	77.4	93.3	77.4	91.8	77.0	1576	2721
LAFAYETTE REGIONAL	30.2	33.7	94.8	77.7	93.1	77.5	91.5	77.2	1443	2857
LAKE CHARLES REGIONAL	30.5	33.8	94.7	77.7	93.0	77.7	91.4	77.6	1434	2849
LAKEFRONT AP	36.0	39.2	93.5	78.7	92.6	78.4	91.1	77.9	1072	3385
LOUIS ARMSTRONG NEW ORLEANS	33.3	36.6	94.2	77.9	92.5	77.7	91.0	77.4	1255	3000
MONROE REGIONAL	25.4	28.4	98.3	77.7	95.8	77.4	93.6	77.1	2183	2506
NEW ORLEANS NAS	31.2	34.5	93.1	77.6	91.6	77.4	90.3	77.2	1375	2728
SHREVEPORT DOWNTOWN AP	26.5	29.1	99.5	76.2	97.1	76.1	94.6	76.0	2173	2654
SHREVEPORT REGIONAL	25.9	29.0	99.2	75.7	96.7	76.0	94.4	76.0	2093	2612
Maine										
AUBURN LEWISTON MUNICIPAL	–5.9	0.1	87.8	71.2	83.6	69.7	81.1	67.6	7585	314
BANGOR INTL	–7.0	–1.8	87.7	70.5	84.2	69.0	81.1	67.0	7614	360
BRUNSWICK NAS	–2.2	2.3	86.2	70.3	82.8	68.9	80.3	67.1	7180	369
PORTLAND INTL JETPORT	0.5	5.2	86.7	71.2	83.3	69.9	80.2	68.1	6930	382
SANFORD SEACOAST REGIONAL	–6.1	0.3	89.7	71.7	85.7	70.0	82.2	68.1	7423	366

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Maryland										
ANDREWS AFB	15.1	18.7	94.1	75.0	91.0	74.0	88.5	72.9	4378	1251
BALTIMORE-WASHINGTON INTL	14.6	18.4	94.2	74.8	91.4	74.0	88.6	72.8	4491	1268
THOMAS POINT	18.2	22.0	86.9	76.4	84.9	75.9	83.1	75.2	4135	1245
Massachusetts										
BARNSTABLE MUNICIPAL	10.3	15.4	84.2	72.9	81.5	71.5	79.3	70.4	5791	526
BOSTON LOGAN INTL	8.5	13.4	90.6	72.6	87.6	71.6	84.3	70.0	5512	776
BUZZARDS BAY	13.1	17.2	76.4	N/A	74.7	N/A	73.4	N/A	5411	339
CHATHAM MUNICIPAL	12.1	17.4	83.0	72.8	80.7	71.7	78.4	70.6	5598	493
LAWRENCE MUNICIPAL	4.9	9.8	90.5	73.0	87.9	71.9	84.2	70.2	6007	684
MARTHAS VINEYARD AP	9.8	14.4	84.1	72.5	81.4	71.3	79.2	70.2	5777	451
NEW BEDFORD REGIONAL	8.6	12.4	88.2	73.4	84.5	71.5	82.0	70.2	5770	586
NORWOOD MEMORIAL	3.1	9.0	90.6	73.5	87.9	72.4	84.4	70.4	6140	606
PLYMOUTH MUNICIPAL	6.1	10.3	88.7	73.0	85.3	71.7	82.3	70.0	6052	575
SOUTH WEYMOUTH NAS	5.9	10.4	91.2	73.8	87.7	72.3	84.7	70.7	5832	646
WORCESTER REGIONAL	2.4	7.1	85.8	70.9	83.2	69.5	80.6	68.0	6603	485
Michigan										
BISHOP INTL	–0.2	4.5	89.4	73.4	86.6	71.7	84.0	70.1	6692	610
DETROIT COLEMAN YOUNG INTL	5.2	9.6	90.6	73.4	88.0	71.9	85.4	70.5	5977	883
DETROIT METRO WAYNE COUNTY AP	3.3	8.2	90.3	74.0	87.4	72.5	84.7	70.9	6066	821
GERALD R FORD INTL	2.8	7.3	89.2	73.1	86.5	71.6	83.9	70.1	6544	663
GROSSE ILE MUNICIPAL	6.5	9.8	89.8	74.5	86.1	73.7	83.5	72.6	5887	858
JACKSON COUNTY AP	0.5	5.4	88.6	73.2	85.9	71.8	83.2	70.2	6596	581
KALAMAZOO BATTLE CREEK INTL	2.5	7.4	90.2	73.1	87.7	71.7	84.3	70.2	6288	731
LANSING CAPITAL REGION INTL	–0.5	4.7	89.0	73.2	86.2	71.6	83.5	70.0	6774	579
MBS INTL	0.4	4.7	89.4	73.0	86.3	71.2	83.6	70.0	6854	586

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
MUSKEGON COUNTY AP	5.5	9.4	86.2	72.2	83.7	70.7	81.6	69.6	6565	538
OAKLAND COUNTY INTL	1.0	5.4	89.7	73.0	86.4	71.1	83.7	69.6	6632	654
SELFRIDGE ANGB	1.7	6.8	90.0	74.0	86.5	72.1	83.9	70.8	6413	663
ST CLAIR COUNTY INTL	0.0	5.0	89.9	73.5	85.9	71.3	82.3	69.6	6800	458
WEST MICHIGAN REGIONAL	6.8	10.1	89.5	73.4	86.4	71.9	83.6	70.7	6218	663
WILLOW RUN AP	1.0	6.4	91.2	73.8	88.3	72.4	85.5	70.7	6342	733
Minnesota										
ANOKA COUNTY AP	–8.8	–4.5	90.2	74.8	87.7	73.7	83.9	71.7	7530	617
CRYSTAL AP	–9.5	–6.0	90.7	73.4	88.1	72.0	84.5	69.9	7599	724
DULUTH INTL	–17.2	–12.0	84.2	69.7	81.1	67.1	78.2	65.3	9286	218
FLYING CLOUD AP	–10.6	–6.3	90.8	74.2	88.2	72.7	85.0	70.9	7415	807
MANKATO MUNICIPAL	–12.3	–8.2	89.9	73.7	86.3	71.9	83.5	70.6	7705	628
MINNEAPOLIS-ST PAUL INTL	–10.6	–5.8	90.8	73.3	87.8	72.0	84.9	70.2	7477	782
ROCHESTER INTL	–12.4	–7.6	87.7	73.3	84.7	71.7	82.2	70.3	7832	528
SKY HARBOR AP	–10.5	–6.3	86.2	71.9	82.3	69.3	80.5	68.0	8561	310
SOUTH ST PAUL MUNICIPAL	–8.8	–4.4	90.8	73.1	88.1	71.6	84.6	69.8	7371	770
ST CLOUD REGIONAL	–16.8	–11.3	89.4	72.5	86.3	70.7	83.4	68.7	8425	484
ST PAUL DOWNTOWN AP	–10.0	–5.8	90.6	73.8	87.9	72.5	84.3	70.5	7465	753
Mississippi										
HATTIESBURG-LAUREL AP	25.0	27.7	96.6	75.7	93.1	75.1	91.1	74.9	2086	2273
JACKSON INTL	23.6	26.9	96.3	76.1	94.2	76.1	92.3	75.9	2272	2315
KEESLER AFB	30.6	34.7	94.3	80.2	92.2	79.4	90.6	78.8	1432	2800
KEY FIELD	22.6	26.1	96.1	75.8	93.9	75.9	92.0	75.7	2351	2161
MERIDIAN NAS MCCAIN FIELD	22.5	26.6	96.8	76.2	94.6	76.2	92.6	75.9	2343	2232
TUPELO REGIONAL	19.5	23.8	96.2	76.0	93.8	75.7	91.7	75.5	2911	2010
Missouri										

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	**99.6%**	**99%**	**DB**	**MCWB**	**DB**	**MCWB**	**DB**	**MCWB**	**HDD**	**CDD 65**
CAPE GIRARDEAU REGIONAL	10.1	15.9	94.4	77.0	92.3	76.7	90.2	75.9	4193	1513
COLUMBIA REGIONAL	3.6	9.2	94.7	75.7	91.7	75.9	89.0	74.9	4917	1276
CR WHEELER DOWNTOWN	5.8	10.2	97.3	76.1	94.1	75.7	91.3	75.2	4567	1663
JEFFERSON CITY MEMORIAL	6.5	11.6	96.1	75.9	92.8	75.2	90.4	74.6	4574	1444
JOPLIN REGIONAL	8.7	14.0	97.4	74.8	94.4	75.3	91.6	74.8	4044	1669
KANSAS CITY INTL	2.4	7.5	96.0	76.5	92.7	76.1	89.7	75.4	5027	1376
LAMBERT-ST LOUIS INTL	7.5	12.7	96.2	76.7	93.5	76.1	91.1	75.0	4403	1676
SPIRIT OF ST LOUIS AP	6.2	11.7	95.5	77.1	92.8	76.3	90.4	75.2	4672	1398
SPRINGFIELD-BRANSON REGIONAL	7.1	12.8	95.5	74.0	92.4	74.1	89.6	74.0	4428	1406
Montana										
BERT MOONEY AP	–17.3	–9.1	87.9	57.4	84.7	56.4	81.7	55.9	9095	79
BILLINGS LOGAN INTL	–9.5	–3.2	94.7	62.7	91.3	61.9	88.1	61.4	6738	648
BOZEMAN YELLOWSTONE INTL	–15.1	–7.7	92.1	61.1	88.7	60.4	85.2	59.4	8172	237
GREAT FALLS	–13.8	–6.9	90.6	60.3	87.3	59.5	84.1	58.9	7706	333
GREAT FALLS INTL	–16.2	–9.5	92.5	60.9	89.1	60.2	85.6	59.3	7557	336
MALMSTROM AFB	–14.9	–8.5	94.5	62.7	90.8	61.5	87.4	60.5	6875	483
MISSOULA INTL	–3.5	3.3	93.1	61.8	89.8	61.2	86.3	60.2	7353	336
Nebraska										
CENTRAL NEBRASKA REGIONAL	–3.2	1.9	95.6	74.0	92.4	73.2	89.4	72.0	6092	1054
EPPLEY AIRFIELD	–3.2	1.4	94.8	76.0	91.8	75.3	88.9	73.8	6024	1165
LINCOLN MUNICIPAL	–2.6	2.0	96.6	74.9	93.3	74.5	90.4	73.6	5969	1191
NORTH OMAHA AP	–6.1	–0.1	94.0	75.0	90.9	74.6	88.0	73.0	5981	1093
OFFUTT AFB	–2.5	1.7	95.0	76.5	91.4	75.8	89.0	74.5	5950	1166
Nevada										
MCCARRAN INTL	31.7	34.3	108.5	67.5	106.3	66.7	104.1	65.9	1962	3557
NELLIS AFB	28.5	31.7	109.2	67.3	107.2	66.7	104.9	65.9	2072	3454

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
RENO TAHOE INTL	13.1	18.0	96.4	61.4	93.8	60.4	91.3	59.2	4961	872
New Hampshire										
CONCORD MUNICIPAL	–3.0	2.1	90.0	71.3	87.0	69.8	84.1	68.5	7061	477
JAFFREY-SILVER RANCH AP	–2.3	2.1	87.1	70.0	83.8	68.6	81.3	67.2	7280	379
MANCHESTER-BOSTON REGIONAL	2.2	7.2	91.1	71.9	88.5	70.6	85.6	69.3	6207	732
PORTSMOUTH INTL	2.7	7.7	89.5	72.5	86.0	71.1	82.8	69.5	6396	556
New Jersey										
ATLANTIC CITY INTL	12.3	16.5	92.3	75.2	89.3	73.8	86.5	72.7	4806	1035
MCGUIRE AFB	11.7	15.9	92.9	75.4	90.3	74.5	87.7	73.2	4849	1073
MILLVILLE MUNICIPAL	11.6	16.2	92.1	75.0	89.4	74.0	86.9	72.8	4845	1057
MONMOUTH EXECUTIVE	11.6	16.0	91.0	73.9	88.4	72.7	84.5	71.3	5052	919
NEWARK LIBERTY INTL	12.8	16.9	94.3	74.4	91.2	73.0	88.3	71.9	4633	1270
TETERBORO AP	11.9	16.1	92.6	74.1	89.9	72.9	87.3	71.5	4915	1090
TRENTON MERCER AP	11.8	16.1	92.6	74.3	89.9	73.2	87.3	72.3	4916	1057
New Mexico										
ALAMOGORDO-WHITE SANDS AP	20.7	24.9	99.8	63.5	98.6	63.7	95.2	63.2	2897	1909
ALBUQUERQUE INTL	18.3	21.9	95.5	59.9	93.3	59.7	91.0	59.5	3956	1438
CANNON AFB	13.7	18.2	98.5	63.2	95.5	63.2	93.0	63.6	3718	1442
CLOVIS MUNICIPAL	12.2	17.6	97.3	64.1	94.6	63.9	91.4	63.9	4053	1247
FOUR CORNERS REGIONAL	7.8	12.3	95.9	59.6	93.3	59.1	91.0	58.9	5271	988
HOLLOMAN AFB	18.7	22.1	99.8	62.7	97.5	62.5	95.2	62.3	3217	1838
ROSWELL INTL AIR CENTER	17.7	21.6	101.1	64.4	98.5	64.7	96.2	64.6	3095	1976
WHITE SANDS	18.4	22.5	99.0	63.7	96.5	63.9	94.2	63.8	2946	1811
New York										
ALBANY INTL	–0.3	4.7	88.9	72.6	86.1	71.0	83.4	69.7	6473	639
AMBROSE LIGHT	13.7	17.9	84.0	N/A	80.9	N/A	78.4	N/A	4850	720

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
BUFFALO NIAGARA INTL	3.5	7.4	86.3	71.2	83.9	69.9	81.6	68.9	6460	580
CHAUTAUQUA COUNTY AP	0.6	4.8	82.4	69.7	81.1	68.6	78.8	66.9	7185	304
DUTCHESS COUNTY AP	2.7	8.4	91.3	73.5	88.4	72.3	85.6	70.9	6003	725
ELMIRA CORNING REGIONAL	–0.1	4.8	89.6	71.5	86.4	69.9	83.7	68.7	6726	470
GREATER BINGHAMTON AP	0.0	4.5	85.2	69.8	82.3	68.3	79.8	67.2	7033	403
GREATER ROCHESTER INTL	2.8	7.1	88.5	73.0	85.6	71.0	82.9	69.6	6466	571
JOHN F KENNEDY INTL	14.1	18.0	89.9	73.0	86.6	71.7	83.9	70.9	4777	1014
LAGUARDIA AP	14.3	18.4	92.5	73.9	89.6	72.5	86.9	71.4	4493	1284
LONG ISLAND MACARTHUR AP	11.9	15.9	88.6	73.5	85.7	71.9	83.1	71.0	5199	839
NIAGARA FALLS INTL	2.8	6.9	87.8	72.4	85.2	71.0	82.5	69.5	6595	581
ONEIDA COUNTY AP	–4.8	1.2	87.1	72.4	84.2	70.6	81.8	69.0	7010	471
PLATTSBURGH INTL	–8.7	–2.7	86.8	71.0	83.0	69.5	80.3	68.2	7534	374
REPUBLIC AP	12.4	17.0	89.9	73.8	86.3	71.9	83.6	71.3	5039	911
STEWART INTL	3.5	9.1	90.2	72.2	87.0	71.3	84.1	69.8	5940	713
SYRACUSE HANCOCK INTL	–0.7	4.9	89.2	73.0	86.4	71.2	83.7	69.8	6495	625
WESTCHESTER COUNTY AP	9.2	13.5	89.6	73.4	86.4	71.8	83.8	70.5	5471	773
North Carolina										
ALBERT J ELLIS AP	20.6	24.8	94.7	77.0	91.4	76.1	90.2	75.5	2950	1737
ASHEVILLE REGIONAL	15.4	19.7	88.0	71.2	85.7	70.4	83.6	69.7	4076	857
CHARLOTTE DOUGLAS INTL	21.6	25.3	94.2	74.7	91.8	74.1	89.6	73.4	3041	1690
FAYETTEVILLE REGIONAL	22.9	26.7	96.3	75.9	93.4	74.9	91.2	74.4	2705	1984
HICKORY REGIONAL	20.0	24.0	92.4	72.7	90.0	72.4	87.8	71.7	3454	1378
NEW RIVER MCAS	23.4	26.8	93.0	78.0	90.7	77.2	88.7	76.4	2536	1924
PIEDMONT TRIAD INTL	19.3	22.8	92.5	74.1	90.3	73.5	88.1	72.7	3533	1454
PITT-GREENVILLE AP	20.8	24.9	95.0	76.1	92.9	75.1	90.8	74.5	2994	1867
POPE AFB	21.1	25.0	97.1	76.3	94.6	75.7	91.8	74.8	2817	2034

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
RALEIGH-DURHAM INTL	20.4	24.2	95.0	75.5	92.5	75.0	90.1	74.3	3208	1685
SEYMOUR-JOHNSON AFB	22.7	26.5	96.6	76.2	93.7	75.7	91.3	75.1	2685	2013
SIMMONS AAF	22.4	26.2	96.7	75.7	94.0	75.2	91.4	74.5	2741	2034
SMITH REYNOLDS AP	20.2	24.4	92.2	73.7	90.2	73.0	88.1	72.4	3416	1469
WILMINGTON INTL	24.7	27.9	93.5	77.9	91.2	77.0	89.0	76.3	2364	2020
North Dakota										
BISMARCK MUNICIPAL	–17.5	–12.2	93.1	69.9	89.4	68.9	86.1	67.6	8452	529
GRAND FORKS AFB	–20.1	–15.5	89.6	72.2	86.1	70.2	83.3	68.5	9311	420
GRAND FORKS INTL	–21.4	–16.6	89.0	71.7	85.9	69.5	83.2	67.9	9363	418
HECTOR INTL	–18.7	–13.9	90.0	72.3	87.0	70.3	84.3	68.7	8740	551
MINOT AFB	–22.3	–17.3	90.5	68.9	87.0	68.1	83.7	66.3	9348	355
MINOT INTL	–18.1	–13.2	90.5	69.3	86.9	68.2	83.7	66.2	8794	431
Ohio										
AKRON-CANTON REGIONAL	3.1	8.2	88.6	72.6	86.0	71.4	83.5	69.9	5993	703
CINCINNATI MUNICIPAL LUNKEN	8.2	13.8	92.7	75.1	90.1	74.3	87.8	73.4	4755	1123
CLEVELAND HOPKINS INTL	4.6	10.0	89.4	73.6	86.8	72.2	84.2	70.9	5801	786
FAIRFIELD COUNTY AP	2.3	9.4	90.5	73.9	88.3	73.1	85.9	71.7	5449	816
FINDLAY AP	1.1	6.9	90.3	73.5	87.8	72.2	85.1	70.6	5912	807
JMCOX DAYTON INTL	2.3	8.5	90.3	73.8	87.9	72.8	85.5	71.3	5481	952
MANSFIELD LAHM REGIONAL	1.5	7.0	88.0	72.8	85.6	71.5	83.2	70.1	6110	670
OHIO STATE UNIVERSITY AP	4.4	9.9	90.5	73.5	88.2	72.7	85.8	71.3	5450	928
PORT COLUMBUS INTL	5.5	11.0	91.3	73.7	89.0	72.7	86.6	71.5	5185	1055
RICKENBACKER INTL	6.8	12.0	92.7	74.3	90.4	73.3	88.2	72.3	5025	1137
TOLEDO EXPRESS AP	1.5	7.0	91.2	74.1	88.4	72.3	85.7	70.9	6050	808
WRIGHT-PATTERSON AFB	3.2	9.6	91.1	74.4	88.6	73.2	86.2	71.8	5363	928
YOUNGSTOWN-WARREN REGIONAL	3.3	8.1	88.2	72.4	85.6	70.7	83.2	69.4	6159	582

