ASHRAE POCKET GUIDE

for Air Conditioning, Heating, Ventilation, Refrigeration

SI

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 This edition also features a renewed emphasis in basic design aids: content on more specialized system types has been replaced by an appendix containing climatic design data for selected worldwide locations.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

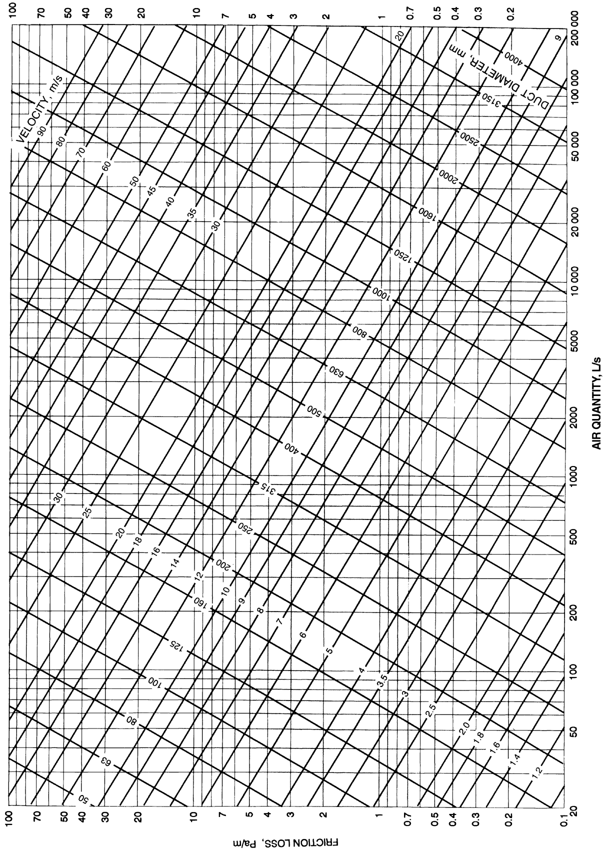


Figure 1.1 Friction Chart for Round Duct (ρ = 1.20 kg/m^3 and ε = 0.09 mm)
[2017F, Ch 21, Fig 10]

Table 1.1 Velocities vs. Velocity Pressures

Velocity V, m/s	Velocity Pressure P_v, Pa
1.0	0.6
2.0	2.4
3.0	5.4
4.0	9.6
5.0	15.1
5.5	18.3
6.0	21.7
6.5	25.5
7.0	29.5
7.5	33.9
8.0	38.5
8.5	43.5
9.0	48.8
9.5	54.3
10.0	60.2
11.0	72.9
12.0	86.7
13.0	101.8
14.0	118.0
15.0	135
17.5	184
20.0	241
22.5	305
25.0	376

$P_v = 0.602\ V^2$

Noncircular Ducts

Hydraulic diameter $D_h = 4A/P$, where A = duct area (mm) and P = perimeter (mm). 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	Sealed[b,c]		Unsealed[c]	
	Predicted Leakage Class C_L	Leakage Rate, L/(s·m^2) at 250 Pa	Predicted Leakage Class C_L	Leakage Rate, L/(s·m^2) at 250 Pa
Metal (flexible excluded)				
Round and flat oval	4	0.14	42	1.5
			(8 to 99)	(0.3 to 3.6)
Rectangular				
≤500 Pa	17	0.62	68	2.5
(both positive and negative pressures)			(17 to 155)	(0.6 to 5.6)
>500 and ≤2500 Pa	8	0.29	68	2.5
(both positive and negative pressures)			(17 to 155)	(0.6 to 5.6)
Flexible				
Metal, aluminum	11	0.40	42	1.5
			(17 to 76)	(0.6 to 2.8)
Nonmetal	17	0.62	30	1.5
			(6 to 76)	(0.2 to 2.8)
Fibrous glass				
Round	4	0.14	na	na
Rectangular	8	0.29	na	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 750 Pa, that transverse joints and longitudinal seams are sealed at 500 and 750 Pa, and that transverse joints are sealed below 500 Pa. Lower leakage classes are obtained by careful selection of joints and sealing methods.