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Oklahoma										
HENRY POST AAF	15.7	20.2	103.5	72.5	100.3	72.8	97.4	72.6	3154	2270
LAWTON-FORT SILL REGIONAL	16.2	20.0	104.5	72.7	101.8	72.8	99.2	73.3	3157	2375
RICHARD L JONES JR AP	14.3	18.5	101.6	75.3	99.0	76.3	96.1	76.2	3513	2084
STILLWATER REGIONAL	12.1	17.7	102.5	74.4	99.5	74.9	96.7	74.8	3610	2072
TINKER AFB	13.8	18.5	100.4	73.5	97.4	73.6	94.6	73.9	3387	1997
TULSA INTL	13.5	18.3	100.4	75.2	97.6	75.6	94.8	75.5	3450	2108
VANCE AFB	10.3	15.6	101.9	73.1	99.4	73.3	96.6	73.5	3920	1960
WILEY POST AP	13.8	18.6	100.4	73.3	98.3	73.5	95.1	73.6	3473	2111
WILL ROGERS WORLD AP	14.5	19.1	100.7	73.7	97.9	74.0	95.0	74.0	3420	2016
Oregon										
AURORA STATE AP	25.2	28.1	91.4	67.1	88.1	66.6	83.9	65.1	4461	375
CORVALLIS MUNICIPAL	24.7	27.5	92.7	67.0	89.7	66.0	85.5	64.4	4306	393
EUGENE AP	22.9	26.9	91.8	66.7	88.0	65.7	84.3	64.5	4664	282
MCMINNVILLE MUNICIPAL	25.8	28.1	92.2	66.7	88.4	66.0	84.2	64.7	4654	301
PORTLAND INTL	25.0	29.0	91.2	67.5	87.1	66.4	83.4	65.2	4232	447
PORTLAND-HILLSBORO AP	22.7	26.5	91.7	67.8	88.0	66.9	83.8	65.3	4792	276
ROBERTS FIELD	5.1	12.0	93.1	61.4	90.2	60.7	87.0	59.4	6541	238
ROGUE VALLEY INTL	22.8	26.0	99.0	67.1	95.5	65.7	92.3	64.6	4262	864
SALEM MUNICIPAL	23.7	27.3	92.2	66.9	88.1	65.9	84.3	64.6	4503	341
Pennsylvania										
ALLEGHENY COUNTY AP	6.7	10.9	89.2	72.1	86.5	70.9	84.2	69.7	5389	838
ALTOONA-BLAIR COUNTY AP	6.4	10.4	88.3	71.8	85.5	70.6	82.9	69.5	5881	619
BUTLER COUNTY AP	3.1	8.8	88.2	72.0	84.5	70.4	82.3	69.0	6100	561
CAPITAL CITY AP	12.3	16.2	92.3	74.0	89.5	72.5	86.9	71.5	5009	1066
ERIE INTL	6.1	10.4	86.7	73.0	84.2	71.7	81.9	70.6	6035	668

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
HARRISBURG INTL	11.7	15.7	92.4	75.1	89.5	73.8	86.8	72.6	5030	1119
LEHIGH VALLEY INTL	8.7	13.0	91.0	73.7	88.3	72.5	85.7	71.3	5482	851
NORTHEAST PHILADELPHIA AP	13.4	17.7	93.2	74.8	90.5	73.6	88.1	72.5	4692	1194
PHILADELPHIA INTL	14.5	18.6	93.6	74.9	90.9	74.0	88.3	72.6	4425	1357
PITTSBURGH INTL	5.6	10.4	89.2	72.0	86.6	70.7	84.3	69.6	5532	785
READING REGIONAL	9.9	14.2	92.6	74.4	89.7	73.2	86.9	72.0	5170	1000
WASHINGTON COUNTY AP	2.7	8.8	88.3	71.2	85.5	69.9	82.7	68.6	5958	540
WILKES-BARRE SCRANTON INTL	4.7	9.4	89.1	71.7	86.2	70.2	83.6	68.9	6007	653
WILLOW GROVE NAS	12.5	16.6	91.9	73.9	89.4	72.6	86.8	71.5	4935	1034
Rhode Island										
T F GREEN AP	8.8	13.3	90.1	73.2	86.7	71.8	83.9	70.4	5481	765
South Carolina										
CHARLESTON INTL	27.5	30.9	94.3	78.0	92.2	77.5	90.3	76.9	1845	2379
COLUMBIA METROPOLITAN	23.4	27.0	97.1	75.4	94.6	74.8	92.5	74.5	2432	2202
FLORENCE REGIONAL	24.1	27.3	96.1	76.6	93.5	75.9	91.4	75.4	2396	2114
FOLLY ISLAND	31.6	35.1	87.5	77.9	86.2	78.0	85.1	77.7	1878	2157
GREENVILLE-SPARTANBURG INTL	21.9	25.6	94.4	73.7	91.8	73.4	89.5	72.7	3016	1642
SHAW AFB	23.7	27.2	96.0	75.7	93.4	75.4	91.2	75.0	2435	2066
South Dakota										
ELLSWORTH AFB	–8.2	–2.8	95.2	66.4	91.4	65.7	88.2	65.1	7028	678
RAPID CITY REGIONAL	–8.8	–3.4	96.7	66.0	92.7	65.7	89.0	64.9	7069	659
SIOUX FALLS REGIONAL	–11.3	–6.5	91.5	73.7	88.4	72.9	85.6	71.3	7510	739
Tennessee										
CHATTANOOGA AP	19.6	23.7	95.0	74.5	92.6	74.2	90.4	73.6	3093	1790
MCGHEE TYSON AP	17.1	21.6	92.7	73.9	90.4	73.5	88.3	72.9	3532	1525
MCKELLAR-SIPES REGIONAL	15.7	19.9	95.1	76.5	92.8	76.5	90.8	76.0	3436	1739

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

			Cooling DB/MCWB						Heat./Cool. Degree-Days	
	Heating DB		0.4%		1%		2%			
Station	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
MEMPHIS INTL	19.5	23.8	96.6	76.9	94.4	76.4	92.5	76.0	2884	2266
MILLINGTON MUNICIPAL	18.6	23.1	99.2	80.3	96.9	78.7	93.5	77.4	3065	2125
NASHVILLE INTL	15.4	19.9	94.6	74.8	92.3	74.6	90.2	73.9	3491	1736
TRI-CITIES REGIONAL	13.5	17.9	90.2	71.8	88.0	71.3	86.0	70.8	4177	1036
Texas										
ABILENE REGIONAL	20.1	24.6	100.3	70.4	98.1	70.5	95.9	70.7	2444	2488
AL MANGHAM JR REGIONAL	25.0	27.6	99.1	75.7	96.8	75.8	93.2	75.6	2144	2460
AMARILLO RICK HUSBAND INTL	10.1	15.8	98.6	65.4	95.8	65.8	93.2	65.8	4043	1471
ANGELINA COUNTY AP	26.9	29.7	99.0	76.1	96.5	76.4	94.3	76.3	1831	2709
AUSTIN-BERGSTROM INTL	26.7	30.2	100.2	74.1	98.2	74.4	96.2	74.5	1634	3005
BROWNSVILLE INTL	38.7	42.5	95.7	77.8	94.5	77.8	93.3	77.8	509	4073
CORPUS CHRISTI INTL	34.6	38.3	96.8	77.6	95.3	77.7	93.7	77.6	826	3625
CORPUS CHRISTI NAS	37.2	41.5	92.8	79.3	91.6	79.4	90.8	79.3	705	3752
DALLAS EXECUTIVE	24.9	28.0	101.7	74.1	99.4	74.6	97.2	74.4	2112	2854
DALLAS FORT WORTH INTL	23.3	27.6	101.4	74.2	99.1	74.5	96.9	74.6	2161	2878
DALLAS HENSLEY FIELD NAS	21.5	27.2	99.6	75.5	97.5	75.4	95.3	75.0	2171	2723
DALLAS LOVE FIELD	24.7	28.7	101.5	74.7	99.3	75.2	97.2	74.9	2031	3024
DEL RIO INTL	31.5	34.6	102.1	71.9	100.1	72.1	98.3	72.2	1269	3534
DRAUGHON-MILLER CENTRAL TEXAS	25.1	28.0	100.1	74.0	98.9	74.0	96.8	74.0	1978	2767
DYESS AFB ABILENE	19.1	23.4	102.4	72.0	100.1	71.8	98.0	71.7	2509	2644
EASTERWOOD FIELD	28.4	31.7	100.0	75.4	98.0	75.6	95.9	75.5	1568	3100
EL PASO INTL	25.1	28.4	101.1	64.1	98.8	63.7	96.6	63.6	2300	2521
FORT WORTH ALLIANCE AP	21.2	25.4	102.3	73.9	100.0	74.4	97.7	74.2	2397	2716
FORT WORTH NAS	24.1	28.1	102.5	73.1	100.2	73.6	98.0	73.7	2092	2978
FT WORTH MEACHAM INTL	22.5	26.8	101.5	73.9	99.3	74.5	97.1	74.3	2241	2778
GALVESTON SCHOLES INTL	36.5	39.6	92.0	79.1	90.9	79.2	90.0	79.1	990	3344

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
GEORGE BUSH INTERCONTINENTAL	31.1	34.1	97.5	76.4	95.5	76.5	93.8	76.5	1340	3123
GEORGETOWN MUNICIPAL	26.5	28.3	99.0	72.8	96.9	72.9	94.5	72.9	1948	2688
HOOKS MEMORIAL	29.7	32.9	98.7	75.7	96.2	76.0	93.5	76.2	1458	3016
HOUSTON ELLINGTON AP	34.1	37.0	96.8	78.1	94.9	78.5	93.0	78.5	1167	3197
JACK BROOKS REGIONAL	31.6	35.0	95.0	77.7	93.2	77.9	91.7	77.9	1328	2948
KILLEEN MUNICIPAL	26.3	28.9	100.0	74.0	98.9	74.1	96.8	74.1	1889	2861
LACKLAND AFB	29.7	33.1	100.4	74.1	99.0	74.1	96.9	74.1	1359	3278
LAREDO INTL	36.2	39.2	104.7	74.5	102.4	74.3	100.4	74.2	783	4454
LAUGHLIN AFB	30.7	34.3	104.3	72.0	102.0	72.5	99.9	72.6	1179	3710
LONGVIEW EAST TEXAS REGIONAL	25.5	28.1	100.2	74.9	97.9	75.2	95.2	75.3	2121	2628
LUBBOCK INTL	15.9	20.2	99.4	66.2	97.1	66.9	94.7	66.9	3256	1915
MCALLEN INTL	38.6	42.1	100.4	75.9	99.0	76.4	97.4	76.3	532	4502
MCGREGOR EXECUTIVE	24.8	27.7	101.7	74.4	99.5	74.6	97.4	74.5	2087	2794
MCKINNEY NATL	20.8	25.0	101.0	74.3	99.1	75.0	96.9	74.9	2511	2546
MIDLAND INTL	20.5	24.4	101.1	66.7	98.6	67.1	96.5	67.3	2554	2392
NEW BRAUNFELS MUNICIPAL	28.3	31.4	100.3	73.7	98.6	73.9	96.6	73.9	1510	3061
PORT ARANSAS	37.5	41.6	86.2	78.0	85.4	78.2	85.0	78.1	795	3088
RANDOLPH AFB	28.5	31.9	100.2	73.7	98.6	73.7	96.5	73.8	1455	3103
REESE AFB LUBBOCK	14.7	19.4	101.0	67.0	97.8	67.3	95.1	67.2	3182	1831
ROBERT GRAY AFF	25.7	29.7	100.4	73.0	99.0	73.2	96.8	73.3	1802	2906
SABINE PASS	32.3	36.2	89.0	77.4	87.5	77.6	86.6	77.6	1428	2660
SAN ANGELO REGIONAL	22.2	26.0	101.7	70.0	99.5	69.8	97.3	69.8	2185	2642
SAN ANTONIO INTL	29.9	33.0	99.3	73.5	97.6	73.5	95.8	73.8	1380	3222
SAN MARCOS REGIONAL	27.6	30.2	99.8	74.0	98.8	74.1	96.7	74.1	1622	3032
STINSON MUNICIPAL	30.6	33.8	101.1	73.6	99.3	73.9	97.3	73.6	1274	3399
VALLEY INTL	37.1	41.1	98.8	77.3	97.2	77.4	95.8	77.4	578	4104

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
VICTORIA REGIONAL	31.4	34.5	97.9	76.5	95.9	76.6	94.1	76.6	1164	3247
WACO REGIONAL	24.6	28.2	101.3	74.3	99.2	74.7	97.2	74.8	2008	2904
WICHITA FALL REGIONAL	18.6	22.7	103.8	72.6	101.2	72.7	98.6	72.9	2793	2546
WILLIAM P HOBBY AP	33.6	36.7	96.0	77.1	94.1	77.1	92.6	77.0	1139	3222
Utah										
HILL AFB	8.7	12.2	93.9	60.9	91.3	60.2	89.2	59.8	5977	953
LOGAN-CACHE AP	–5.8	0.3	94.8	61.6	92.2	61.0	89.9	60.4	7196	507
PROVO MUNICIPAL	7.5	11.9	94.6	62.4	91.2	62.1	89.7	61.8	5972	778
SALT LAKE CITY INTL	9.9	14.4	98.0	62.5	95.4	61.9	92.9	61.4	5466	1262
ST GEORGE MUNICIPAL	25.2	27.8	106.2	66.2	103.4	65.2	100.4	64.3	2974	2721
Vermont										
BURLINGTON INTL	–7.3	–1.8	88.3	71.3	85.4	69.8	82.5	68.4	7227	525
Virginia										
DANVILLE REGIONAL	18.3	21.7	93.9	74.5	91.3	74.1	89.6	73.4	3649	1440
DAVISON AAF	15.1	19.2	96.0	75.7	93.1	74.7	90.5	73.7	4229	1357
DINWIDDIE COUNTY	16.3	19.4	98.0	77.0	94.8	76.1	91.5	74.8	3682	1610
LANGLEY AFB	20.8	24.9	92.8	76.2	90.5	75.5	88.3	74.7	3415	1584
LESSBURG EXECUTIVE	14.2	18.2	95.0	76.3	92.2	75.0	90.3	74.2	4431	1347
LYNCHBURG REGIONAL	15.8	19.7	92.1	73.8	89.6	73.0	87.1	72.0	4192	1112
MANSAS REGIONAL	11.7	16.1	93.0	74.2	90.7	73.9	88.4	72.9	4826	1075
NEWPORT NEWS WILLIAMSBURG	20.4	24.3	94.5	77.1	92.0	76.1	89.9	75.2	3408	1638
NORFOLK INTL	22.9	26.6	93.7	76.7	91.3	76.0	88.9	75.0	3165	1715
NORFOLK NAS	24.0	27.6	94.4	77.1	91.9	76.3	89.9	75.6	2972	1889
OCEANA NAS	21.9	25.9	92.9	77.3	90.4	76.3	88.2	75.4	3262	1594
QUANTICO MCAF	17.0	20.5	92.6	76.3	90.2	75.6	87.9	74.6	4129	1376
RICHMOND INTL	18.5	22.1	95.1	75.7	92.7	75.1	90.1	74.0	3634	1564

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
ROANOKE-BLACKSBURG REGIONAL	16.8	20.6	92.2	72.7	89.7	71.9	87.3	71.1	3961	1239
RONALD REAGAN WASHINGTON NATL	17.9	21.6	94.7	75.5	92.0	74.7	89.4	73.5	3901	1587
SHENANDOAH VALLEY REGIONAL	11.8	16.4	93.2	73.8	90.9	73.4	88.7	72.7	4467	1129
VIRGINIA TECH MONTGOMERY EXEC	11.0	15.9	89.7	72.8	86.5	71.2	83.9	70.5	4798	787
WASHINGTON DULLES INTL	13.3	17.3	93.5	74.6	90.9	73.7	88.3	72.7	4591	1188
Washington										
ARLINGTON MUNICIPAL	19.5	24.5	82.0	66.0	79.3	64.4	75.3	63.0	5401	62
BELLINGHAM INTL	19.2	23.7	79.4	65.3	76.2	63.9	73.2	62.2	5339	56
BREMERTON NATL	22.6	26.6	85.6	65.1	81.7	63.5	78.8	62.2	5641	98
FAIRCHILD AFB	5.2	11.2	91.8	62.0	89.0	61.3	85.6	60.5	6792	401
FELTS FIELD	8.5	14.2	94.4	64.9	91.0	63.7	87.9	62.7	6122	464
GRAY AFF	20.0	24.5	87.5	65.5	83.1	64.2	79.6	62.8	5164	153
KING COUNTY INTL AP	25.8	29.4	85.6	65.3	81.9	64.0	79.0	62.7	4354	268
MCCHORD AFB	20.5	24.5	86.3	65.2	82.4	63.7	79.2	62.5	5161	134
OLYMPIA REGIONAL	20.4	24.3	87.3	66.0	83.3	64.7	79.7	63.4	5377	110
PAINE FIELD	25.0	28.9	80.1	63.5	76.3	62.2	73.1	61.0	5192	81
PEARSON FIELD	24.1	27.2	90.7	66.4	87.2	65.8	83.0	64.6	4416	382
SANDERSON FIELD	22.3	25.8	87.4	65.2	82.9	64.3	79.0	62.7	5491	100
SEATTLE-TACOMA INTL	25.4	29.5	85.3	65.1	81.6	63.8	78.2	62.7	4705	196
SOUTHWEST WASHINGTON REGIONAL	22.9	26.8	87.9	67.2	82.3	65.5	79.4	63.9	4832	182
SPOKANE INTL	5.1	11.2	92.9	62.8	89.6	61.8	86.2	60.6	6627	461
TACOMA NARROWS AP	27.3	30.9	83.6	64.3	80.3	62.9	76.7	61.7	4792	145
TRI-CITIES AP	10.2	16.4	99.2	69.4	96.2	67.8	92.6	66.7	4945	815
WALLA WALLA REGIONAL	11.0	17.3	98.5	66.3	94.7	64.9	91.0	63.8	4810	934
WEST POINT	29.6	33.1	70.5	61.0	68.1	60.3	66.2	59.5	4927	9
YAKIMA AIR TERMINAL	8.3	13.8	96.4	66.2	93.3	65.4	89.9	63.9	5845	556

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
West Virginia										
HUNTINGTON TRI-STATE AP	10.8	15.8	91.7	73.8	89.2	73.3	86.9	72.4	4398	1143
MID-OHIO VALLEY REGIONAL	8.6	13.6	90.6	73.8	88.1	72.8	85.8	71.6	4886	943
YEAGER AP	10.7	15.8	91.2	73.1	88.8	72.6	86.5	71.8	4402	1078
Wisconsin										
APPLETON INTL	–6.0	–0.3	88.4	75.2	84.9	72.8	82.3	70.9	7210	608
AUSTIN STRAUBEL INTL	–7.9	–2.8	88.0	73.7	85.0	71.7	82.4	70.1	7590	480
CENTRAL WISCONSIN AP	–11.0	–7.3	86.5	72.3	83.6	70.5	81.2	68.1	8303	361
CHIPPEWA VALLEY REGIONAL	–13.1	–8.0	90.0	73.1	86.7	71.0	84.0	69.2	7843	587
DANE COUNTY REGIONAL	–6.3	–1.2	89.3	74.1	86.6	72.4	83.9	71.0	7083	640
FOND DU LAC COUNTY AP	–6.3	–1.0	88.9	73.5	85.8	71.5	82.8	70.1	7173	601
GENERAL MITCHELL INTL	–1.0	3.6	89.6	74.4	86.4	72.3	83.4	70.6	6676	693
KENOSHA REGIONAL	–2.4	2.2	90.3	74.3	87.5	73.2	84.0	71.2	6735	629
LA CROSSE MUNICIPAL	–9.2	–4.2	91.3	74.9	88.4	73.0	85.7	71.4	7026	820
MANITOWOC COUNTY AP	–4.3	0.5	84.7	71.7	81.9	70.4	79.5	68.4	7573	354
SHEBOYGAN	–1.8	2.8	83.0	71.5	79.5	70.6	76.6	69.8	7248	340
SHEBOYGAN COUNTY MEMORIAL	–4.5	0.1	88.4	73.9	84.5	71.4	81.9	70.0	7469	446
WAUSAU DOWNTOWN AP	–12.0	–7.1	87.6	71.7	84.4	69.4	81.8	67.6	8025	455
WITTMAN REGIONAL	–6.2	–1.6	88.5	73.6	85.4	71.6	82.4	70.1	7343	571
Wyoming										
CASPER-NATRONA COUNTY INTL	–8.5	–1.0	94.0	59.6	91.2	59.0	88.4	58.5	7308	469
CHEYENNE REGIONAL	–4.1	2.6	89.5	58.1	86.9	57.6	84.0	57.2	7056	355
Canada										
Alberta										
BOW ISLAND	–20.7	–14.4	88.8	64.4	85.4	63.2	81.9	62.3	8604	200
CALGARY INTL	–19.2	–12.8	83.4	60.9	79.8	59.9	76.5	58.8	9197	67

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
CANADIAN OLYMPIC PARK UPPER	–17.2	–11.6	82.5	59.4	78.8	58.1	75.3	57.2	9071	74
EDMONTON CITY CENTRE AWOS	–20.1	–14.5	83.2	64.4	79.9	62.8	76.8	61.0	9423	132
EDMONTON INTL	–26.7	–20.6	82.1	64.3	78.7	62.9	75.8	61.0	10473	45
EDMONTON NAMAO AWOS	–22.2	–16.6	82.1	64.3	78.8	62.5	75.6	60.6	10060	67
FORT MCMURRAY CS	–30.5	–25.5	83.7	63.8	80.0	61.8	77.0	60.2	11244	75
GRANDE PRAIRIE	–32.7	–24.4	81.8	62.2	78.3	60.6	75.2	59.0	10674	47
LACOMBE CDA 2	–25.6	–19.0	82.6	64.8	79.0	63.1	75.9	61.3	10318	41
LETHBRIDGE CDA	–18.1	–12.1	89.1	62.4	85.4	61.4	81.9	60.7	8108	205
MEDICINE HAT RCS	–19.5	–13.4	91.5	62.7	88.0	61.6	84.5	61.0	8497	322
RED DEER	–25.6	–18.8	82.3	63.1	78.8	61.4	75.7	60.0	10309	42
SPRINGBANK	–24.5	–17.9	80.4	60.1	76.8	58.3	73.7	57.5	10296	8
British Columbia										
ABBOTSFORD	18.1	22.9	85.6	67.4	81.9	66.0	78.5	64.5	5240	145
AGASSIZ RCS	18.7	23.1	86.2	68.5	82.8	67.3	79.6	66.1	5142	209
BALLENAS ISLAND	30.3	32.9	74.4	66.3	72.1	65.3	70.2	64.1	4772	102
COMOX	23.5	27.1	80.2	64.0	76.7	62.9	73.6	61.6	5561	102
DISCOVERY ISLAND	29.6	34.1	72.8	N/A	69.2	N/A	66.3	N/A	4995	18
ENTRANCE ISLAND	29.5	32.3	74.5	64.8	72.0	63.9	69.9	63.0	4848	104
ESQUIMALT HARBOUR	27.3	30.7	71.8	60.6	68.7	59.5	66.1	58.6	5475	11
HOWE SOUND PAM ROCKS	26.9	30.3	76.4	66.3	73.5	64.8	71.3	64.0	4769	139
KAMLOOPS	–3.2	4.0	93.3	64.8	89.5	63.7	85.5	62.3	6359	501
KELOWNA	0.8	7.3	91.6	64.7	88.1	63.5	84.4	62.1	7009	252
MALAHAT	21.6	25.6	81.8	63.2	78.2	61.9	75.1	60.8	5857	173
PENTICTON	7.8	12.7	91.2	65.7	87.8	64.5	84.5	63.2	6166	410
PITT MEADOWS CS	19.2	23.5	86.4	67.7	82.7	66.3	79.2	65.0	5382	141
POINT ATKINSON	28.7	31.9	76.2	N/A	73.8	N/A	71.9	N/A	4380	185

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
PRINCE GEORGE	–22.4	–13.7	82.5	61.6	78.7	60.0	75.1	58.3	9244	40
SANDHEADS CS	25.9	29.8	72.0	N/A	70.1	N/A	68.3	N/A	4982	53
SUMMERLAND CS	6.8	12.6	91.2	63.8	87.9	63.0	84.4	61.9	6306	458
VANCOUVER HARBOUR CS	26.8	30.5	78.5	64.2	75.9	63.0	73.5	62.0	4829	127
VANCOUVER INTL	21.0	25.8	77.2	65.2	74.5	64.2	72.3	63.0	5240	83
VERNON	2.3	8.2	91.2	65.2	87.5	64.1	83.6	62.7	6787	369
VICTORIA GONZALES CS	26.6	30.7	76.3	62.4	72.0	60.8	68.9	59.4	5170	40
VICTORIA HARTLAND CS	25.3	29.1	83.2	65.7	79.9	64.2	76.7	63.1	5142	171
VICTORIA INTL	24.7	27.9	80.1	63.8	76.3	62.5	73.3	61.1	5413	46
VICTORIA UNIVERSITY CS	27.2	30.9	80.5	64.6	77.0	63.4	73.9	62.3	4990	64
WEST VANCOUVER	21.6	26.2	80.7	65.1	77.3	64.3	74.4	63.2	5398	136
WHITE ROCK CAMPBELL SCI	22.4	26.7	76.7	65.9	73.8	64.5	71.6	63.4	5018	55
YOHO PARK	–21.8	–15.8	77.9	56.7	74.1	55.3	70.3	53.9	11693	2
Manitoba										
WINNIPEG INTL	–25.7	–21.4	86.5	70.3	83.5	68.6	80.7	66.9	10365	286
New Brunswick										
FREDERICTON INTL	–9.8	–5.1	85.9	69.8	82.5	67.9	79.6	66.2	8253	257
MONCTON INTL	–8.0	–3.5	83.6	69.7	80.5	67.6	77.8	66.1	8397	199
SAINT JOHN	–8.1	–3.1	79.2	65.8	76.3	64.2	73.5	62.6	8445	60
Newfoundland and Labrador										
ST JOHN'S INTL	4.8	8.6	76.6	66.4	73.9	65.0	71.4	63.8	8629	60
Northwest Territories										
YELLOWKNIFE	–40.8	–36.9	77.7	60.9	74.7	59.4	71.9	58.3	14687	66
Nova Scotia										
HALIFAX STANFIELD INTL	–0.8	3.0	82.2	68.7	79.1	66.9	76.3	65.4	7655	199
SHEARWATER	1.0	4.9	79.2	67.1	76.3	65.4	73.4	64.3	7529	130

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
SYDNEY	0.2	4.7	81.6	68.5	78.6	67.0	75.5	65.4	8092	158
Nunavut										
IQALUIT CLIMATE	–32.0	–29.4	64.3	54.6	58.7	52.1	54.7	49.6	17043	1
Ontario										
BEAUSOLEIL	–12.4	–5.9	85.9	74.0	82.6	71.7	79.7	70.3	7909	382
BELLE RIVER	6.5	10.6	88.6	75.3	85.7	74.0	82.8	72.4	6023	762
ERIEAU	5.3	9.3	80.3	73.2	78.5	72.0	77.0	71.1	6502	507
GUELPH TURFGRASS INSTITUTE	–6.6	–1.1	85.6	71.1	82.5	69.6	79.9	67.9	7981	275
HAMILTON INTL	–0.7	4.1	87.1	72.4	84.2	71.1	81.3	69.5	7066	449
LONDON INTL	–0.8	3.9	86.4	72.4	83.5	71.0	80.9	69.3	7097	430
NORTH BAY	–17.4	–12.0	82.3	68.1	79.4	66.5	76.8	65.1	9272	227
OTTAWA INTL	–11.5	–6.3	87.1	71.3	84.2	69.5	81.2	68.0	8070	434
PETERBOROUGH TRENT U	–8.3	–2.5	87.4	70.4	84.0	68.8	81.2	67.1	7782	319
PORT WELLER	8.9	12.6	84.4	73.0	81.7	71.7	79.2	70.5	6291	558
REGION OF WATERLOO INTL	–5.0	0.7	87.0	71.5	84.0	70.1	81.0	68.5	7634	332
SAULT STE MARIE	–12.2	–6.8	83.1	70.0	79.9	68.0	77.1	66.2	8893	162
SUDBURY	–17.8	–12.3	84.2	68.3	80.9	66.3	78.0	64.5	9385	223
THUNDER BAY CS	–19.9	–15.2	84.5	68.4	81.3	66.2	78.1	64.7	9945	145
TIMMINS	–27.7	–21.6	85.3	67.9	81.6	65.3	78.4	64.0	10768	158
TORONTO BILLY BISHOP	3.8	8.4	83.1	70.9	80.3	69.9	77.5	68.9	6627	432
TORONTO BUTTONVILLE	–3.4	1.8	88.9	72.2	85.5	70.3	82.4	68.8	7232	475
TORONTO PEARSON INTL	–0.5	4.1	88.4	72.4	85.2	70.6	82.1	69.1	6907	547
TRENTON	–7.0	–1.6	84.5	71.8	81.8	70.3	79.4	68.9	7450	375
WINDSOR	4.0	8.6	89.5	73.6	86.7	72.2	84.2	70.7	6158	788
Prince Edward Island										
CHARLOTTETOWN	–4.3	0.1	80.4	69.4	77.8	67.4	75.3	65.9	8230	194

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Québec										
BAGOTVILLE	–21.6	–16.8	84.8	67.0	81.3	65.3	78.0	64.0	10092	183
BIG TROUT LAKE	–34.0	–29.8	80.6	65.7	77.5	63.8	74.4	62.5	13229	94
JONQUIERE	–22.4	–16.9	84.2	67.6	80.8	66.1	77.6	64.8	9914	177
LA BAIE	–22.4	–17.8	84.4	67.4	80.7	66.2	77.4	64.9	10224	128
LAC SAINT-PIERRE	–11.6	–6.4	82.3	70.0	79.6	68.6	77.3	67.3	8230	359
L'ACADIE	–11.8	–7.1	86.2	70.8	83.4	69.7	80.8	68.3	7936	412
L'ASSOMPTION	–14.1	–8.6	86.8	71.4	83.8	69.6	81.0	68.1	8288	380
LENNOXVILLE	–13.7	–7.9	84.9	70.7	82.2	69.2	79.7	67.7	8253	272
MONT-JOLI	–10.7	–6.6	80.3	67.6	77.2	65.7	74.5	64.1	9511	130
MONT-ORFORD	–19.4	–13.4	77.2	65.4	74.3	64.0	71.6	62.9	10186	96
MONTREAL MCTAVISH	–7.5	–2.5	86.2	71.4	83.4	69.6	81.0	68.1	7463	547
MONTREAL MIRABEL INTL	–14.8	–9.3	85.2	71.4	82.4	69.4	79.8	67.9	8509	323
MONTREAL ST-HUBERT	–10.2	–5.4	86.1	71.8	83.4	70.0	80.9	68.6	7945	423
MONTREAL TRUDEAU INTL	–9.5	–4.6	86.0	71.7	83.3	69.9	80.9	68.4	7771	490
NICOLET	–14.1	–9.0	83.8	72.2	81.0	70.4	78.5	68.8	8436	302
POINTE-AU-PERE INRS	–7.2	–2.6	73.5	65.4	70.7	63.7	68.3	61.9	9554	20
QUEBEC CITY JEAN LESAGE INTL	–12.0	–7.4	83.7	69.5	81.2	68.3	78.4	66.5	8783	247
SAINTE-FOY U LAVAL	–11.6	–6.9	84.3	69.1	81.5	67.3	78.7	65.7	8675	270
SHERBROOKE	–17.2	–11.5	83.8	70.0	81.1	68.3	78.7	66.9	8843	185
ST-ANICET 1	–12.3	–7.0	87.0	72.8	84.3	71.2	81.7	69.6	7950	393
STE-ANNE-DE-BELLEVUE 1	–10.8	–5.5	86.0	71.2	83.3	69.8	80.7	68.3	7928	416
TROIS-RIVIERES	–11.6	–6.8	81.2	70.3	79.0	69.4	76.9	68.3	8298	332
VARENNES	–10.3	–5.7	86.5	71.2	83.5	69.6	80.9	68.1	8059	373
Saskatchewan										
MOOSE JAW CS	–22.2	–17.3	89.2	65.5	85.4	64.6	81.8	63.4	9705	209

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
PRINCE ALBERT	–32.2	–26.3	84.0	66.1	80.6	64.4	77.7	62.6	11167	120
REGINA RCS	–26.1	–21.2	87.9	67.2	83.9	65.3	80.7	63.8	10393	199
SASKATOON INTL	–29.5	–24.0	86.4	66.1	82.8	64.6	79.5	63.4	10628	166
SASKATOON KERNEN FARM	–28.3	–23.0	87.2	63.8	83.4	62.4	80.2	61.0	10626	182
Yukon Territory										
WHITEHORSE	–39.4	–30.2	78.3	57.5	74.1	56.0	70.4	54.4	12188	13
Albania										
TIRANA RINAS	26.9	29.6	93.6	73.2	91.6	73.9	89.5	73.7	2757	1236
Algeria										
CONSTANTINE BOUDIAF INTL	31.1	32.9	102.0	68.4	98.5	68.3	95.1	67.7	2984	1540
DAR EL BEIDA	35.2	37.3	95.3	72.3	92.4	72.5	89.7	72.5	1772	1614
ORAN INTL	35.9	38.8	93.7	69.9	90.5	70.4	87.9	70.7	1621	1643
Argentina										
CORDOBA INTL	31.6	34.4	94.8	70.7	91.6	70.3	89.2	69.8	1737	1350
CORRIENTES INTL	39.6	42.6	98.5	76.0	96.4	76.2	93.6	75.6	712	2943
EL PLUMERILLO	30.7	33.6	96.4	67.4	93.5	67.3	91.3	66.9	2191	1653
EZEIZA INTL	31.8	34.2	92.9	72.6	89.9	71.8	87.6	71.1	2126	1202
JORGE NEWBERY INTL	39.5	42.4	88.1	73.8	85.9	73.4	83.9	72.4	1604	1369
MAR DEL PLATA	29.8	32.0	87.9	70.1	84.4	68.8	80.9	67.8	3343	433
PARANA	36.5	38.9	94.1	73.7	91.5	72.7	89.2	72.0	1500	1646
POSADAS INTL	40.8	43.7	97.0	75.2	95.1	75.1	93.3	74.9	572	3190
RESISTENCIA INTL	35.2	38.8	98.9	75.4	96.7	75.8	94.4	75.6	837	2875
ROSARIO	30.6	33.7	93.6	73.8	91.4	73.0	89.2	72.3	1821	1450
SALTA AP	30.0	32.4	91.6	65.2	89.0	65.6	86.1	65.6	1671	1034
SAN JUAN	28.1	30.9	100.4	67.5	97.9	67.3	95.3	66.6	2095	2082
SAN MIGUEL DE TUCUMAN	37.6	40.6	97.3	74.0	94.9	74.0	92.1	73.5	1011	2270