[c] Leakage classes assigned anticipate about 0.82 joints per metre 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	Leakage Rate, L/(s·m^2) at 250 Pa
Metal		
Round	4	0.14
Flat oval	4	0.14
Rectangular	8	0.29
Flexible	8	0.29
Fibrous glass		
Round	4	0.14
Rectangular	8	0.29

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

where

Q = leakage rate, L/s/100 m^2 surface area

ΔP_s = static pressure difference, Pa 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	≤500 Pa of water	>500 Pa 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, L per metre Seam Length at 250 Pa Static Pressure	
Type of Duct/Seam		Range	Average
Rectangular	Pittsburgh lock		
	26 gage	0.015 to 0.03	0.025
	22 gage	0.0015 to 0.003	0.0025
	Button punch snaplock		
	26 gage	0.05 to 0.23	0.12
	22 gage	NA (1 test)	0.005
Round	Spiral (26 gage)	NA (1 test)	0.023
	Snaplock	0.06 to 0.22	0.17
	Grooved	0.17 to 0.28	0.19

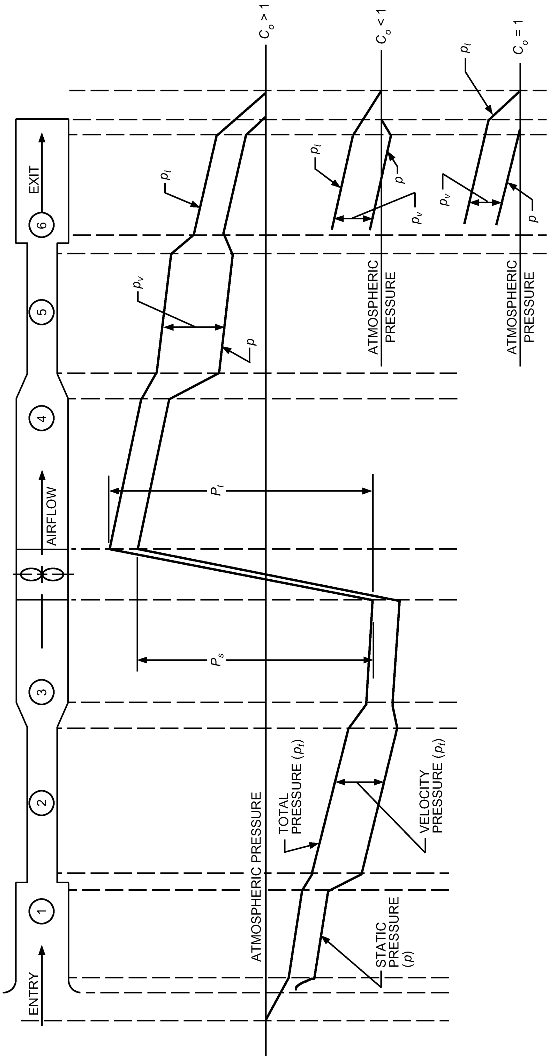


Figure 1.2 At Exit, the Fitting Coefficient C_o Affects ρ_t Loss [2017F, Ch 21, Fig 7]