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
SANTIAGO DEL ESTERO	30.5	34.5	102.5	74.3	99.5	73.9	96.7	73.4	1057	2697
SAUCE VIEJO	32.8	36.0	95.2	75.6	92.5	74.5	89.9	73.5	1456	1881
Armenia										
YEREVAN EREBUNI	8.4	12.7	97.0	70.7	94.7	69.8	91.8	68.7	4932	1388
Australia										
ADELAIDE INTL	39.0	41.0	97.2	65.0	93.1	64.3	89.2	63.4	2080	861
BRISBANE ARCHERFIELD	41.7	44.0	91.2	73.1	88.6	72.6	86.4	71.9	649	1923
BRISBANE INTL	42.5	45.0	87.6	72.7	85.5	73.3	84.0	72.4	600	1825
CANBERRA	26.1	28.2	93.1	64.1	89.3	63.2	85.5	62.3	3687	491
CANTERBURY RACECOURSE	38.8	40.6	90.5	68.0	86.5	68.4	83.3	68.0	1610	932
GOLD COAST	42.9	46.0	84.6	74.0	83.3	73.6	82.2	72.9	570	1697
GOLD COAST SEAWAY	49.3	51.4	87.0	73.3	84.7	72.9	83.0	72.5	335	2004
JANDAKOT AP	35.3	37.7	97.6	67.9	94.4	67.4	91.0	66.6	1692	1254
KENT TOWN	40.5	42.4	99.8	66.2	95.4	65.4	91.4	64.3	1916	1087
LAVERTON RAAF	35.4	37.4	95.0	66.1	89.6	65.2	84.5	64.3	2981	429
MELBOURNE INTL	37.2	39.0	95.1	64.7	90.1	63.8	85.8	63.3	2993	483
MELBOURNE MOORABBIN	36.8	39.1	93.6	66.7	88.8	65.6	84.0	65.0	2856	418
MELBOURNE REGIONAL OFFICE	40.5	42.4	94.8	65.9	90.2	65.0	85.6	64.3	2264	632
MOUNT LOFTY	36.4	37.7	88.0	60.9	84.3	59.4	80.9	58.1	4598	331
NEWCASTLE NOBBYS SIGNAL	45.9	47.6	86.7	67.4	81.7	67.4	78.4	68.7	1065	1020
PERTH INTL	38.9	41.1	99.0	66.9	95.7	66.7	92.5	66.2	1390	1462
PERTH METRO	38.9	41.2	97.4	68.6	94.0	67.8	90.7	67.1	1327	1409
SCORESBY RESEARCH INSTITUTE	36.1	38.1	93.4	66.8	89.3	66.1	85.3	65.4	2944	488
SWANBOURNE	43.6	45.5	95.3	68.2	91.2	67.8	87.6	67.4	1127	1284
SYDNEY BANKSTOWN	38.0	40.0	92.8	69.1	88.6	68.8	85.0	68.2	1639	1007
SYDNEY INTL	43.5	45.0	91.3	66.8	86.4	68.2	83.3	68.0	1205	1192

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
SYDNEY OBSERVATORY HILL	45.1	46.5	88.1	67.7	84.0	68.4	81.2	68.5	1137	1109
SYDNEY OLYMPIC PARK ARCHERY	42.2	43.9	92.4	67.7	88.4	67.6	84.9	67.4	1324	1168
TUGGERANONG ISABELLA PLAINS	25.4	27.4	93.1	64.7	89.1	63.9	85.4	62.8	3710	512
WILLIAMTOWN	39.1	41.2	93.9	69.3	89.4	69.3	85.6	68.8	1463	1063
Austria										
GUMPOLDSKIRCHEN	14.6	18.5	88.3	70.8	84.9	69.0	81.8	67.4	5349	488
TULLN	11.8	16.9	89.0	70.8	85.6	69.1	82.5	67.5	5577	408
WIEN HOHE WARTE	15.2	19.2	88.1	71.4	84.8	69.5	81.8	67.9	5305	487
WIEN INNERE STADT	17.6	21.3	89.2	71.9	86.1	70.3	83.1	68.7	4830	695
WIEN SCHWECHAT	13.4	17.4	88.2	69.2	85.0	68.0	82.1	66.7	5509	437
Belarus										
BREST	–0.2	6.4	86.4	67.9	82.9	66.3	79.7	64.7	6742	273
GOMEL	–5.0	1.2	87.1	68.3	83.8	67.1	80.6	65.5	7366	322
GRODNO	–2.9	3.4	83.9	67.7	80.3	65.9	77.2	64.1	7383	165
MINSK 1	–3.2	2.8	84.5	67.5	81.5	65.8	78.6	64.4	7667	208
MOGILEV	–7.2	–1.4	84.0	67.7	80.5	66.1	77.3	64.8	8037	174
VITEBSK	–6.8	–0.9	83.4	67.4	80.2	66.0	77.0	64.2	7982	195
Belgium										
ANTWERP INTL	20.3	24.1	84.8	69.0	81.0	67.3	77.7	65.5	4988	196
BRUSSELS NATL	20.3	24.0	84.4	68.2	80.7	66.9	77.2	65.1	5148	177
UCCLE	21.0	24.5	84.1	67.6	80.5	66.1	77.1	64.3	5093	203
Benin										
COTONOU CAJEHOUN	71.6	73.0	91.3	80.7	90.0	80.8	89.7	80.8	0	6156
Bolivia										
COCHABAMBA AP	35.5	37.6	86.1	59.1	84.3	58.5	82.6	58.1	919	505
EL ALTO INTL	23.1	25.0	64.1	42.7	62.5	42.4	61.0	42.2	7014	0

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
VIRU VIRU INTL	48.6	51.6	94.9	74.5	93.1	74.9	91.4	75.2	157	3957
Bosnia and Herzegovina										
BJELASNICA	–1.7	2.7	67.1	53.5	64.3	52.5	61.9	51.9	10823	2
SARAJEVO	8.7	13.9	91.5	67.4	88.0	67.4	84.5	66.5	5599	426
SARAJEVO-BJELAVE	11.6	15.9	91.3	67.3	87.8	66.6	84.4	65.5	5358	519
Brazil										
AFONSO PENA AP	37.1	40.9	87.6	68.2	85.7	68.4	83.8	68.4	1129	1066
ANAPOLIS AB	55.8	57.6	91.2	65.9	89.3	66.6	87.5	67.3	14	2937
ARACAJU AP	69.8	71.4	89.8	79.9	89.2	79.6	88.0	79.2	0	5548
BELEM VAL DE CANS INTL	73.0	73.2	91.8	78.7	91.4	78.5	90.0	78.4	0	6120
BELO HORIZONTE AP	52.0	54.0	91.3	68.3	89.5	68.3	87.8	68.3	39	2839
BRASILIA INTL	51.0	53.2	90.0	63.6	88.2	64.3	87.0	64.7	24	2566
CAMPO GRANDE AB	46.5	50.4	97.2	72.5	95.4	72.9	93.6	73.2	115	4557
CONFINS INTL	51.4	53.3	89.6	68.1	87.8	68.5	86.0	68.6	107	2237
CUIABA INTL	55.4	58.9	100.7	71.8	98.9	72.0	97.2	72.7	22	6018
EDUARDO GOMES INTL	71.2	71.4	96.5	79.3	95.0	78.8	93.5	78.7	0	6123
FLORIANOPOLIS INTL	46.1	49.4	89.8	77.4	87.8	77.0	85.8	76.0	388	2343
FORTALEZA INTL	73.0	73.3	89.8	76.9	89.4	76.7	88.2	76.2	0	6054
GOIANIA AP	54.5	56.9	95.4	67.5	93.5	68.3	91.8	68.9	5	4260
LONDRINA AP	46.3	49.8	93.2	71.1	91.3	71.4	89.6	71.7	215	2929
MACAPA INTL	73.0	73.2	95.2	79.8	94.8	79.8	93.4	79.9	0	6490
MACEIO INTL	65.8	66.4	91.5	77.9	89.9	77.2	89.4	77.0	0	4836
MANAUS AB	72.2	73.1	94.8	78.5	93.4	78.5	92.1	78.4	0	6219
NATAL AB	69.5	70.0	91.1	77.9	89.9	77.4	89.3	77.3	0	5606
PORTO ALEGRE INTL	39.0	42.4	94.8	76.5	91.6	75.5	89.4	74.6	879	2051
PORTO VELHO INTL	65.8	68.0	96.8	76.0	95.2	76.4	93.6	77.0	1	6018

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	**99.6%**	**99%**	**DB**	**MCWB**	**DB**	**MCWB**	**DB**	**MCWB**	**HDD**	**CDD 65**
RECIFE INTL	70.4	71.5	93.2	80.7	91.8	79.7	91.2	79.4	0	6068
RIO GALEAO TOM JOBIN INTL	58.8	60.5	98.5	77.1	95.4	76.6	93.3	76.6	11	4330
RIO SANTOS DUMONT AP	61.1	62.7	93.3	77.2	91.1	76.9	89.3	76.6	7	4082
SALVADOR INTL	69.4	70.1	90.2	80.0	89.6	79.7	88.2	79.1	0	5433
SAO LUIS INTL	73.1	73.5	93.0	79.0	91.7	78.4	91.1	78.1	0	6486
SAO PAULO CONGONHAS AP	48.2	50.4	89.9	68.3	88.0	68.4	86.2	68.3	382	2093
SAO PAULO INTL	46.1	48.4	91.0	70.7	88.2	70.4	86.4	70.1	396	1916
TERESINA AP	71.4	72.7	101.9	73.0	100.4	73.8	98.9	74.3	0	7254
VIRACOPOS INTL	48.1	50.4	91.7	69.6	90.0	69.8	88.2	69.8	178	2557
VITORIA AP	62.2	64.0	93.4	77.9	91.8	77.3	90.6	77.1	0	4697
Bulgaria										
CHERNI VRAH	–2.4	1.1	63.2	51.7	60.6	51.0	58.4	50.2	11500	0
PLOVDIV	13.6	18.5	94.7	69.3	91.5	69.1	89.2	68.2	4556	1007
SOFIA	8.9	14.1	91.4	66.2	87.8	65.6	84.5	65.2	5476	536
VARNA	15.6	19.5	89.4	72.1	86.3	71.7	84.2	70.9	4567	782
Burkina Faso										
BOBO DIOULASSO	65.2	67.3	100.7	68.5	99.3	68.6	98.0	68.8	0	6224
OUAGADOUGOU INTL	61.1	63.2	105.5	68.5	103.9	68.7	102.3	69.1	0	6961
Chad										
N'DJAMENA INTL	55.6	58.6	109.4	71.5	107.7	71.2	105.9	70.8	1	7005
Chile										
CERRO MORENO INTL	49.9	51.6	75.5	66.0	74.4	65.2	73.4	64.5	1281	315
SANTIAGO PUDAHUEL INTL	30.1	32.1	89.4	63.0	87.5	62.8	85.7	62.6	2675	475
China										
ANQING	28.2	30.4	96.2	80.9	94.4	80.7	92.4	80.1	2835	2390
ANYANG	16.3	19.7	95.7	73.8	93.2	75.0	90.8	75.0	4243	1787

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
BAITA	–9.6	–4.2	89.7	63.3	87.0	62.8	84.3	62.2	7970	611
BAODING	13.5	16.8	95.5	72.7	92.7	73.2	90.3	73.4	4738	1727
BAOJI	21.1	23.7	95.1	71.1	92.6	70.6	90.0	70.5	4216	1468
BEIJING CAPITAL INTL	11.9	15.0	95.0	71.6	92.3	72.0	89.8	72.2	5121	1567
BENGBU	22.9	25.7	96.1	79.7	93.9	79.0	91.5	77.7	3424	2062
BENXI	–8.2	–4.2	88.3	71.3	86.1	71.1	84.1	70.7	7363	893
CANGZHOU	15.1	18.1	93.6	73.7	91.4	74.3	89.4	74.1	4769	1668
CHANGCHUN LONGJIA	–14.0	–9.6	87.7	69.2	85.4	69.6	83.0	69.1	8701	759
CHANGDE	30.1	32.1	97.6	80.1	95.5	80.0	93.3	79.5	2656	2403
CHANGSHA	29.9	31.9	97.5	78.9	95.6	78.9	93.6	78.7	2628	2500
CHAOYANG	–2.3	1.6	92.6	70.1	89.9	70.1	87.5	69.7	6640	1158
CHENGDE	–0.8	2.5	91.6	68.8	88.8	68.8	86.3	68.3	6860	936
CHENGDU SHUANGLIU	33.4	35.4	93.0	77.0	91.0	76.2	88.2	74.9	2434	1833
CHONGQING JIANGBEI INTL	37.0	38.9	99.0	76.9	96.6	77.2	94.2	77.0	2108	2341
DALIAN INTL	10.0	13.3	88.2	74.2	86.1	73.3	84.1	72.7	5549	1154
DANDONG	2.9	6.6	85.9	74.4	83.5	73.4	81.4	72.6	6498	824
DATONG	–5.3	–1.8	89.6	63.5	86.7	62.8	84.2	62.6	7561	641
DEZHOU	17.1	19.7	93.6	75.7	91.3	75.8	89.4	75.2	4492	1738
FUZHOU CHANGLE INTL	40.3	42.4	96.0	80.0	93.8	79.7	91.8	79.3	1265	2953
GANYU	19.3	22.2	91.9	79.5	89.3	78.4	87.0	77.9	4150	1544
GUANGZHOU BAIYUN INTL	42.4	44.4	96.6	79.2	94.8	79.1	93.1	78.9	693	3790
GUIYANG LONGDONGBAO INTL	26.4	28.8	86.6	70.1	84.7	69.9	82.9	69.4	3122	1180
HAIKOU	51.3	54.1	95.1	80.3	93.6	80.1	92.2	79.9	187	4551
HANGZHOU XIAOSHAN INTL	28.2	30.3	97.9	79.4	95.9	79.4	93.5	79.1	2810	2328
HARBIN	–18.0	–13.9	88.5	68.7	85.9	69.7	83.4	69.1	9392	720
HEFEI LUOGANG	24.4	26.8	96.1	81.9	93.8	81.2	91.7	80.2	3279	2148

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
JINAN TSINAN	16.9	20.0	95.1	73.8	92.8	74.0	90.6	73.6	4078	1942
JINGDEZHEN	29.6	31.8	97.2	79.6	95.4	79.3	93.6	78.8	2386	2567
JINGZHOU JIANGLING	28.9	30.9	95.1	81.8	93.3	81.0	91.5	79.9	2838	2234
JINZHOU	3.3	6.7	89.2	71.2	86.9	71.1	84.8	70.7	6289	1112
JIXI	–12.7	–9.2	86.8	69.1	84.1	68.6	81.5	68.1	9406	513
KUNMING WUJIABA	33.2	35.5	82.4	61.5	80.5	61.5	78.8	61.8	2072	641
LANZHOU	11.6	14.3	90.7	64.5	87.9	63.3	85.4	62.3	5516	811
LIANGJIANG	33.8	36.0	95.2	77.8	93.4	77.6	91.6	77.5	1854	2706
LINGXIAN	12.9	16.4	95.1	74.4	92.4	75.1	90.0	75.1	4642	1657
LIUZHOU	38.1	40.5	95.5	78.1	94.0	78.1	92.6	77.9	1262	3400
MENGJIN	20.1	22.7	94.9	71.6	92.2	72.1	89.7	72.3	3948	1676
MUDANJIANG	–15.7	–11.8	88.3	70.0	85.7	68.9	83.0	68.3	9268	621
NANCHANG CHANGBEI INTL	30.7	32.7	96.6	80.0	94.9	79.8	93.0	79.6	2449	2636
NANJING LUKOU	23.4	26.3	96.2	80.1	93.9	79.8	91.6	79.1	3340	2039
NANNING WUXU INTL	40.7	43.0	95.1	79.1	93.4	79.0	91.8	78.7	887	3527
NEIJIANG	36.5	38.5	95.6	78.7	93.3	78.0	91.0	77.1	2193	2088
QINGDAO INTL	17.2	19.8	91.0	74.9	88.0	74.6	85.9	73.8	4489	1440
QINGJIANG	21.1	24.7	92.8	80.8	90.8	79.7	88.8	78.3	3767	1735
QIQIHAR SANJIAZI	–18.6	–14.7	89.5	69.3	86.3	68.6	83.8	68.1	9714	732
SHANGHAI BAOSHAN	28.3	30.7	95.5	79.9	93.3	79.6	91.0	79.2	2875	2148
SHANGHAI HONGQIAO INTL	26.7	29.8	96.7	80.8	93.6	80.7	91.7	80.0	2880	2255
SHANTOU	45.0	47.7	93.5	80.8	91.7	80.6	90.0	80.1	606	3393
SHAOGUAN	35.9	38.2	95.6	78.4	93.9	78.2	92.3	77.9	1393	3113
SHENGYANG TAOXIAN	–11.4	–7.4	89.5	73.2	87.4	73.2	85.6	72.2	7468	1012
SHENYANG	–9.5	–5.2	88.6	73.5	86.6	72.6	84.7	71.9	7402	982
SHENZHEN BAO'AN INTL	44.7	47.3	93.2	79.4	91.6	79.3	90.4	79.2	453	3994

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
SHIJIAZHUANG	17.1	19.9	96.9	71.4	94.0	72.9	91.4	73.2	4361	1876
SIPING	–10.1	–6.4	87.9	70.5	85.6	70.4	83.7	70.0	8044	864
TAI SHAN	1.7	5.5	72.8	63.0	71.0	63.3	69.7	63.8	8047	82
TAIYUAN WUSU INTL	5.1	8.8	92.4	67.6	89.7	67.8	87.6	67.3	5744	1006
TANGSHAN	7.8	11.4	92.2	73.5	89.8	73.4	87.7	72.9	5347	1446
TIANJIN	13.0	16.1	93.9	73.7	91.5	73.9	89.2	73.4	4910	1681
TIANJIN BINHAI INTL	12.2	15.6	93.7	73.7	91.6	73.5	89.6	73.4	4966	1657
URUMQI DIWOPU	–11.4	–7.5	95.4	64.3	93.2	63.9	91.0	63.3	7739	1394
WEIFANG	12.8	15.7	93.9	75.5	91.3	75.2	88.9	74.5	4847	1466
WENZHOU	34.3	37.0	93.0	81.2	91.2	80.7	89.6	80.0	1937	2350
WUHAN TIANHE	27.5	29.9	97.0	81.7	95.1	81.1	93.1	80.6	2878	2389
WUHUXIAN	26.4	28.7	97.2	81.3	95.1	80.7	92.8	80.1	3058	2208
WULUMUQI	–7.7	–3.9	92.1	61.4	89.3	60.9	86.8	60.4	7823	977
XIAMEN GAOQI INTL	44.2	46.1	94.6	79.3	92.8	79.3	91.1	78.9	813	3188
XIANYANG	17.3	20.8	97.1	73.5	94.9	73.2	91.8	72.9	4245	1630
XIHUA	21.9	24.5	95.3	78.0	93.1	78.4	90.9	77.3	3720	1847
XINGTAI	18.9	21.4	96.7	72.1	93.8	73.0	91.5	73.5	4188	1911
XINING	0.9	4.1	81.7	59.3	78.8	57.7	76.1	56.5	7559	88
XINYANG	23.7	26.1	94.6	79.5	92.4	78.5	90.3	77.5	3403	1926
XINZHENG INTL	20.4	22.9	95.6	74.2	93.3	75.0	91.0	75.0	3883	1802
XUZHOU	20.5	23.5	94.7	78.5	92.5	77.6	90.2	76.6	3829	1870
YANGJIANG	44.6	46.9	91.5	79.7	90.0	79.4	88.8	79.2	503	3666
YANJI	–8.1	–4.4	87.9	70.7	85.0	69.5	82.2	68.5	8523	533
YICHANG	30.2	32.2	96.5	79.9	94.2	78.9	91.9	77.8	2649	2204
YINCHUAN	2.5	6.4	90.9	65.9	88.6	65.4	86.2	64.5	6209	921
YINGKOU	0.3	4.0	86.8	75.2	84.9	74.3	83.3	73.6	6532	1107