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), mm																			
	100	125	150	175	200	225	250	275	300	350	400	450	500	550	600	650	700	750	800	900
100	109																			
150	133	150	164																	
200	152	172	189	204	219															
250	169	190	210	228	244	259	273													
300	183	207	229	248	266	283	299	314	328											
400	207	235	260	283	305	325	343	361	378	409	437									
500	227	258	287	313	337	360	381	401	420	455	488	518	547							
600	245	279	310	339	365	390	414	436	457	496	533	567	598	628	656					
700	261	298	331	362	391	418	443	467	490	533	573	610	644	677	708	737	765			
800	275	314	350	383	414	442	470	496	520	567	609	649	687	722	755	787	818	847	875	
900	289	330	367	402	435	465	494	522	548	597	643	686	726	763	799	833	866	897	927	984
1000	301	344	384	420	454	486	517	546	574	626	674	719	762	802	840	876	911	944	976	1037
1200	324	370	413	453	490	525	558	590	620	677	731	780	827	872	914	954	993	1030	1066	1133
1400	344	394	439	482	522	559	595	629	662	724	781	835	886	934	980	1024	1066	1107	1146	1220
1600	362	415	463	508	551	591	629	665	700	766	827	885	939	991	1041	1088	1133	1177	1219	1298
1800	379	434	485	533	577	619	660	698	735	804	869	930	988	1043	1096	1146	1195	1241	1286	1371
2000	395	453	506	555	602	646	688	728	767	840	908	973	1034	1092	1147	1200	1252	1301	1348	1438
2200	410	470	525	577	625	671	715	757	797	874	945	1013	1076	1137	1195	1251	1305	1356	1406	1501
2400	424	486	543	597	647	695	740	784	826	905	980	1050	1116	1180	1241	1299	1355	1409	1461	1561
2600	437	501	560	616	688	717	764	810	853	935	1012	1085	1154	1220	1283	1344	1402	1459	1513	1617
2800	450	516	577	634	688	738	787	834	879	964	1043	1119	1190	1259	1324	1387	1447	1506	1562	1670

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), mm																			
	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900
1000	1093																			
1100	1146	1202																		
1200	1196	1256	1312																	
1300	1244	1306	1365	1421																
1400	1289	1354	1416	1475	1530															
1500	1332	1400	1464	1526	1584	1640														
1600	1373	1444	1511	1574	1635	1693	1749													
1700	1413	1486	1555	1621	1684	1745	1803	1858												
1800	1451	1527	1598	1667	1732	1794	1854	1912	1968											
1900	1488	1566	1640	1710	1778	1842	1904	1964	2021	2077										
2000	1523	1604	1680	1753	1822	1889	1952	2014	2073	2131	2186									
2100	1558	1640	1719	1793	1865	1933	1999	2063	2124	2183	2240	2296								
2200	1591	1676	1756	1833	1906	1977	2044	2110	2173	2233	2292	2350	2405							
2300	1623	1710	1793	1871	1947	2019	2088	2155	2220	2283	2343	2402	2459	2514						
2400	1655	1744	1828	1909	1986	2060	2131	2200	2266	2330	2393	2453	2511	2568	2624					
2500	1685	1776	1862	1945	2024	2100	2173	2243	2311	2377	2441	2502	2562	2621	2678	2733				
2600	1715	1808	1896	1980	2061	2139	2213	2285	2355	2422	2487	2551	2612	2672	2730	2787	2842			
2700	1744	1839	1929	2015	2097	2177	2253	2327	2398	2466	2533	2598	2661	2722	2782	2840	2896	2952		
2800	1772	1869	1961	2048	2133	2214	2292	2367	2439	2510	2578	2644	2708	2771	2832	2891	2949	3006	3061	
2900	1800	1898	1992	2081	2167	2250	2329	2406	2480	2552	2621	2689	2755	2819	2881	2941	3001	3058	3115	3170