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
YUEYANG	29.9	32.0	94.0	80.9	92.6	80.3	91.2	79.8	2638	2402
YUNCHENG	16.3	19.8	97.6	72.3	95.1	72.4	92.6	72.0	4178	1881
ZHANGJIAKOU	1.5	4.7	91.5	65.7	88.6	65.5	86.0	65.2	6673	981
ZHANJIANG	45.7	48.2	92.9	79.8	91.6	80.0	90.3	79.9	398	4012
ZHAOQING GAOYAO	43.4	45.5	95.2	79.0	93.7	78.9	92.2	78.8	686	3756
ZUNYI	30.7	32.7	90.8	72.8	88.9	72.4	87.0	72.1	2943	1540
Colombia										
ALFONSO BONILLA ARAGON INTL	64.0	64.4	89.9	71.7	89.1	71.6	87.7	71.5	0	3952
ELDORADO INTL	38.4	40.9	70.2	56.1	69.5	56.0	68.2	55.8	2999	0
ERNESTO CORTIZZOS INTL	73.0	73.6	93.6	81.1	92.8	81.0	91.5	80.5	0	6611
JOSE MARIA CORDOVA INTL	50.1	51.7	75.0	60.7	73.7	60.2	73.4	60.1	705	43
RAFAEL NUNEZ AP	73.8	75.0	91.5	81.7	90.0	81.0	89.7	80.8	0	6535
Congo										
BRAZZAVILLE MAYA MAYA INTL	64.4	66.1	93.5	76.2	92.4	76.2	91.3	76.1	0	5124
Costa Rica										
JUAN SANTAMARIA INTL	62.3	63.1	87.5	69.4	86.0	68.8	84.5	68.8	0	3305
Côte d'Ivoire										
ABIDJAN	70.2	71.6	91.2	81.5	89.9	80.9	89.3	80.7	0	5838
Croatia										
ZAGREB MAKSIMIR	14.4	19.3	90.7	70.7	87.6	69.6	84.8	68.6	4881	633
ZAGREB PLESO	12.1	17.2	91.0	71.3	87.9	70.2	85.1	69.3	5086	570
Cuba										
CAMAGUEY INTL	58.8	61.2	92.8	74.5	91.5	74.8	90.4	74.8	8	4820
HAVANA JOSE MARTI INTL	50.2	53.6	91.6	77.0	90.9	77.0	89.7	77.0	54	4146
SANTIAGO ANTONIO MACEO INTL	65.4	67.1	89.7	77.6	88.6	77.7	88.0	77.7	0	5157
Czech Republic										

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
BRNO-TURANY	10.3	14.3	87.5	68.4	84.1	67.0	80.9	65.6	6067	345
OSTRAVA MOSNOV	5.1	10.6	86.8	68.4	83.2	67.1	80.0	65.5	6392	228
PRAHA-KBELY	9.3	14.0	85.5	67.1	82.1	66.1	79.1	64.8	6132	257
PRAHA-LIBUS	10.4	14.8	87.2	66.6	83.6	65.5	80.3	64.1	6080	279
PRAHA-RUZYNE	8.5	13.4	85.3	66.6	81.8	65.3	78.4	64.0	6540	187
Denmark										
DROGDEN FYR	22.5	24.8	72.1	65.1	70.1	64.0	68.4	63.0	6130	54
KOEBENHAVNS KASTRUP	19.6	22.6	78.0	64.9	75.3	63.9	72.7	62.7	6324	98
ROSKILDE	15.6	19.4	78.6	64.9	75.3	63.8	72.8	62.7	6681	60
VAERLOSE	14.4	19.2	79.9	65.1	76.8	64.3	73.4	62.9	6775	72
Dominican Republic										
LAS AMERICAS INTL	64.5	66.1	91.5	79.5	90.0	79.3	89.6	79.2	0	5212
SANTO DOMINGO	67.6	68.9	90.7	80.9	89.7	80.8	89.0	80.5	0	5503
Ecuador										
JOSE JOAQUIN DE OLMEDO INTL	66.0	66.6	91.3	75.1	89.8	75.2	89.2	75.1	0	4999
QUITO PARQUE BICENTENARIO	44.2	45.1	71.4	53.3	70.0	53.4	69.4	53.4	2535	0
Egypt										
ALEXANDRIA INTL	44.6	46.4	91.8	72.3	89.4	74.0	87.6	74.7	814	2433
ASSIUT	40.5	42.4	106.0	69.0	103.6	68.9	101.0	68.3	853	3806
CAIRO INTL	46.2	48.1	100.7	70.2	98.4	70.7	96.2	70.9	619	3397
LUXOR INTL	42.4	44.5	109.8	73.1	107.9	72.8	106.1	72.3	476	5117
PORT SAID	49.6	51.5	89.9	77.6	88.2	77.7	87.6	77.6	497	2914
PORT SAID EL GAMIL	49.3	51.2	89.4	77.8	87.9	77.6	86.7	76.9	554	2751
Estonia										
TALLINN	–1.3	4.8	80.0	66.4	76.6	64.4	73.3	62.7	8206	74
Finland										

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
HELSINKI VANTAA	–6.2	–0.4	80.7	64.7	77.4	62.9	74.8	61.6	8458	95
ISOSAARI	0.6	6.8	72.9	66.1	70.8	64.9	68.7	63.5	8073	55
France										
CAP COURONNE	26.8	32.7	87.3	72.7	85.0	72.2	83.0	71.3	2841	1021
CAP FERRAT	39.3	41.4	84.4	72.3	82.6	72.3	80.9	71.8	2286	1008
CAP POMEGUES	29.4	35.2	83.5	71.6	81.1	71.5	79.3	70.7	2741	823
LYON-BRON	22.0	25.0	91.7	67.7	88.2	67.2	85.0	66.6	4269	663
LYON-SAINT EXUPERY	21.3	24.4	90.6	67.9	87.3	67.4	84.0	66.6	4435	597
MARSEILLE PROVENCE	27.5	30.0	91.1	69.9	88.7	69.5	86.4	68.9	2908	1159
NICE COTE D'AZUR	35.5	37.5	85.1	72.7	83.3	72.4	81.8	71.9	2490	984
PARIS CHARLES DE GAULLE	22.8	25.9	87.5	67.8	83.6	66.4	80.0	65.2	4599	309
PARIS LE BOURGET	23.6	26.5	87.9	67.7	84.0	66.4	80.4	65.2	4484	303
PARIS ORLY	23.0	26.3	87.9	68.0	84.2	66.9	80.7	65.4	4601	329
PARIS-MONTSOURIS	26.4	28.7	88.1	68.2	84.3	67.1	80.8	65.4	4127	423
TOULOUSE BLAGNAC	25.2	28.1	91.7	69.2	88.2	68.4	85.3	67.5	3638	719
TRAPPES	23.2	26.1	86.3	67.3	82.4	65.9	79.0	64.6	4802	249
VELIZY-VILLACOUBLAY	23.2	26.2	86.0	67.2	82.4	66.0	78.9	64.6	4860	279
Gabon										
LIBREVILLE INTL	71.4	72.3	89.2	81.3	88.1	80.9	87.6	80.7	0	5425
Gambia										
BANJUL INTL	61.9	63.1	100.0	68.5	96.9	68.4	94.9	69.8	0	5654
Georgia										
TBILISI	21.2	24.6	94.9	71.1	92.3	70.5	89.6	70.1	4114	1283
Germany										
BERLIN DAHLEM	10.4	15.6	84.7	66.3	81.1	64.7	78.1	63.4	6102	213
BERLIN SCHONEFELD	7.6	13.1	85.9	66.5	82.3	64.8	78.9	63.5	6273	201

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
BERLIN TEGEL	9.7	15.4	86.1	65.6	82.5	64.4	79.2	62.8	5970	264
BERLIN TEMPELHOF	10.8	15.4	86.0	66.1	82.3	64.7	79.1	63.4	5910	265
BREMEN	15.1	18.8	84.1	67.0	80.4	65.7	77.0	64.0	6048	157
CELLE	13.4	17.8	86.4	66.3	82.8	65.0	79.3	63.6	5807	229
DRESDEN	8.1	13.5	85.7	65.9	81.9	64.8	78.7	63.5	6094	243
DUSSELDORF	14.1	19.7	85.3	67.3	82.1	65.6	78.9	64.2	5271	251
ESSEN MULHEIM	14.3	19.5	82.8	66.7	79.8	65.2	76.7	63.7	5720	186
FRANKFURT AM MAIN	16.2	20.4	88.5	66.4	84.8	65.6	81.6	64.6	5447	350
FURSTENFELDBRUCK	4.8	10.2	84.3	66.0	80.7	64.6	77.4	63.0	6670	147
GUTERSLOH	16.2	20.9	86.4	66.6	82.4	65.5	78.9	64.3	5523	219
HAMBURG FUHLSBUTTEL	11.1	16.0	82.1	66.1	78.7	64.5	75.3	62.9	6324	110
HANNOVER	9.2	14.5	84.0	66.9	80.6	65.3	77.3	63.7	6063	144
HEIDELBERG	17.3	21.5	89.9	68.8	86.3	67.5	82.8	65.7	4897	497
ITZEHOE	14.7	18.8	83.2	65.7	79.0	65.0	75.5	63.9	6274	112
KOLN BONN	17.1	20.8	86.5	67.3	82.7	65.9	79.4	64.4	5495	211
LEIPZIG HALLE	8.0	13.4	85.7	66.5	81.9	65.3	78.7	63.8	6108	216
LEIPZIG-HOLZHAUSEN	12.2	17.2	86.5	66.8	83.0	65.2	79.6	64.2	5704	276
MUNICH	8.6	13.9	85.9	66.5	82.2	65.5	79.0	64.2	6285	195
NORVENICH	18.1	22.4	87.2	67.1	83.1	66.0	79.5	64.4	5208	235
NURNBERG	6.1	12.3	86.3	65.2	82.9	64.2	79.7	62.6	6311	231
POTSDAM	9.2	14.0	85.5	66.0	81.8	64.9	78.7	63.8	6186	214
QUICKBORN	14.5	18.7	82.9	66.0	79.2	65.2	75.8	63.8	6229	101
ROTH	8.5	13.8	87.8	66.8	84.1	65.2	80.7	64.0	6373	207
STUTTGART FILDERSTADT	9.1	14.1	84.8	66.0	81.7	65.1	78.5	63.9	6281	191
STUTTGART SCHNARREN	11.3	15.8	85.2	67.3	82.0	65.5	79.0	64.3	5674	288
WUNSTORF	15.0	19.1	86.4	66.5	82.5	65.4	79.0	63.8	5615	229

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Greece										
ATHINAI HELLINIKON	35.2	37.9	96.4	70.0	93.5	69.8	91.5	70.0	1976	2073
ELEFSIS	33.5	35.8	98.4	69.1	95.3	68.8	93.3	68.2	2190	2133
THESSALONIKI MAKEDONIA	26.5	29.7	94.6	71.0	91.6	71.0	89.5	70.2	3173	1537
Guatemala										
LA AURORA INTL	51.6	53.5	82.5	64.0	80.8	64.1	79.2	64.1	111	1271
Honduras										
RAMON VILLEDA MORALES INTL	64.0	65.8	98.4	78.3	96.4	78.7	94.8	78.8	0	5770
TONCONTIN INTL	53.2	55.3	89.7	66.7	87.9	67.4	86.2	67.4	19	2718
Hong Kong										
HONG KONG INTL	47.9	50.3	93.2	79.7	91.7	79.3	91.0	79.1	324	4222
HONG KONG OBSERVATORY	49.3	51.6	90.0	79.7	89.0	79.6	88.1	79.4	426	3556
Hungary										
BUDAORS	11.8	15.8	87.8	68.2	84.8	67.4	82.0	66.5	5530	443
BUDAPEST FERIHEGY	10.6	15.5	91.5	71.5	88.0	69.5	84.5	67.9	5589	515
BUDAPEST PESTSZENTLORINC	14.3	17.9	91.7	68.8	88.3	67.7	85.2	66.5	5213	670
India										
AHMEDABAD	51.5	53.7	108.5	73.4	106.1	73.1	104.1	73.2	20	6294
AKOLA	53.9	56.4	109.7	71.5	107.7	71.0	105.8	70.9	2	6429
AURANGABAD	51.4	53.9	104.2	73.1	102.5	72.9	100.7	72.3	10	5044
BANGALURU	59.5	60.9	93.6	67.9	92.2	67.7	90.7	67.7	0	3888
BELGAUM	55.4	57.6	97.3	66.6	95.5	66.7	93.9	67.0	0	4004
BHOPAL	49.2	51.5	107.2	71.4	105.1	70.9	102.9	70.7	110	4932
BHUBANESHWAR	57.1	59.1	102.3	80.8	99.9	80.5	97.7	80.1	1	6084
BIKANER	42.9	45.4	111.5	70.6	109.2	71.5	106.9	72.3	319	6236
CHENNAI INTL	67.8	69.4	102.0	78.6	99.3	78.5	97.2	78.4	0	6894

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
COIMBATORE INTL	64.9	66.5	97.3	71.6	96.0	72.2	94.4	72.7	0	5651
CWC VISHAKHAPATNAM WALTAIR	68.0	69.4	92.9	80.9	91.6	81.6	90.7	81.4	0	6207
DEVI AHILYA BAI HOLKAR INTL	48.9	51.4	105.2	68.0	103.2	68.0	101.1	67.9	80	4750
GUWAHATI INTL	51.4	53.2	94.7	80.8	93.1	80.7	91.7	80.5	102	4370
GWALIOR	42.6	44.6	110.5	73.7	108.4	73.8	106.1	73.7	345	5445
HYDERABAD BEGUMPET AP	56.5	58.8	105.0	71.1	102.6	71.1	100.6	71.1	1	5586
JABALPUR	47.0	49.3	108.4	69.3	106.4	69.2	104.1	69.6	152	5126
JAIPUR	44.8	47.2	108.4	70.4	106.1	70.4	103.9	70.5	298	5370
JAMSHEDPUR	50.1	52.3	107.9	72.6	105.2	73.0	102.3	73.7	44	5569
JODHPUR	47.8	49.9	108.7	70.3	106.3	71.0	104.2	71.4	136	6083
KOLKATA BOSE INTL	52.3	54.3	100.0	81.0	98.0	81.0	96.3	80.6	36	5612
KOZHIKODE	72.6	73.6	94.0	82.4	93.0	81.8	92.1	81.3	0	6450
LUCKNOW	44.1	46.3	107.9	73.6	105.7	73.5	102.6	74.7	347	5055
MANGALURU INTL	69.4	70.8	93.9	76.8	92.8	76.9	91.9	76.6	0	6006
MUMBAI SHIVAJI INTL	62.3	64.6	96.6	72.9	94.8	73.4	93.2	73.9	0	6257
NAGPUR AMBEDKAR INTL	52.5	55.1	111.5	72.7	109.5	72.6	107.4	72.2	10	5972
NELLORE	68.9	70.2	105.3	80.5	102.8	80.7	100.6	80.4	0	7386
NEW DELHI INDIRA GANDHI INTL	42.4	44.4	109.8	72.3	107.7	72.2	105.4	72.3	526	5347
NEW DELHI SAFDARJUNG	42.8	44.7	108.2	73.3	105.5	73.8	103.1	74.0	480	5082
PATIALA	41.0	43.0	107.2	77.2	104.6	76.8	101.6	76.8	721	4414
PATNA	46.0	48.1	106.1	73.5	103.7	73.7	100.6	75.0	261	5206
PUNE INTL	49.6	51.7	100.8	67.8	99.0	67.6	97.2	67.7	13	4275
RAJKOT	53.3	55.8	106.0	72.2	104.0	72.0	102.0	72.9	11	6211
SOLAPUR	59.8	62.3	105.9	72.6	104.1	73.0	102.2	72.5	0	6325
SURAT	57.4	59.7	100.2	72.6	97.6	73.2	95.4	73.6	1	6165
THIRUVANANTHAPURAM	71.9	73.0	93.1	78.6	92.1	78.4	91.2	78.2	0	6160

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
TIRUCHIRAPPALLI	68.0	69.5	102.4	78.7	100.8	78.4	99.9	78.2	0	7289
Indonesia										
BALI NGURAH RAI INTL	71.3	73.0	90.6	79.9	89.7	79.6	89.1	79.5	0	6087
HASANUDDIN INTL	68.3	69.9	93.7	75.3	92.9	75.7	91.7	76.4	0	5948
JUANDA INTL	69.7	71.3	93.4	75.9	92.4	76.1	91.5	76.4	0	6287
MEDAN POLONIA INTL	72.6	73.1	93.6	78.7	92.8	78.8	91.6	78.8	0	6218
MINANGKABAU INTL	70.4	71.6	90.0	78.6	89.4	78.6	88.7	78.4	0	5700
SAM RATULANGI INTL	69.2	70.6	91.5	76.0	90.7	76.2	89.8	76.4	0	5560
SOEKARNO HATTA INTL	71.9	73.1	93.2	77.9	91.8	78.2	91.4	78.2	0	6245
SYARIF KASIM II INTL	71.6	72.3	94.3	79.5	93.4	79.5	92.6	79.4	0	6336
Iran, Islamic Republic of										
ABADAN	39.1	42.0	118.3	72.5	116.4	72.2	114.5	71.6	748	5964
AHVAZ	40.6	42.9	118.1	73.6	116.3	73.2	114.5	72.6	772	5987
ARAK	3.6	11.1	97.7	61.0	95.5	60.4	93.4	59.6	4363	1611
BANDAR ABBASS INTL	48.3	51.4	107.4	75.1	104.1	77.5	102.0	78.5	133	5853
BANDAR ANZALI	34.2	36.7	87.5	77.7	86.2	77.3	84.9	76.6	2705	1600
HAMEDAN	–0.3	6.3	96.0	62.2	94.0	61.2	91.9	60.5	5097	1000
ISFAHAN SHAHID BEHESHTI INTL	17.4	21.0	102.4	62.7	100.5	61.9	98.5	61.5	3627	1900
KASHAN	23.0	27.4	107.6	67.2	105.4	66.6	103.2	65.9	2665	3327
KERMAN	19.1	22.7	100.5	61.1	98.7	60.3	96.7	59.8	2911	1855
MASHHAD INTL	15.5	21.0	98.9	64.1	96.9	63.7	94.9	62.9	3688	1885
MEHRABAD INTL	25.5	28.6	101.9	64.5	99.3	64.2	97.2	63.7	2865	2787
SHAHID ASHRAFI ESFAHANI	17.8	21.9	103.8	63.9	101.8	63.3	99.4	62.1	3697	1832
SHIRAZ SHAHID DASTGHAIB INTL	28.0	30.2	102.6	64.1	100.8	63.3	98.9	62.6	2467	2565
TABRIZ INTL	10.8	15.5	96.6	62.2	93.6	61.5	91.5	61.0	4763	1482
URMIA	10.5	14.8	91.7	63.8	89.6	63.8	87.5	63.2	5184	823

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
ZAHEDAN INTL	22.7	26.3	102.4	61.6	100.5	60.9	98.6	59.8	2126	2644
ZANJAN	6.3	11.5	93.7	60.7	91.3	60.9	88.9	60.0	5289	818
Ireland										
CASEMENT	26.4	29.1	72.8	63.3	70.0	61.9	67.8	60.7	5666	13
DUBLIN	27.1	29.9	71.4	62.6	68.9	61.3	66.8	60.1	5696	7
Israel										
TEL AVIV BEN GURION	42.4	44.5	95.2	69.0	91.8	71.8	89.9	73.0	940	2589
TEL AVIV SDE DOV	45.1	47.7	88.5	74.2	86.8	75.6	85.9	75.5	879	2410
Italy										
BARI KAROL WOJTYLA	33.6	35.6	93.2	72.3	89.8	71.9	87.5	71.3	2668	1241
BOLOGNA	24.4	26.8	93.9	72.3	91.4	71.9	89.2	71.2	3834	1227
CATANIA FONTANAROSSA	34.8	37.2	93.9	73.8	91.1	73.5	88.2	73.1	1978	1492
CATANIA SIGONELLA	33.7	35.9	98.9	72.0	95.7	71.8	93.1	71.7	1937	1782
FIRENZE PERETOLA	26.5	29.8	95.4	71.3	93.0	71.0	89.9	70.1	3000	1307
GENOVA SESTRI	34.2	37.2	85.9	73.4	84.0	74.2	82.3	73.8	2440	1165
GRAZZANISE	30.4	32.4	91.2	74.6	89.1	74.3	86.4	73.9	2761	1141
MILANO LINATE	23.3	26.3	91.7	74.6	89.6	73.4	87.5	72.3	3856	1141
NAPOLI CAPODICHINO	34.0	36.2	91.3	73.6	89.3	73.5	87.4	73.3	2297	1424
PALERMO FALCONE-BORSELLINO	44.2	46.0	91.6	72.3	88.0	73.4	85.8	74.4	1438	1753
PRATICA DI MARE	33.5	35.6	87.8	73.6	85.9	73.9	84.2	74.4	2428	1098
ROMA CIAMPINO	30.1	32.2	93.1	71.0	90.7	70.9	88.0	70.2	2832	1225
ROMA LEONARDO DA VINCI	31.6	33.6	88.1	71.5	86.2	72.1	84.5	72.0	2668	1030
TORINO BRIC DELLA CROCE	23.1	26.2	82.7	69.1	80.6	68.3	78.6	67.2	4680	510
TORINO CASELLE	21.6	24.6	87.7	72.2	85.6	71.0	83.1	69.7	4386	718
TRIESTE	28.8	32.0	89.1	73.9	86.6	73.6	84.8	72.5	3153	1197
Jamaica										

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
NORMAN MANLEY INTL	72.0	73.3	91.8	78.8	91.4	78.8	90.2	78.5	0	6455
Japan										
AKITA	23.2	24.9	88.9	75.7	86.1	74.7	83.5	73.6	5032	935
ASAHIKAWA	0.6	4.6	85.4	72.2	82.5	70.1	80.1	68.7	7687	441
ASHIYA	30.4	32.2	90.3	78.4	88.2	78.4	86.4	77.9	3080	1517
ATSUGI NAS	30.3	32.1	91.5	77.8	89.6	77.0	87.7	76.6	3007	1607
CHIBA	32.7	34.1	90.5	78.2	88.7	77.7	87.0	77.1	2921	1594
FUJISAN	–19.6	–16.1	53.4	41.4	50.6	40.5	48.2	39.5	15985	0
FUKUOKA	31.9	33.7	93.2	78.1	91.4	77.8	89.5	77.1	2757	1908
FUKUYAMA	27.3	29.1	93.3	77.5	91.4	77.4	89.7	77.0	3332	1738
FUSHIKI	27.7	29.3	92.2	76.9	89.5	76.8	86.8	76.1	3956	1330
FUTENMA	51.9	53.5	90.0	79.5	89.5	79.5	88.1	79.3	334	3366
GIFU	26.2	28.1	94.6	78.0	91.7	77.2	89.5	76.4	3550	1662
HAMAMATSU	30.5	32.2	91.4	78.1	88.2	77.6	86.4	76.9	2884	1609
HIMEJI	28.1	29.7	92.2	77.8	90.4	77.4	88.6	76.8	3377	1670
HIROSHIMA	30.7	32.3	92.8	77.7	91.1	77.3	89.4	76.7	2957	1881
IIZUKA	28.8	30.7	92.4	78.0	90.7	77.8	88.7	77.1	3119	1705
IRUMA	26.4	28.2	93.5	77.6	91.4	77.1	88.2	76.1	3609	1414
KADENA AB	48.4	51.4	91.6	80.2	90.0	80.0	89.5	79.9	389	3385
KAGOSHIMA	34.6	36.6	92.3	78.4	90.7	78.0	89.2	77.7	1953	2328
KANAZAWA	29.8	31.1	91.7	76.6	89.8	76.4	87.8	75.9	3623	1484
KANSAI INTL	35.3	37.0	91.4	77.7	89.7	77.3	87.9	77.1	2738	1932
KOBE	32.2	34.1	91.7	77.2	89.8	76.9	88.0	76.5	2844	1943
KOCHI	30.7	32.5	91.3	77.5	89.7	77.2	88.1	76.8	2490	1907
KOMATSU	28.5	30.2	91.8	76.3	89.6	76.2	87.6	75.9	3777	1363
KUMAGAYA	28.2	29.9	95.8	77.8	93.3	77.2	90.5	76.1	3344	1620

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
KUMAMOTO	29.1	31.1	94.2	77.6	92.4	77.2	90.5	76.6	2697	2085
KURE	32.0	33.7	90.5	77.3	89.0	76.9	87.3	76.3	2891	1789
KYOTO	30.6	32.1	95.0	76.4	93.0	76.0	90.8	75.5	3137	1916
MATSUYAMA	32.0	33.7	92.2	76.6	90.7	76.3	89.1	75.9	2821	1847
MINAMITORISHIMA	64.4	65.7	89.5	79.4	88.6	79.1	87.7	78.9	0	4948
MIYAZAKI	31.8	34.0	92.8	78.1	90.6	78.1	88.6	77.8	2239	1986
NAGANO	20.0	22.2	91.2	74.4	88.8	73.7	86.2	72.7	4893	1176
NAGASAKI	33.5	35.4	91.0	77.9	89.3	77.8	87.6	77.4	2440	1930
NAGOYA	28.4	30.2	95.2	77.1	93.1	76.6	91.0	76.1	3207	1887
NAHA	55.0	55.6	90.0	79.6	89.5	79.6	88.1	79.7	211	3676
NARA	28.2	29.5	93.3	76.8	91.5	76.5	89.4	75.9	3498	1589
NAZE	49.2	50.7	91.2	78.9	90.0	78.8	88.9	78.6	627	2910
NIIGATA	28.9	30.3	91.1	77.1	88.9	76.5	86.4	75.8	4019	1327
NYUTABARU	30.0	32.2	91.2	78.0	88.2	78.3	86.3	78.0	2449	1732
OITA	31.3	33.2	92.3	77.6	90.5	77.3	88.6	76.8	2760	1761
OKAYAMA	30.0	31.7	94.0	77.3	92.2	76.9	90.4	76.4	3112	1931
OMAEZAKI	32.4	34.2	86.6	78.6	85.3	78.1	84.0	77.4	2618	1520
ONAHAMA	27.9	29.6	84.4	75.5	82.4	75.1	80.7	74.3	3851	907
OSAKA	33.3	34.8	93.9	76.6	92.1	76.3	90.4	75.9	2761	2090
OSAKA INTL	29.9	31.7	94.7	77.9	92.8	77.3	91.0	76.8	3105	1920
OTARU	14.9	17.0	82.8	72.0	79.9	70.2	77.4	69.0	6654	405
OZUKI	31.6	33.4	90.0	78.5	88.2	78.5	87.4	78.4	3068	1596
SAPPORO	14.1	16.5	84.6	72.6	81.9	70.9	79.4	69.3	6437	537
SENDAI	25.5	27.3	88.6	76.0	85.8	75.1	83.3	74.0	4474	902
SHIMOFUSA	28.4	30.3	92.4	78.5	89.9	77.5	87.9	77.1	3262	1492
SHIMONOSEKI	34.9	36.9	89.9	78.2	88.3	77.8	86.8	77.3	2575	1796

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
SHIZUHAMA	31.6	33.4	89.9	78.4	87.9	78.5	86.0	77.7	2740	1564
SHIZUOKA	31.7	33.6	91.2	77.5	88.9	77.2	87.0	76.6	2602	1682
SUMOTO	32.1	33.6	89.3	77.8	87.5	77.4	85.6	76.9	3135	1558
TADOTSU	31.9	33.7	92.6	77.1	90.8	76.7	89.0	76.3	2953	1858
TAKAMATSU	31.4	33.1	93.9	77.3	92.1	77.1	90.1	76.8	2953	1917
TOKUSHIMA	33.0	34.6	91.9	77.8	90.1	77.5	88.4	77.0	2756	1858
TOKYO	33.6	35.1	92.3	77.4	90.4	76.8	88.5	76.1	2800	1742
TOKYO INTL	33.7	35.4	91.3	78.5	89.4	77.7	87.5	77.4	2836	1621
TOYAMA	27.6	29.3	92.8	77.8	90.5	77.3	88.0	76.5	3856	1418
TSUIKI	28.3	30.1	91.0	78.6	88.2	78.5	86.4	77.9	3333	1503
UTSUNOMIYA	24.6	26.4	92.4	77.7	90.0	77.0	87.5	75.9	3841	1343
WAKAYAMA	32.9	34.4	92.1	76.5	90.2	76.5	88.6	76.4	2784	1917
YOKOHAMA	33.2	34.6	90.7	77.8	88.8	77.1	87.0	76.4	2868	1589
YOKOSUKA	35.5	37.2	93.3	78.6	89.8	77.4	87.4	76.7	2514	1688
YOKOTA AB	25.1	27.2	93.2	78.5	91.1	77.7	88.1	76.3	3568	1394
Jordan										
AMMAN	34.0	36.8	96.8	65.5	93.6	64.9	91.6	64.6	2146	2064
IRBID MET	35.1	37.9	93.9	67.3	91.3	66.9	89.0	66.8	1975	1926
QUEEN ALIA INTL	30.6	33.3	98.6	67.6	95.4	66.2	93.3	66.0	2437	1492
Kazakhstan										
ALMATY	–4.3	1.0	93.5	65.1	90.9	64.4	87.9	63.7	6455	859
ASTANA INTL	–27.0	–21.9	90.1	63.9	86.4	63.0	83.2	62.2	10323	344
KARAGANDY INTL	–26.9	–20.4	89.7	62.1	86.1	61.3	82.7	60.1	10123	299
PAVLODAR	–30.9	–24.7	91.4	65.1	87.8	64.7	84.4	63.6	10306	442
SHYMKENT	2.9	8.7	100.0	66.2	97.1	65.4	95.0	64.9	4646	1556
TARAZ	–4.3	1.5	96.5	64.3	93.5	63.7	91.1	63.1	5721	1095

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Kenya										
MOMBASA INTL	68.2	69.5	91.6	77.3	90.7	77.2	89.8	77.0	0	5465
NAIROBI JOMO KENYATTA INTL	50.1	52.1	84.3	60.4	82.7	60.5	81.3	60.9	152	1062
Korea, Democratic People's Republic of										
CHONGJIN	9.6	12.4	81.4	72.6	79.1	71.6	77.1	70.8	6796	417
HAMHUNG	8.8	12.2	88.7	75.0	85.9	74.1	83.1	72.6	5736	762
KAESONG	9.0	12.3	87.7	77.3	85.3	75.7	83.2	74.5	5509	1026
NAMPO	9.0	12.4	86.4	77.9	84.3	76.5	82.5	75.2	5693	1050
PYONGYANG SUNAN INTL	5.6	9.2	88.2	75.9	86.2	75.0	84.3	74.0	5838	1114
SINUIJU	4.0	7.6	87.5	75.3	85.0	74.1	82.8	73.3	6266	945
WONSAN	13.6	16.9	89.0	74.3	86.1	73.5	83.4	72.6	5252	822
Korea, Republic of										
BUSAN	23.0	26.0	88.1	77.8	86.2	77.2	84.3	76.4	3351	1291
BUSAN GIMHAE INTL	21.0	23.1	91.1	78.5	88.2	77.7	86.2	76.7	3776	1415
CHANGWON	23.0	26.0	90.3	77.8	88.0	77.1	85.8	76.1	3542	1435
CHEONGJU	12.8	16.2	91.0	76.3	88.7	74.9	86.5	73.7	4782	1359
CHEONGJU INTL	8.2	12.1	91.6	79.1	89.5	77.5	87.4	76.4	5130	1303
DAEGU	19.7	22.5	93.7	75.4	91.2	74.7	88.9	73.8	3933	1551
DAEGU AB	17.4	19.8	94.7	77.8	91.6	76.7	89.5	75.6	4193	1482
DAEJEON	12.8	16.2	90.6	77.4	88.2	75.9	86.1	74.6	4827	1252
GIMPO INTL	8.3	11.8	89.8	76.6	87.7	76.0	85.7	74.2	5404	1177
GWANGJU	20.3	23.0	90.7	77.0	88.6	76.1	86.6	75.0	4069	1462
GWANGJU INTL	19.3	21.5	93.4	79.3	91.1	78.1	88.2	76.6	4295	1509
INCHEON	13.7	17.1	87.9	76.5	85.5	75.3	83.4	74.1	4862	1148
JEJU	32.7	34.4	89.7	77.5	87.8	77.4	86.1	77.0	2965	1481
JEJU INTL	31.9	33.7	89.6	79.2	87.6	79.6	85.8	78.8	3168	1368

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
JEONJU	16.3	19.6	91.8	76.8	89.7	76.0	87.5	74.5	4386	1470
JINJU	16.8	19.4	91.3	76.7	89.0	76.0	86.6	75.0	4306	1303
OSAN AB	8.4	12.1	91.4	79.3	89.2	77.9	86.3	76.1	5173	1276
POHANG	19.2	21.5	93.2	78.6	90.9	77.8	87.8	76.6	4027	1234
PYEONGTAEK A511	9.3	12.6	91.1	78.5	88.2	76.9	86.2	75.5	5172	1262
SEOGWIPO	33.0	35.1	88.7	79.6	87.1	79.1	85.5	78.4	2492	1629
SEOUL AB	5.3	10.3	91.7	77.6	89.5	76.3	86.4	74.4	5240	1207
SEOUL CITY	11.5	15.2	89.8	75.4	87.6	73.9	85.5	72.9	4831	1325
SEOUL SINYONGSAN	10.8	13.9	91.7	77.3	89.6	76.0	87.4	75.3	4716	1433
SUWON	11.6	15.2	89.9	77.0	87.6	75.6	85.5	74.3	4991	1281
ULSAN	21.5	24.3	91.7	76.6	89.3	76.1	86.9	75.3	3750	1303
WANDO	24.9	27.4	88.3	78.6	86.3	77.7	84.2	76.7	3773	1270
YEOSU	23.5	26.1	86.8	77.2	84.9	76.5	83.1	75.9	3633	1253
Kyrgyzstan										
BISHKEK	–0.2	5.2	95.3	66.6	92.7	65.1	90.1	64.2	5434	1152
Latvia										
RIGA	–2.5	4.8	84.0	68.2	80.6	67.4	77.2	65.0	7421	168
Lebanon										
BEIRUT RAFIC HARIRI INTL	46.6	49.2	90.6	74.2	88.3	75.6	87.4	75.7	718	2688
Libyan Arab Jamahiriya										
BENINA INTL	44.3	46.1	98.7	69.2	95.3	68.9	92.6	68.4	1090	2453
MISRATA	46.9	48.6	98.3	70.7	94.0	70.5	90.5	70.8	807	2535
TRIPOLI INTL	40.3	42.4	107.5	73.3	103.9	72.2	100.3	71.5	1136	3037
Lithuania										
KAUNAS	–1.7	4.4	83.1	67.8	79.8	66.0	76.7	64.2	7380	151
VILNIUS INTL	–2.7	3.2	83.7	66.8	80.3	65.3	77.0	63.7	7692	159

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
Macao										
MACAU INTL	45.3	48.1	91.3	80.7	89.7	80.4	88.2	80.2	508	3626
Macedonia, the former Yugoslav Republic of										
SKOPJE ALEXANDER THE GREAT	10.5	17.2	96.8	68.6	93.5	68.1	91.0	67.3	4623	984
Madagascar										
IVATO	46.3	48.2	85.9	67.4	84.1	67.5	82.5	67.5	529	1298
Malaysia										
KOTA KINABALU INTL	73.0	73.4	92.9	82.4	91.7	81.8	91.2	81.6	0	6284
KUALA LUMPUR SUBANG	72.9	73.5	94.7	79.3	93.4	79.1	92.8	78.9	0	6629
KUANTAN	71.1	71.8	93.3	80.2	92.2	80.0	91.5	79.9	0	6044
KUCHING INTL	71.6	72.3	93.2	78.6	91.8	78.7	91.2	78.7	0	5898
SANDAKAN	73.2	73.9	92.8	79.7	91.5	79.5	90.5	79.5	0	6238
TAWAU	71.8	72.8	90.9	78.9	89.9	79.0	89.4	79.0	0	5841
Mali										
BAMAKO-SENOU INTL	58.9	61.2	104.4	67.5	103.2	67.7	101.9	68.0	0	6328
Mauritania										
NOUAKCHOTT	55.2	57.4	106.2	68.4	103.4	68.2	100.0	68.0	4	5374
Mexico										
ACAPULCO INTL	66.6	69.4	91.8	79.9	91.5	79.8	91.0	79.5	0	5904
CANCUN INTL	55.7	58.8	93.3	80.7	91.7	80.2	91.3	80.1	5	5145
CHETUMAL INTL	60.1	63.3	93.4	80.3	92.1	80.2	91.1	80.1	0	5867
GUADALAJARA INTL	35.4	37.6	91.7	60.4	89.9	59.6	88.1	59.0	633	1297
GUANAJUATO INTL	39.3	42.6	93.5	58.5	91.4	58.4	89.5	58.4	480	1430
HERMOSILLO INTL	40.7	43.2	109.0	73.0	106.4	73.5	104.4	73.1	379	4827
MAZATLAN INTL	46.8	49.8	93.2	77.9	91.7	77.5	91.2	77.3	56	3835
MERIDA INTL	56.8	60.4	101.8	75.8	99.0	76.0	97.2	76.0	2	5896