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

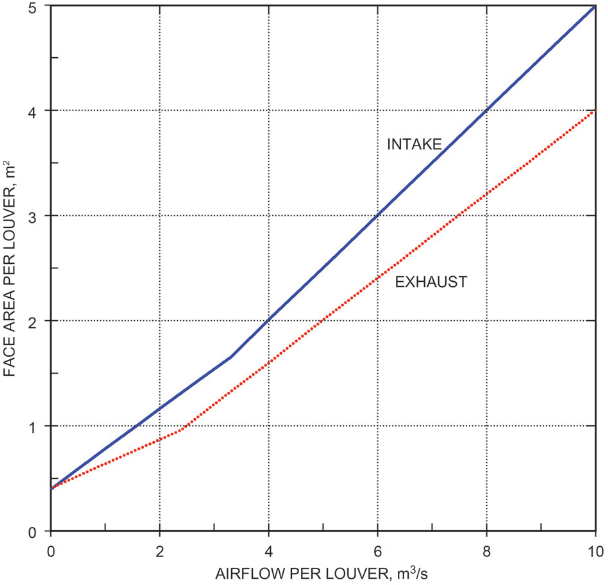
Table 1.8 Equivalent Flat Oval Duct Dimensions* [2017F, Ch 21, Tbl 3]

Circular Duct Diameter, mm	Minor Axis *a*, mm																
	70	100	125	150	175	200	250	275	300	325	350	375	400	450	500	550	600
	Major Axis *A*, mm																
125	205																
140	265	180															
160	360	235	190														
180	475	300	235	200													
200		380	290	245	215												
224		490	375	305	—	240											
250			475	385	325	290											
280				485	410	360	—	285									
315				635	525	—	—	345	325								
355				840	—	580	460	425	395	375							
400				1115	—	760	—	530	490	460	435						
450				1490	—	995	—	675	—	570	535	505					
500						1275	—	845	—	700	655	615	580				
560						1680	—	1085	—	890	820	765	720				
630								1425	—	1150	1050	970	905	810			
710										1505	1370	1260	1165	1025			
800											1800	1645	1515	1315	1170	1065	
900												2165	1985	1705	1500	1350	
1000														2170	1895	1690	
1120															2455	2170	1950
1250																2795	2495

* 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, m/s
Louvers	
Intake	
7000 cfm3300 L/s and greater	2
Less than 7000 cfm3300 L/s	See Figure 1.3
Exhaust	
5000 cfm2400 L/s and greater	2.5
Less than 5000 cfm2400 L/s	See Figure 1.3
Filters	
Panel filters	
Viscous impingement	1 to 4
Dry-type, extended-surface	
Flat (low efficiency)	Duct velocity
Pleated media (intermediate efficiency)	Up to 3.8
HEPA	1.3
Renewable media filters	
Moving-curtain viscous impingement	2.5
Moving-curtain dry media	1
Electronic air cleaners	
Ionizing type	0.8 to 1.8
Heating Coils	
Steam and hot water	2.5 to 5
	1 min., 8 max.
Electric	
Open wire	Refer to mfg. data
Finned tubular	Refer to mfg. data
Dehumidifying Coils	2 to 3
Air Washers	
Spray type	Refer to mfg. data
Cell type	Refer to mfg. data
High-velocity spray type	6 to 9



Parameters Used to Establish Figure	Intake Louver	Exhaust Louver
Minimum free area (1220 mm square test section), %	45	45
Water penetration, $mL/(m^2 \cdot 0.25\ h)$	Negligible (less than 0.3)	N/A
Maximum static pressure drop, Pa	35	60

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 $(\text{point of rating})_1 = (\text{point of rating})_2$

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	$\text{Pressure}_1 = \text{Pressure}_2{}^{c}$	$\times \left(\frac{D_1}{D_2}\right)^2 \times \left(\frac{N_1}{N_2}\right)^2$	$\times \frac{\rho_1}{\rho_2}$
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	$\text{Pressure}_1 = \text{Pressure}_2$	$\times \left(\frac{D_2}{D_1}\right)^4 \times \left(\frac{Q_1}{Q_2}\right)^2$	$\times \frac{\rho_1}{\rho_2}$
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 101.325 kPa and 20°C (1.204 kg/m^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,

$$Q_2 = 3 \text{ m}^3/\text{s and } P_{tf_2} = 228 \text{ Pa} \tag{1.2}$$

Using Fan Law 1a at Point E

$$Q_1 = 3 \times 650 / 600 = 3.25 \text{ m}^3/\text{s} \tag{1.3}$$

Using Fan Law 1b

$$P_{tf_1} = 228 \times (650/600)^2 = 268 \text{ Pa} \tag{1.4}$$

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.