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
MEXICO CITY INTL	37.6	40.9	84.5	54.7	82.5	54.3	80.7	54.1	1045	368
MONTERREY INTL	37.5	41.0	101.9	73.4	100.0	73.3	98.3	73.5	666	3855
PUERTO VALLARTA INTL	57.4	59.3	91.8	80.3	91.5	80.2	91.0	80.0	1	4629
SAN LUIS POTOSI INTL	32.7	35.9	89.9	58.5	87.7	58.7	85.6	58.7	1200	775
TAMPICO INTL	50.4	53.6	93.4	80.1	91.8	79.9	91.3	79.8	138	4609
TAPACHULA	68.0	69.6	95.8	78.8	94.6	78.7	93.5	78.7	0	6268
TIJUANA INTL	42.5	44.3	90.0	68.5	86.3	67.6	84.0	66.9	1268	909
TOLUCA INTL	28.3	30.5	79.0	53.8	77.1	53.4	75.2	53.0	3196	3
TORREON INTL	38.9	42.3	100.4	69.1	98.4	68.7	96.6	68.4	567	3687
VERACRUZ INTL	58.7	60.8	95.2	80.3	93.4	80.3	91.8	79.9	6	4937
Moldova, Republic of										
CHISINAU	6.3	11.3	90.4	67.6	87.1	66.8	84.3	65.9	5758	718
Mongolia										
CHINGGIS KHAAN INTL	–33.0	–29.0	87.7	60.3	83.8	58.9	80.1	57.7	12689	177
Morocco										
AGADIR AL MASSIRA INTL	40.9	43.1	101.9	67.7	94.9	66.4	89.8	66.1	708	1688
CASABLANCA ANFA	43.6	45.8	85.5	70.7	81.6	71.3	79.2	71.2	1144	1166
CASABLANCA MOHAMMED V INTL	37.3	39.5	96.7	70.6	91.7	70.4	87.9	69.8	1469	1424
FES-SAISS	33.5	35.5	103.2	67.9	99.6	67.9	96.4	67.5	2161	1577
INEZGANE	41.0	43.6	95.4	66.9	89.3	65.8	84.2	65.2	939	1168
MARRAKECH MENARA	39.2	41.4	107.6	69.0	103.8	68.9	100.2	68.5	1116	2574
MEKNES BASSATINE	36.2	38.8	102.2	69.7	98.3	69.5	94.6	68.9	1975	1565
OUJDA ANGADS	33.6	35.9	100.3	69.0	96.6	69.2	93.3	68.8	2000	1616
RABAT SALE INTL	40.7	42.6	91.2	70.7	86.0	70.2	82.4	70.5	1444	981
TANGIER IBN BATOUTA	39.4	42.4	91.8	70.5	89.4	70.3	86.3	69.9	1428	1315
TETOUAN SANIA RAMEL	43.1	45.6	91.4	68.8	88.0	68.8	85.5	68.8	1127	1504

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Mozambique										
MAPUTO INTL	53.4	55.3	95.7	74.7	92.7	74.7	89.9	74.7	31	3561
Netherlands										
AMSTERDAM SCHIPHOL	20.8	24.5	82.0	67.6	78.1	66.2	74.7	64.5	5272	128
HOEK VAN HOLLAND	22.7	26.0	80.9	66.7	76.4	65.2	73.0	64.4	4962	126
IJMUIDEN	20.5	24.6	78.0	65.7	74.4	64.1	71.3	63.6	5257	92
ROTTERDAM THE HAGUE	20.8	24.6	82.5	67.8	78.5	66.4	75.2	64.6	5223	128
VALKENBURG	20.7	24.4	80.8	67.4	76.7	65.6	73.3	64.0	5294	101
WOENSDRECHT AB	18.2	22.7	84.4	67.6	80.1	66.3	76.7	64.7	5334	140
New Zealand										
AUCKLAND AERO AWS	40.0	42.1	77.3	67.4	75.7	66.5	74.2	65.8	2199	282
AUCKLAND INTL	39.2	41.3	78.3	67.8	76.6	66.9	74.9	66.0	2193	309
CHRISTCHURCH AP AWS	27.6	29.4	81.7	61.8	77.9	60.5	74.4	59.4	4675	89
CHRISTCHURCH INTL	26.9	28.7	82.4	62.4	78.7	60.8	75.1	59.8	4715	100
Nicaragua										
MANAGUA INTL	67.9	69.6	96.8	75.7	95.3	75.5	94.6	75.4	0	6263
Niger										
NIAMEY DIORI HAMANI INTL	60.7	62.6	108.1	68.9	107.2	68.8	105.6	68.8	0	7649
Norway										
HAKADAL	–2.3	3.0	79.8	63.8	76.7	62.4	73.4	60.7	8124	79
OSLO-BLINDERN	6.2	10.2	80.1	63.2	76.7	61.9	73.5	60.2	7528	98
Oman										
AL BURAIMI	49.7	52.2	113.2	70.7	111.4	70.3	109.7	70.2	128	6761
Pakistan										
BENAZIR BHUTTO INTL	35.8	37.8	106.0	72.9	102.9	73.0	100.4	72.9	1150	3679
JINNAH INTL	50.2	53.3	102.0	72.9	98.8	73.5	96.8	74.2	43	5846

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
LAHORE ALLAMA IQBAL INTL	37.4	40.6	109.8	74.1	107.5	74.0	104.2	73.9	800	4695
Palestinian Territory, Occupied										
JERUSALEM ATAROT	33.4	35.9	91.6	65.0	89.2	64.9	86.6	64.6	2327	1421
Panama										
PANAMA PACIFICO INTL	71.9	73.2	95.0	77.7	93.5	77.5	92.9	77.4	0	6418
TOCUMEN INTL	69.5	70.8	93.4	77.6	92.1	77.1	91.5	76.9	0	6053
Paraguay										
SILVIO PETTIROSSI INTL	41.2	44.7	98.9	75.0	97.0	75.3	95.1	75.3	462	3796
Peru										
AREQUIPA INTL	42.7	44.3	75.3	52.5	73.8	51.7	73.3	51.5	2049	3
CHICLAYO INTL	58.8	59.3	90.0	75.5	89.2	75.2	87.5	74.4	2	2927
CUSCO INTL	32.2	34.0	73.5	50.3	71.9	49.9	71.0	49.6	3602	0
HUANCHACO INTL	57.7	58.6	82.4	74.0	80.8	73.4	79.1	72.1	235	1208
IQUITOS INTL	66.3	68.4	93.5	79.5	92.7	79.5	91.4	79.3	0	5502
LIMA CALLAO INTL	57.2	58.0	83.9	72.9	82.0	72.0	80.4	71.5	327	1402
PIURA INTL	60.5	61.5	93.3	77.7	91.8	77.1	90.8	76.7	0	4295
PUCALLPA INTL	64.0	66.2	94.7	79.2	93.4	79.1	92.3	78.8	1	5655
Philippines										
CAGAYAN DE ORO	71.8	73.0	94.3	81.5	93.3	81.3	92.5	81.1	0	6482
FRANCISCO BANGOY INTL	73.0	73.5	93.1	80.4	91.8	80.3	91.4	80.2	0	6394
GENERAL SANTOS INTL	72.9	73.4	95.1	81.1	93.8	80.9	92.8	80.7	0	6395
ILOILO	73.0	73.9	94.6	81.8	93.3	81.6	92.0	81.3	0	6408
MACTAN CEBU INTL	74.0	74.9	91.7	81.1	90.7	80.8	89.9	80.7	0	6359
MANILA	73.7	74.8	94.2	79.5	93.0	79.5	91.8	79.3	0	6708
NINOY AQUINO INTL	71.2	72.3	95.1	78.9	93.5	78.7	92.6	78.6	0	6386
SANGLEY POINT AB	72.2	74.1	95.2	83.4	93.9	83.0	92.8	82.6	0	6805

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
SCIENCE GARDEN	68.5	70.0	95.3	79.3	94.0	79.2	92.7	79.1	0	6105
ZAMBOANGA INTL	72.8	73.7	93.4	81.5	92.6	81.3	91.7	81.0	0	6533
Poland										
GDANSK LECHA WALESY	4.9	10.4	80.9	66.3	77.3	64.8	74.9	63.6	7102	93
GDANSK-SWIBNO	1.4	9.1	78.5	67.0	74.6	65.0	71.5	63.7	7003	62
HEL	15.6	19.3	77.7	68.5	74.8	66.8	72.4	65.5	6462	104
KATOWICE	5.2	10.6	85.5	67.9	82.0	66.0	78.9	64.6	6605	192
KRAKOW BALICE	3.5	8.8	86.1	68.7	82.6	67.2	79.3	65.7	6527	233
LODZ	5.5	10.6	85.9	67.4	82.4	65.9	79.1	64.4	6650	228
LUBLIN RADAWIEC	2.2	8.0	84.3	68.4	80.9	67.2	77.8	65.5	6967	191
POZNAN LAWICA	7.6	13.0	86.4	66.9	82.9	65.5	80.1	64.3	6338	247
RACIBORZ	4.8	10.4	85.9	68.3	82.4	66.8	79.1	65.5	6301	220
SZCZECIN	10.4	15.8	84.4	68.0	80.8	66.4	77.6	65.1	6233	179
TERESPOL	–0.9	5.8	85.7	68.8	82.1	67.5	79.0	65.5	6933	219
WARSZAWA OKECIE	3.6	9.6	85.9	68.5	82.4	66.7	79.1	64.9	6620	250
WROCLAW STRACHOWICE	6.9	12.7	86.5	68.0	83.2	66.6	80.0	65.3	6159	245
Portugal										
LISBOA	40.3	42.4	92.8	68.6	89.1	67.5	85.4	66.6	1858	1022
Puerto Rico										
JOSE APONTE DE LA TORRE AP	68.4	70.0	89.9	78.4	89.4	78.1	88.2	77.7	0	5760
LUIS MUNOZ MARIN INTL	69.7	70.6	91.6	77.4	89.8	77.8	88.9	77.8	0	5692
Qatar										
DOHA INTL	53.3	55.4	111.3	71.9	109.2	72.4	107.2	73.0	102	6706
Romania										
BUCURESTI BANEASA	8.5	13.6	93.6	69.7	90.8	69.1	87.8	68.1	5387	738
BUCURESTI HENRI COANDA INTL	7.0	12.4	93.3	70.5	89.9	70.4	87.6	69.1	5314	800

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
CLUJ NAPOCA	5.6	10.8	87.8	68.6	84.6	67.4	81.8	66.2	6224	338
CONSTANTA	15.8	19.6	86.3	74.2	84.1	73.0	82.1	72.1	4632	856
CRAIOVA	9.7	14.5	93.3	70.3	90.1	69.7	87.3	69.0	5141	849
IASI	3.5	9.3	91.6	69.8	88.2	68.7	85.4	67.8	5771	651
MIHAIL KOGALNICEANU	12.2	16.0	91.3	70.9	88.0	69.8	85.7	69.3	5078	791
TIMISOARA TRAIAN VUIA	11.1	15.8	93.4	69.8	90.1	69.4	87.4	68.1	5111	685
Russian Federation										
ARKHANGELSK TALAGI	–27.1	–20.3	81.6	67.0	77.3	64.4	73.4	62.4	11188	84
ASTRAKHAN	–2.1	4.0	97.1	70.1	94.2	69.4	91.2	68.8	6042	1265
BALANDINO	–20.5	–15.1	87.4	66.7	83.8	65.6	80.4	64.4	10015	265
BARNAUL	–28.9	–23.1	86.2	65.9	83.1	64.9	80.4	63.7	10514	295
BOLSHOYE SAVINO	–23.4	–17.0	85.8	68.6	82.2	66.5	78.7	64.6	10332	198
BOLVANSKIY	–26.6	–23.1	57.2	52.9	53.2	50.5	49.7	47.7	15996	0
BRYANSK	–7.8	–1.9	84.6	67.6	81.1	65.5	78.3	64.3	8106	231
CHEREPOVETS	–21.2	–15.4	82.9	68.0	79.1	66.3	75.5	64.1	9925	101
CHERTOVITSKOYE	–11.6	–5.7	91.3	66.6	87.2	65.6	82.7	64.4	7876	385
CHITA KADALA	–35.1	–31.4	87.7	66.4	84.1	64.6	80.6	62.8	12615	183
GUMRAK	–8.2	–3.4	95.6	65.7	91.9	65.3	88.3	64.6	7327	847
IRKUTSK	–32.1	–25.8	83.8	64.0	80.5	63.5	77.2	62.1	11892	96
IZHEVSK	–20.8	–14.9	85.3	67.8	81.9	66.0	78.7	64.6	10155	223
KALUGA	–13.5	–7.2	83.0	66.7	80.0	65.5	77.0	64.5	8725	140
KAZAN	–18.1	–12.2	87.6	67.8	83.9	66.8	80.5	65.2	9414	328
KEMEROVO	–29.0	–23.7	84.1	66.2	80.7	64.8	77.4	63.4	11184	206
KHABAROVSK	–22.2	–18.8	86.6	72.2	83.9	71.1	80.8	69.5	10906	408
KHRABROVO	2.5	8.6	82.7	67.9	78.9	65.9	75.6	64.1	6885	131
KIROV	–20.3	–14.7	85.1	68.7	81.7	66.6	78.5	64.8	10066	226

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
KOLTSOVO	–23.3	–18.0	85.4	67.1	82.0	65.4	78.7	63.9	10572	173
KRASNODAR	5.6	12.0	94.0	72.4	91.0	71.6	87.7	70.4	5098	973
KRASNOYARSK	–28.6	–23.9	83.1	65.0	79.9	63.7	76.6	62.3	11257	122
KRASNOYARSK MININO	–33.9	–28.9	84.1	66.6	80.8	65.3	77.6	63.7	11099	183
KURGAN	–26.6	–21.0	88.5	66.6	84.9	65.8	81.6	64.7	10516	303
KURSK	–7.9	–2.3	87.4	67.3	83.8	65.9	80.6	64.7	7834	369
KURUMOCH	–16.2	–10.4	90.0	66.8	86.4	65.9	83.2	64.7	8907	439
MAGNITOGORSK	–21.3	–16.6	87.3	65.3	83.8	64.0	80.7	62.9	10346	261
MAKHACHKALA	9.3	15.1	88.9	73.7	86.5	73.8	84.4	73.1	4943	1047
MOSCOW BIBIREVO	–7.9	–2.2	85.4	69.6	82.0	68.1	78.7	66.4	8333	248
MOSCOW SHEREMETYEVO	–11.2	–5.7	86.0	66.5	82.4	65.3	78.8	64.5	8741	216
MOSCOW VNUKOVO	–9.7	–4.2	85.7	66.8	82.1	65.7	78.7	64.6	8664	230
MURMANSK	–26.0	–20.2	75.6	61.0	71.4	58.8	67.4	57.0	11988	18
NIZHNY NOVGOROD STRIGINO	–16.2	–10.0	86.2	68.2	82.7	66.8	79.4	65.4	9044	241
NIZHNY TAGIL	–24.1	–19.4	83.0	66.8	79.8	65.1	76.8	63.5	10971	112
NOVOKUZNETSK	–27.5	–22.4	84.8	66.8	81.6	65.4	78.6	64.4	10695	189
NOVOSIBIRSK TOLMACHEVO	–33.0	–27.2	85.7	66.0	82.4	64.7	79.2	63.3	11102	219
OMSK	–27.4	–22.3	87.9	65.8	84.3	64.8	80.9	63.6	10874	294
ORENBURG	–21.6	–15.3	94.4	67.0	90.9	66.2	87.1	65.1	9158	574
ORYOL	–10.4	–4.4	86.7	68.1	83.1	66.9	80.0	65.3	8064	298
PENZA	–16.9	–11.1	89.6	68.0	85.7	67.1	82.2	65.6	8842	339
PSKOV	–9.6	–2.5	83.7	68.5	80.3	66.6	77.0	64.6	8165	155
ROSTOV-NA-DONU	0.2	5.3	94.9	70.4	91.2	69.9	87.7	68.4	6121	921
RYAZAN	–11.6	–5.9	86.5	67.9	82.8	66.3	79.5	64.9	8545	282
SARATOV TSENTRALNY	–10.5	–5.5	91.8	67.7	88.0	66.8	84.5	65.8	8079	649
SMOLENSK	–7.8	–2.2	82.3	67.9	79.1	66.1	76.1	64.6	8433	149

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
SOCHI INTL	28.5	31.2	87.4	75.0	84.6	74.1	82.7	73.4	3511	917
ST PETERSBURG PULKOVO	–9.0	–3.5	82.8	67.3	79.1	65.6	75.7	63.8	8555	127
STAVROPOL INTL	–0.2	6.4	92.5	67.3	89.3	66.5	85.8	65.9	6002	734
SURGUT	–40.3	–35.5	84.0	65.4	80.5	63.8	76.7	62.9	13306	150
TOMSK	–33.4	–27.7	83.5	67.5	80.5	65.9	77.5	64.4	11499	176
TRUBCHEVSK	–9.1	–2.8	86.0	68.9	82.4	67.4	79.2	66.1	7879	243
TULA	–12.3	–6.3	86.1	68.1	82.5	66.4	79.5	65.3	8491	251
TVER	–13.0	–7.1	85.1	67.1	81.4	66.0	78.0	64.4	8766	193
TYUMEN	–25.9	–20.8	85.2	67.2	82.0	65.8	78.9	64.7	10798	213
UFA	–24.2	–18.4	88.3	69.0	85.3	67.6	82.1	66.2	9806	290
ULAN-UDE	–34.0	–29.2	88.7	64.7	84.6	63.7	81.1	62.4	12387	245
VELIKIYE LUKI	–8.5	–2.3	82.4	66.5	79.5	65.7	76.5	64.3	8265	134
VLADIKAVKAZ	5.7	11.1	87.6	68.6	84.4	67.8	81.3	66.6	6066	480
VLADIMIR	–13.8	–8.3	85.0	69.4	81.4	68.0	78.1	66.4	8968	228
VLADIVOSTOK	–13.4	–8.7	82.8	70.3	80.0	69.0	76.9	67.7	8953	298
VORONEZH	–10.3	–4.8	90.2	68.1	86.7	67.2	83.1	65.7	7651	499
YELABUGA	–19.7	–13.8	87.7	68.7	83.8	67.4	80.5	65.5	9575	317
Saudi Arabia										
ABHA	43.7	46.1	88.2	55.7	87.4	55.7	85.9	56.0	864	1475
DHAHARAN KING ABDULAZIZ	46.2	48.4	113.4	74.2	111.3	74.2	109.4	74.0	312	6187
GASSIM PRINCE ABDULAZIZ	37.8	41.3	113.1	69.3	111.3	67.8	109.7	66.9	742	5388
JEDDAH KING ABDULAZIZ INTL	60.4	62.3	105.7	74.7	103.8	75.6	101.9	76.2	1	6863
KHAMIS MUSHAIT KING KHALED AB	45.1	47.8	89.5	59.7	88.1	59.4	87.4	59.2	563	1876
MAKKAH	61.3	64.0	113.3	75.9	111.4	75.6	109.6	75.5	1	8694
MEDIAN PRINCE ABDULAZIZ INTL	48.5	51.6	113.3	66.5	111.5	65.8	109.8	65.2	137	6846
RIYADH KING SALMAN AB	42.7	46.0	112.5	66.8	111.1	66.1	109.5	65.4	497	6058