$$\text{fan power, kW} = \frac{\text{L/s} \times \text{pressure difference, kPa}}{40350 \times \text{fan efficiency} \times \text{motor efficiency}} \tag{1.5}$$

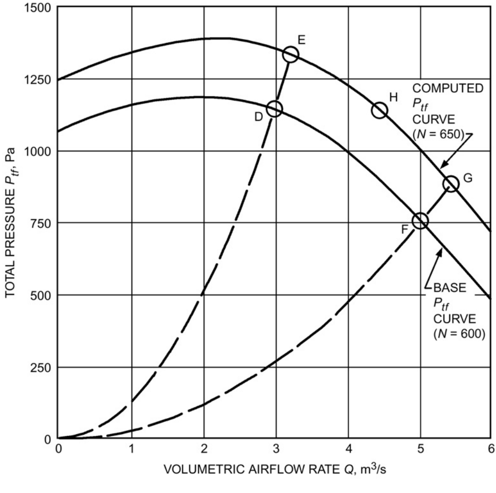


Figure 1.4 Example Calculation of Fan Laws [2016S, Ch 21, Fig 4]

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

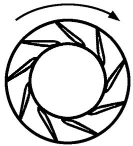
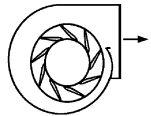
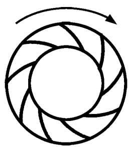
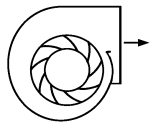
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans	Airfoil		• Blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. • Air leaves impeller at velocity less than tip speed. • For given 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)***

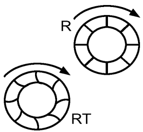
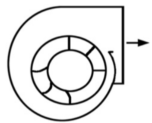
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans (continued)	Radial (R) Radial Tip (Rt)		• Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. • Curve may have a break to left of peak pressure.		• Scroll similar to 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)***

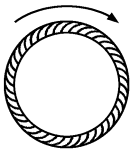
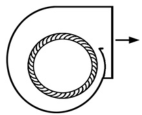
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans (continued)	Forward-Curved		• Flatter pressure curve and lower peak efficiency than the airfoil, backward-curved, and backward-inclined.		• Scroll similar to and often identical to other centrifugal fan designs. • 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)***

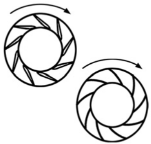
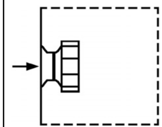
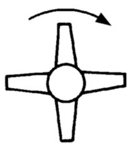
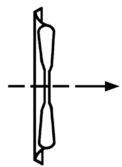
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Centrifugal Fans (continued)	Plenum/ Plug		• Plenum and plug fans typically use airfoil, backward inclined, or backward curved impellers in a single inlet configuration. Relative benefits of each impeller are the same as those 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)***

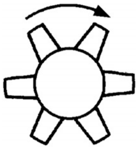
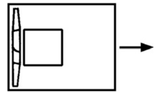
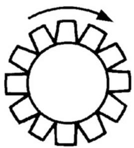
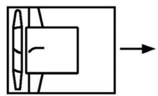
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Axial Fans (continued)	Tubeaxial		• Somewhat more efficient and capable of developing more useful static pressure than propeller fan. • Usually has 4 to 8 blades with airfoil or single- thickness cross section. • Hub is usually 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)***

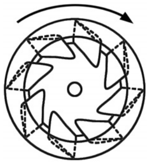
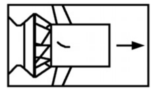
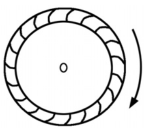
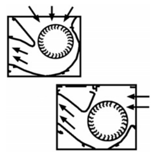
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Mixed-Flow	Mixed-Flow		• Combination of axial and centrifugal characteristics. Ideally suited in applications in which the air has to flow in or out axially. Higher pressure characteristic than axial fans.		• The majority 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)***

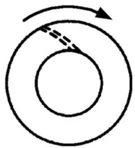
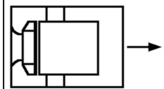
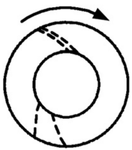
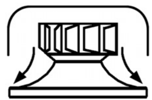
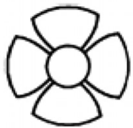
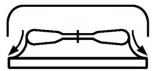
	Type	Impeller Design		Housing Design		Performance Characteristics	Applications
Other Designs	Tubular Centrifugal		• Performance similar to backward-curved fan except capacity and pressure are lower. • Lower efficiency than backward-curved fan. • Performance curve may have a dip to the left of peak pressure.		• 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

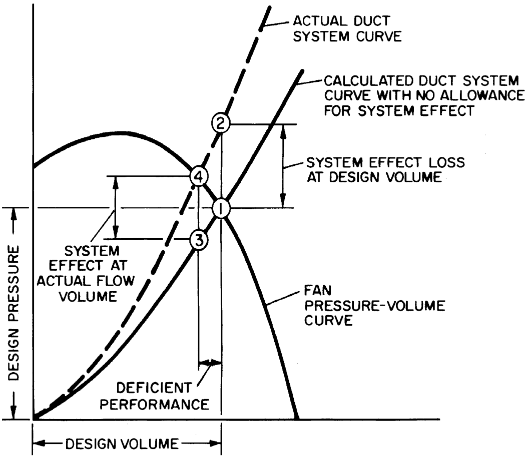


Figure 1.5 Deficient Fan/System Performance

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.

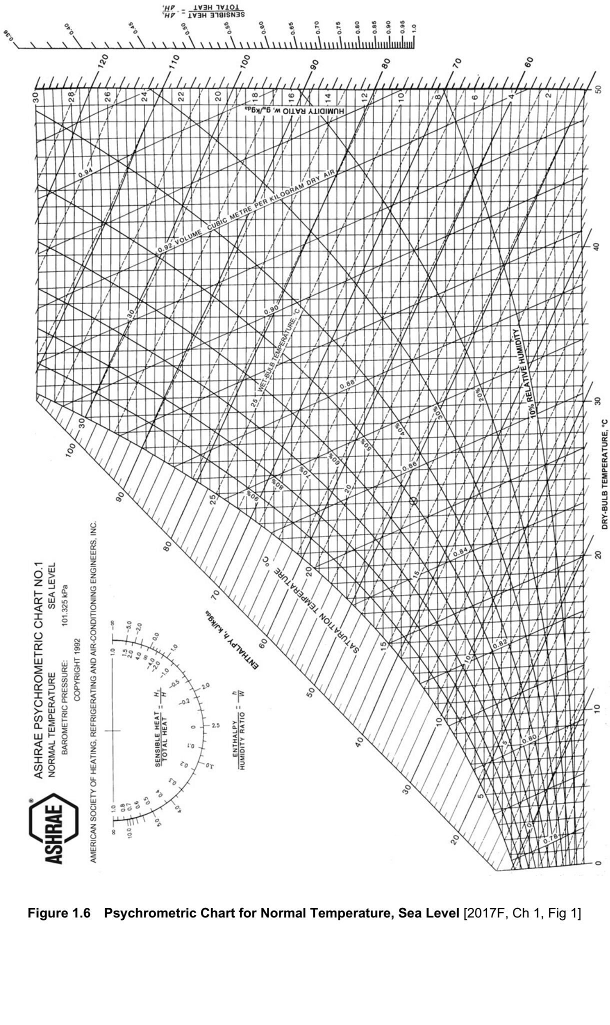


Figure 1.6 Psychrometric Chart for Normal Temperature, Sea Level [2017F, Ch 1, Fig 1]