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
TABUK	35.4	37.6	106.0	66.0	103.8	65.1	101.9	64.5	1182	3839
Senegal										
LEOPOLD SEDAR SENGHOR INTL	62.3	62.9	90.0	73.7	88.2	77.0	87.6	77.6	0	4356
Serbia										
BEOGRAD	16.1	19.9	93.7	70.4	90.7	69.7	87.7	68.5	4404	998
BEOGRAD SURCIN	13.7	17.7	94.0	70.2	91.0	70.3	87.7	68.9	4739	820
Singapore										
SINGAPORE CHANGI INTL	73.6	74.8	91.8	79.5	91.3	79.5	90.0	79.3	0	6489
Slovakia										
BRATISLAVA-STEFANIK	12.5	17.4	90.2	69.1	87.2	67.9	84.0	66.5	5386	534
South Africa										
BLOEMFONTEIN INTL	23.1	25.4	92.8	59.9	90.3	59.7	88.1	59.7	2506	906
CAPE TOWN INTL	39.1	41.3	88.9	67.4	85.6	66.5	82.4	65.6	1574	742
DE AAR	30.7	33.1	94.5	60.9	92.5	60.6	90.5	60.3	2048	1372
DURBAN INTL	48.4	50.5	86.8	75.0	84.9	74.4	83.6	73.9	254	1984
EAST LONDON	46.3	48.1	87.2	68.1	83.9	68.9	81.0	69.2	747	1042
JOHANNESBURG INTL	32.2	35.5	84.2	58.8	82.3	59.1	80.4	59.5	1969	483
PORT ELIZABETH INTL	41.4	44.2	84.6	66.1	81.3	67.2	79.0	67.6	1194	722
PRETORIA	37.2	39.2	90.2	63.2	88.0	63.0	86.2	63.1	1060	1556
Spain										
A CORUNA	40.1	42.3	78.4	66.4	75.4	65.3	73.1	64.4	2515	219
ALICANTE	37.9	40.5	90.8	70.2	88.2	70.8	86.4	71.3	1616	1593
BARCELONA EL PRAT	34.9	37.1	87.4	74.5	85.2	74.1	83.7	73.5	2344	1163
BILBAO	31.7	33.9	89.9	69.4	85.6	68.4	82.0	67.3	2740	642
GRAN CANARIA	56.8	57.7	86.3	67.7	83.4	68.2	81.2	69.1	113	1965
MADRID BARAJAS	25.0	27.6	97.6	65.6	95.3	64.8	93.0	64.2	3547	1182

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
MALAGA	39.3	41.9	95.2	68.2	91.4	68.3	87.9	67.8	1457	1596
MURCIA	36.5	39.1	96.9	71.3	94.4	71.0	92.2	70.8	1592	2016
PALMA DE MALLORCA	32.4	35.2	91.7	72.6	89.3	72.6	87.0	72.6	2274	1280
SEVILLA	35.6	38.2	102.6	72.1	100.3	71.2	97.1	69.9	1527	2218
TORREJON AB	23.3	26.4	97.1	67.4	95.0	66.2	92.8	65.0	3779	1070
VALENCIA	33.5	35.8	91.8	69.5	89.4	70.5	87.4	70.9	1958	1502
VALLADOLID	25.1	27.3	93.8	64.7	90.9	64.1	87.6	63.3	4311	652
ZARAGOZA AB	28.1	30.4	97.2	70.2	94.4	69.2	91.2	68.3	3066	1277
Sri Lanka										
BANDARANAIKE INTL	69.8	71.6	91.6	76.3	90.1	77.3	89.6	77.6	0	6129
Sweden										
GOTEBORG	10.4	14.9	80.3	64.5	77.4	63.8	74.3	62.3	6514	114
MALMO	14.6	18.9	79.7	66.0	76.9	65.1	73.7	63.6	6310	85
STOCKHOLMN BROMMA	5.0	10.5	80.6	64.2	77.3	62.7	74.6	61.6	7469	99
UPPSALA	–1.9	4.3	80.1	65.2	76.8	63.4	73.7	62.0	8059	55
Switzerland										
BERN-ZOLLIKOFEN	14.3	18.2	85.4	67.0	82.2	66.2	79.0	64.8	6028	230
LAEGEREN	12.7	16.5	79.4	63.3	76.1	62.1	73.4	61.2	7001	127
ZUERICH-FLUNTERN	17.0	20.6	84.4	66.8	81.2	65.5	78.1	64.2	5756	269
Syrian Arab Republic										
ALEPPO INTL	28.3	30.6	102.8	67.9	100.2	67.5	97.4	67.4	2693	2511
DAMASCUS INTL	25.9	28.7	103.5	65.7	100.5	64.9	98.5	64.6	2588	2105
DARAA	33.5	36.1	97.2	66.8	94.6	67.0	92.1	67.2	2051	1957
HAMA	29.8	32.6	102.7	69.6	99.9	69.1	97.5	68.3	2306	2547
LATAKIA	39.2	41.7	91.1	72.6	89.2	74.8	87.7	75.6	1282	2164
Taiwan										

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
GANGSHAN	49.9	52.2	91.8	80.9	91.2	80.8	89.9	80.4	143	4071
HENGCHUN	60.8	62.4	90.8	80.3	89.9	80.1	89.0	79.8	10	4742
HSINCHU	48.4	50.3	91.7	82.3	90.7	81.9	89.7	81.4	496	3323
HSINCHU CITY	48.4	50.6	91.7	80.0	90.6	79.8	89.4	79.5	480	3262
KAOHSIUNG	55.2	57.3	90.9	80.9	90.0	80.7	89.1	80.5	55	4575
KAOHSIUNG INTL	53.8	56.5	91.8	80.0	91.4	80.0	90.0	79.5	57	4698
KEELUNG	50.9	52.7	93.0	78.6	91.3	78.4	89.9	78.4	426	3316
KINMEN	44.8	46.6	91.3	83.2	89.8	82.7	88.9	82.3	894	2865
MATSU NANGAN	40.1	42.1	88.9	81.9	87.6	81.6	86.1	80.8	1795	2155
PINGTUNG NORTH	52.0	54.8	93.9	81.1	93.2	80.9	91.8	80.4	72	4520
PINGTUNG SOUTH	53.3	55.3	94.9	81.2	93.5	80.8	92.7	80.5	57	4721
TAICHUNG	46.3	48.6	93.6	82.1	92.9	82.0	91.6	81.5	326	3744
TAICHUNG AP	46.3	48.2	89.9	80.9	89.2	80.7	88.0	80.4	562	2974
TAINAN	50.4	53.3	92.3	81.9	91.5	81.7	90.1	81.1	131	4275
TAIPEI	49.7	51.7	94.9	80.4	93.4	80.0	92.1	79.6	394	3758
TAIPEI SONGSHAN	49.6	51.6	95.2	79.7	93.5	79.6	92.8	79.6	393	3805
TAIWAN TAOYUAN INTL	48.4	50.4	94.6	80.1	93.0	80.4	91.4	80.0	475	3493
TAOYUAN	47.7	49.6	93.2	82.8	91.7	82.2	90.9	81.8	594	3252
WU-CHI OBSERVATORY	50.6	52.5	91.1	81.2	90.2	80.9	89.3	80.6	347	3574
Tajikistan										
DUSHANBE	14.4	21.0	100.2	66.0	98.2	65.6	95.6	65.1	3498	1668
Tanzania, United Republic of										
DAR ES SALAAM INTL	64.6	65.9	91.7	78.2	91.1	78.0	89.9	77.6	0	5319
Thailand										
BANGKOK INTL	67.2	69.7	99.0	79.4	97.6	79.5	96.5	79.4	0	7194
PHUKET	74.8	75.5	94.0	78.6	92.9	78.5	92.0	78.4	0	6853

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

Station	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
	99.6%	99%	DB / MCWB		DB / MCWB		DB / MCWB		HDD / CDD 65	
Togo										
LOME-TOKOIN	70.9	72.0	91.8	79.0	91.3	79.5	90.0	79.5	0	6157
Tunisia										
TUNIS-CARTHAGE INTL	41.3	43.3	100.1	72.5	96.4	72.5	93.2	72.1	1351	2275
Turkey										
ADANA INCIRLIK	33.2	35.5	97.6	71.9	95.1	72.6	93.2	73.1	1859	2423
ADANA SAKIRPASA	34.0	37.3	98.2	72.0	95.2	73.3	93.4	74.0	1654	2665
ANKARA ETIMESGUT	12.3	16.5	94.8	64.0	91.5	63.8	88.2	62.8	5042	800
ANTALYA	35.5	37.6	100.9	68.2	98.3	68.2	95.0	68.6	1791	2321
BURSA	26.3	28.5	93.9	71.6	91.4	71.2	89.1	70.4	3445	1201
DIYARBAKIR	15.4	21.1	104.4	66.6	102.5	66.3	100.4	66.2	3851	2186
ERZURUM	–20.4	–15.1	86.8	58.8	84.1	58.5	81.0	58.1	8893	146
ESENBOGA	5.1	11.9	92.9	62.9	89.6	62.5	86.4	61.8	5705	525
ESKISEHIR	12.7	17.3	91.8	66.5	89.3	65.9	86.2	64.8	5125	632
GAZIANTEP OGUZELI	23.3	26.5	102.3	69.9	100.1	69.1	97.4	68.3	3443	2122
ISTANBUL ATATURK	28.7	31.7	89.4	70.5	87.1	70.1	84.6	69.5	3260	1285
IZMIR ADNAN MENDERES	27.4	30.1	98.8	69.2	96.5	68.7	94.2	68.2	2777	1857
IZMIR CIGLI	28.8	31.7	98.3	70.5	95.3	70.1	93.3	69.6	2433	1871
KAYSERI	3.2	9.7	93.9	63.1	91.1	62.5	87.8	61.5	5530	537
KONYA	10.5	15.5	93.5	61.8	91.1	61.6	88.0	60.8	5047	899
MALATYA ERHAC	11.5	16.2	100.4	66.8	98.3	65.5	95.3	65.0	4675	1488
SAMSUN	30.4	32.5	83.5	72.1	81.9	71.7	80.5	71.3	3398	800
VAN	8.3	11.7	84.1	65.4	82.1	65.4	80.4	65.2	6268	398
Turkmenistan										
ASHGABAT	17.9	23.2	104.4	67.0	102.3	66.8	100.2	66.7	3339	2717
Ukraine										

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	99.6%	99%	DB	MCWB	DB	MCWB	DB	MCWB	HDD	CDD 65
CHERNIHIV	–2.9	2.2	87.3	68.3	84.0	67.2	80.8	65.8	7242	355
DNEPROPETROVSK	0.2	5.2	92.4	69.8	88.8	68.6	85.6	67.4	6528	698
DONETSK	–1.9	3.4	91.7	66.9	88.1	66.4	84.6	65.5	6811	601
KHARKIV INTL	–2.8	2.1	91.0	67.4	87.1	66.3	83.9	65.4	7017	550
KHERSON	4.0	9.3	93.7	70.1	90.2	68.9	87.0	67.7	5798	793
KIEV ZHULIANY INTL	1.3	6.5	87.7	68.6	84.4	67.6	81.3	66.0	6762	430
KRYVYI RIH	0.0	5.1	91.7	69.0	88.2	67.6	85.2	66.6	6483	613
LUHANSK	–5.1	1.1	94.5	69.0	90.7	68.2	87.1	67.1	6628	666
LVIV INTL	1.3	7.0	84.5	68.1	81.5	66.6	78.7	65.1	6870	213
MARIUPOL'	3.5	8.5	89.7	70.4	86.6	70.0	84.0	69.2	6230	747
ODESA INTL	8.3	13.6	90.1	69.1	87.4	68.4	84.2	67.6	5604	743
POLTAVA	–1.7	3.5	89.3	68.3	85.9	67.1	82.8	66.0	6924	510
SIMFEROPOL INTL	9.7	14.3	92.2	68.5	89.2	67.8	85.8	66.8	5313	718
VINNYTSIA	–1.8	4.4	85.5	67.3	82.3	66.0	79.6	65.0	7092	283
ZAPORIZHIA INTL	0.4	5.4	93.3	68.4	89.8	67.6	86.3	66.6	6343	719
United Arab Emirates										
ABU DHABI INTL	53.3	55.4	112.9	73.4	110.4	73.9	107.9	74.1	49	6616
AL AIN INTL	51.7	53.6	114.8	73.0	113.1	73.0	111.4	72.9	74	7141
BATEEN	56.5	58.6	111.3	74.0	109.0	74.5	106.1	75.1	24	6695
DUBAI INTL	55.6	57.5	109.5	74.3	107.3	74.8	104.8	75.5	29	6651
SHARJAH INTL	50.3	53.1	111.6	74.7	109.5	75.1	107.5	75.7	75	6285
United Kingdom										
AUGHTON	26.8	29.3	76.0	63.3	72.3	62.1	68.9	60.6	5748	33
BINGLEY NO2	24.9	27.2	74.6	63.1	71.0	61.2	67.8	59.6	6439	17
BIRMINGHAM	23.2	26.3	79.2	64.1	75.5	62.6	72.8	61.4	5584	55
BRISTOL	26.2	28.2	76.9	63.7	73.2	62.3	69.9	61.1	5508	38

	Heating DB		Cooling DB/MCWB 0.4%		Cooling DB/MCWB 1%		Cooling DB/MCWB 2%		Heat./Cool. Degree-Days	
Station	**99.6%**	**99%**	**DB**	**MCWB**	**DB**	**MCWB**	**DB**	**MCWB**	**HDD**	**CDD 65**
BRISTOL WEATHER CENTRE	28.5	31.1	80.1	65.0	76.6	63.0	73.4	61.9	4670	106
CARDIFF WEATHER CENTRE	30.2	32.2	79.2	64.8	75.7	63.4	72.8	61.9	4554	103
CHURCH LAWFORD	24.3	26.9	79.6	65.5	75.8	63.4	72.7	61.9	5583	54
CILFYNYDD	24.5	27.3	78.0	64.3	74.2	62.2	70.8	60.9	5899	40
CROSBY	25.9	28.9	75.9	64.8	72.0	63.3	69.0	62.0	5230	34
EDINBURGH	22.6	26.2	71.9	61.9	69.3	60.9	66.6	59.1	6161	7
EMLEY MOOR	26.1	27.8	74.7	63.5	71.1	61.7	68.3	60.3	6289	25
GLASGOW	21.3	24.9	73.6	62.7	70.1	61.1	67.7	59.9	6077	12
GRAVESEND-BROADNESS	27.7	29.8	81.9	67.2	78.3	65.5	75.1	63.8	4701	134
HAWARDEN	24.2	27.2	77.2	64.8	73.6	63.4	70.7	62.0	5428	33
KENLEY	26.2	28.4	79.2	64.4	75.8	62.9	72.7	61.6	5358	76
LECONFIELD	25.3	28.2	76.8	64.8	73.6	63.2	70.9	61.7	5668	31
LEEDS BRADFORD	25.2	28.1	75.1	64.1	71.5	61.9	68.2	60.3	6182	22
LEEDS WEATHER CENTRE	27.7	29.8	79.1	64.3	75.4	62.5	72.3	61.1	5306	74
LIVERPOOL JOHN LENNON	28.0	30.0	77.0	63.9	73.4	62.4	70.2	61.3	5166	50
LONDON HEATHROW	27.5	29.8	82.9	65.4	79.2	63.9	76.1	62.7	4641	167
LONDON WC CLERKENWELL	30.9	32.8	82.8	65.0	79.2	63.6	76.0	62.3	4180	222
MANCHESTER	24.7	28.0	77.9	64.1	74.2	62.5	71.3	61.2	5550	49
NORTHOLT	25.1	27.7	82.7	65.3	79.0	63.9	75.7	62.5	4945	127
NOTTINGHAM EAST MIDLANDS	25.2	28.2	79.1	64.4	75.4	62.6	72.0	61.2	5440	67
VALLEY ANGLESEY	29.1	31.4	73.3	63.1	69.5	61.0	66.4	59.6	5165	18
Uruguay										
CARRASCO INTL	34.1	37.1	89.2	71.4	86.0	70.6	82.8	69.9	2170	853
MONTEVIDEO PRADO	37.4	39.8	88.8	72.5	86.2	71.6	83.9	71.0	1987	1032
Uzbekistan										
NAMANGAN	15.8	20.3	98.3	70.1	96.0	69.4	94.0	68.8	4023	2009

Appendix: Climatic Design Conditions for Selected Locations (*Continued*)

	Heating DB		Cooling DB/MCWB						Heat./Cool. Degree-Days	
			0.4%		1%		2%			
Station	**99.6%**	**99%**	**DB / MCWB**		**DB / MCWB**		**DB / MCWB**		**HDD / CDD 65**	
SAMARKAND	12.7	18.5	97.2	66.0	95.0	65.5	93.0	64.8	3991	1541
TASHKENT INTL	13.8	18.2	101.6	67.4	98.9	66.6	96.8	66.1	3801	1903
Venezuela										
JUAN VICENTE GOMEZ INTL	67.8	69.3	95.4	74.1	94.4	73.9	93.2	73.7	0	5934
SIMON BOLIVAR INTL	69.4	70.8	93.1	82.5	91.7	81.9	91.1	81.7	0	6077
Viet Nam										
DA NANG INTL	62.1	63.4	97.1	78.9	95.3	79.0	93.6	79.1	6	5259
HAI PHONG PHU LIEN	49.5	51.5	93.1	83.8	91.5	83.5	90.1	82.8	323	3933
HO CHI MINH TAN SON NHAT INTL	68.0	69.9	96.0	78.4	94.6	78.4	93.3	78.3	0	6476
NOI BAI INTL	49.9	51.8	96.8	81.3	94.9	81.4	93.2	81.3	313	4286
Zimbabwe										
HARARE INTL	43.2	45.3	88.0	61.6	86.1	61.5	84.3	61.4	629	1372

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