Air-Conditioning Processes

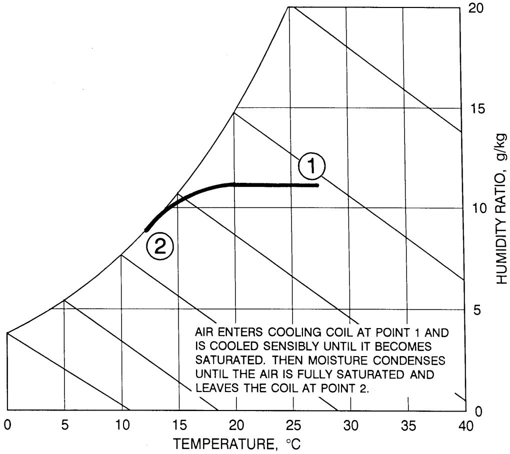
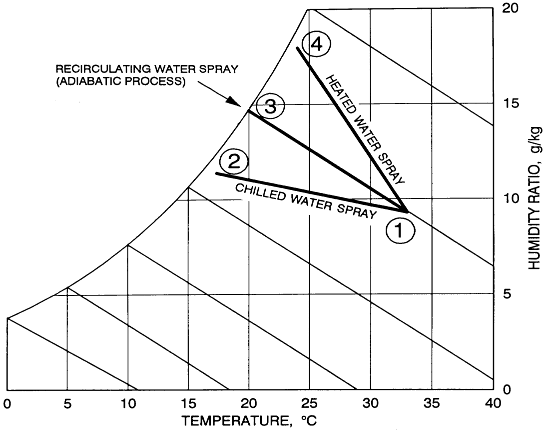
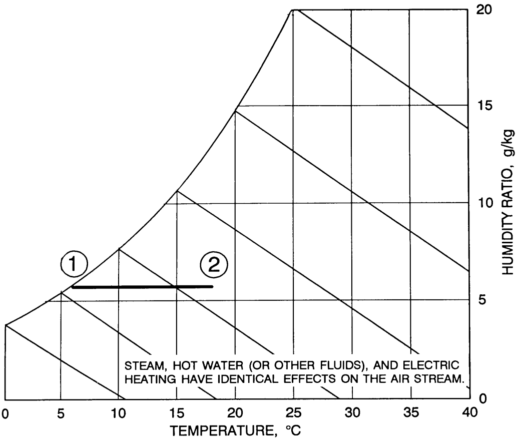


Figure 1.7 Direct Expansion or Chilled Water Cooling and Dehumidification



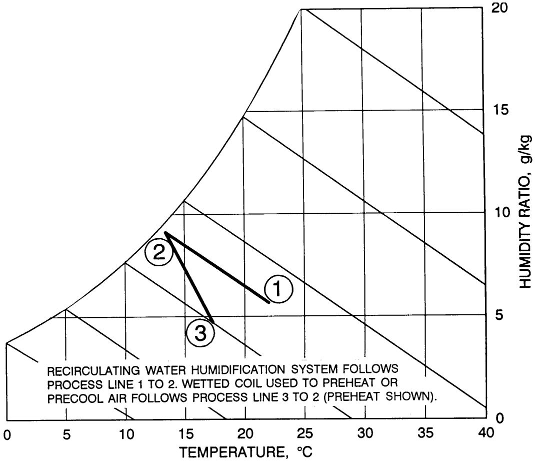
Water Spray Cooling

Figure 1.8 Direct Expansion or Chilled Water Cooling and Dehumidification



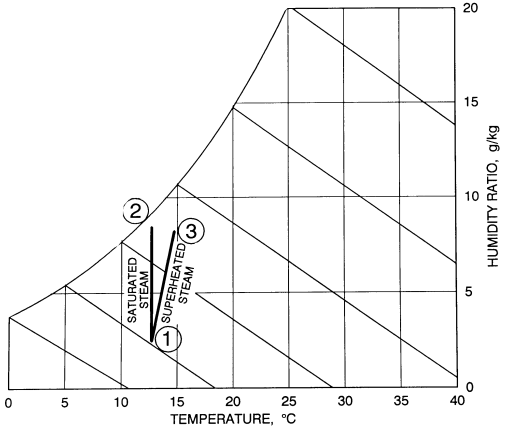
Steam, Hot-Water, and Electric Heating

Figure 1.9 Direct Expansion or Chilled Water Cooling and Dehumidification



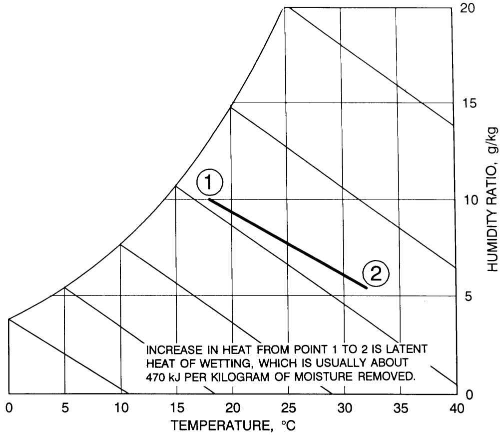
Humidification

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, 101.325 kPa
[2017F, Ch 1, Tbl 2, Abridged]

Temp., °C	Specific Enthalpy, kJ/kg_{da}	Temp., °C	Specific Enthalpy, kJ/kg_{da}
–60	–60.325	26	80.801
–55	–55.280	27	85.289
–50	–50.222	28	89.979
–45	–45.144	29	94.882
–40	–40.031	30	100.009
–35	–34.859	31	105.372
–30	–29.593	32	110.985
–25	–24.181	33	116.860
–20	–18.542	34	123.013
–10	–6.070	35	129.458
–8	–3.282	36	136.213
–6	–0.356	37	143.294
–4	2.728	38	150.720
–2	5.995	39	158.510
0	9.475	40	166.685
2	12.981	45	214.169
4	16.696	50	275.349
6	20.644	55	355.144
8	24.853	60	460.880
10	29.354	70	803.464
12	34.181	80	1541.765
14	39.371	90	3867.556
16	44.966		
18	51.011		
20	57.558		
21	61.037		
22	64.663		
23	68.444		
24	72.388		
25	76.503		

Table 1.13 Standard Atmospheric Data for Altitudes to 10 000 m
[2017F, Ch 1, Tbl 1]

Altitude, m	Temperature, °C	Pressure, kPa
–500	18.2	107.478
0	15.0	101.325
500	11.8	95.461
1000	8.5	89.875
1500	5.2	84.556
2000	2.0	79.495
2500	–1.2	74.682
3000	–4.5	70.108
4000	–11.0	61.640
5000	–17.5	54.020
6000	–24.0	47.181
7000	–30.5	41.061
8000	–37.0	35.600
9000	–43.5	30.742
10 000	–50	26.436

Source: Adapted from NASA (1976).

At sea level, standard temperature is 15°C; standard barometric pressure is 101.325 kPa. The temperature is assumed to decrease linearly with increasing altitude throughout the troposphere (lower atmosphere), and to be constant in the lower reaches of the stratosphere. The lower atmosphere is assumed to consist of dry air that behaves as a perfect gas. Gravity is also assumed constant at the standard value, 9.806 65 m/s^2.

The values in Table 1.13 may be calculated from Equation 1.6:

$$p = 101.325(1 - 2.25577 \times 10^{-5} Z)^{5.2559} \tag{1.6}$$

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.