The *Journal of Cotton Science 23:182–217 (2019)*
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ENGINEERING & GINNING

Cotton Gin Pneumatic Conveying Systems

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ABSTRACT

Cotton gins use air to move seed cotton, lint, cottonseed, and trash through conveying pipes. In gins, pneumatic conveying systems are the principal means of moving material from one processing stage to another throughout the entire ginning plant. Further, material drying or moisture restoration can be accomplished by heating or humidifying the conveying air. Pneumatic systems are a critical and fundamental component of cotton ginning. Cotton gins use large quantities of air for pneumatic conveying. It is common for a gin to use 4,248 m^3 (150,000 ft^3) or more of air per minute in its various material conveying systems. Because the density of dry standard air is approximately 1.2 kg/m^3 (0.075 lb/ft^3), a typical gin using 4,248 m^3/min (150,000 ft^3/min) of air moves 305,860 kg (675,000 lb) of air per hour. This mass of air per hour is approximately 1.5 times the total mass of material handled per hour. Typically, more than 60 to 65% of the total electrical power consumed by a cotton gin is attributed to moving material pneumatically. Properly taking air measurements, determining air flow requirements, sizing conveying pipes, sizing fans to generate required air flow rates, and accounting for specific machinery air requirements are essential to maximizing machine utilization, minimizing energy costs, and decreasing system downtime. This update of the Cotton Ginners Handbook provides current technical information on cotton gin pneumatic systems. It draws heavily on previous versions of the Cotton Ginners Handbook (Stedronsky 1964; McCaskill et al., 1977; Baker et al., 1994) and the knowledge and experience of current and past instructors of the Air Systems classes from the National Cotton Ginners' Association Gin Schools.

AIR FLOW BASICS

Air Velocity and Flow. Air velocity is defined as the speed of the moving air stream and is usually expressed in units of meters per second, m/s (feet per minute, ft/min or fpm). The volume of air flowing through a conveying duct per unit time is defined as the air flow rate, which is normally expressed in terms of cubic meters per minute, m^3/min (cubic feet per minute, ft^3/min or cfm). Air flow rate is determined by multiplying the air velocity by the cross-sectional area of the duct (Fig. 1). The following equations demonstrate the relationship between air flow rate, air velocity, and cross-sectional area:

$$Q = VA \quad \text{or} \quad V = Q/A \quad \text{or} \quad A = Q/V \qquad \text{(Eq. 1)}$$

where:
Q = air flow rate, m^3/min (ft^3/min)
V = air velocity, m/s × 60 (ft/min)
A = cross-sectional area, m^2 (ft^2).

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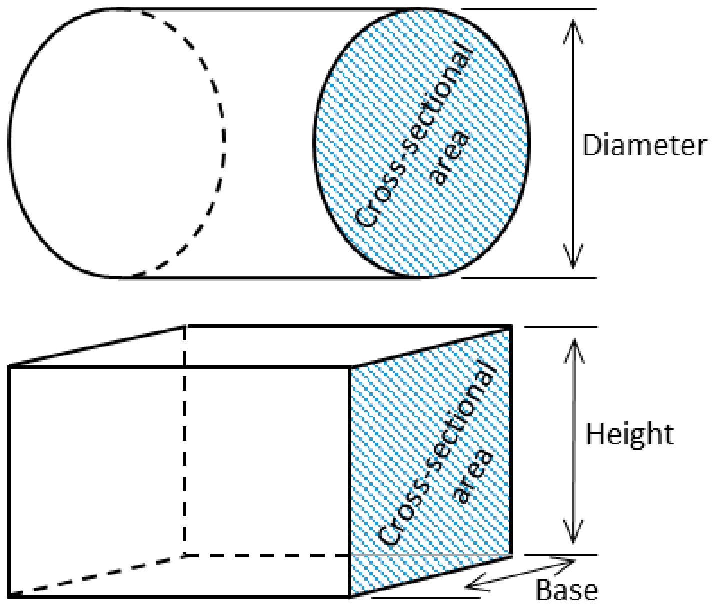


Figure 1. Pipe and duct cross-sectional area.

For a round pipe the cross-sectional area (Fig. 1) is calculated using

$$A = \pi D^2 / 4 \qquad \text{(Eq. 2)}$$

where:

D = pipe diameter, mm/1000 or m (in./12 or ft)

π = 3.1416.

For a square or rectangular duct the cross-sectional area (Fig. 1) is calculated using

$$A = bh \qquad \text{(Eq. 3)}$$

where:

b = width of the duct, mm/1000 or m (in./12 or ft)

h = height of the duct, mm/1000 or m (in./12 or ft).

An air flow lookup table is provided in Table 1 [Table 1a for International System of Units (SI) and Table 1b for U.S. customary units (USC)] that has air flow rates for a range of air velocities in the second column and round pipe diameters across the top row. The corresponding circumference and cross-sectional area for each pipe diameter are in the second and third rows and the first column, velocity pressure, is discussed in the Air Measurements section. An example of the use of Table 1 and Equations 1 and 2 follows:

Example 1. For air moving inside a 450-mm (18-in.) diameter pipe at 20 m/s (4,000 ft/min), determine the air flow rate.

Using Table 1, move straight down the column for 450-mm (18-in.) pipe diameter and move straight right across the row for 20 m/s (4,000 ft/min). The corresponding air flow rate is located where the column and row intersect: 191 m^3/min (7,069 ft^3/min).

Table 1 shows that a 450-mm diameter pipe has cross-sectional area = 0.159 m^2 (18-in. diameter pipe has 1.767 ft^2 cross-sectional area). Using Equation 2 also gives the pipe cross-sectional area:

$$A = \frac{3.1416 \times \left(\dfrac{450 \text{ mm}}{1000 \text{ mm / m}} \right)^2}{4} = 0.159 \text{ m}^2$$

or

$$A = \frac{3.1416 \times \left(\dfrac{18 \text{ in.}}{12 \text{ in. / ft}} \right)^2}{4} = 1.767 \text{ ft}^2$$

Then using Equation 1 to find the air flow rate:

$$Q = 20\ \text{m/s} \times 60\ \text{s/min} \times 0.159\ \text{m}^2 = 191\text{m}^3/\text{min}$$

or

$$Q = 4000\ \text{ft/min} \times 1.767\ \text{ft}^2 = 7068\text{ft}^3/\text{min}$$

Note: The International System of Units (SI) and U.S. customary units (USC) used in the examples are not always exact conversions. Values used for details like nominal pipe sizes are adjusted to be approximately equivalent and make sense in the context.

Table 1a. Air flow rates for air velocity or velocity pressure and round pipe sizes[z]—SI units

Diameter (mm)		200	225	250	275	300	325	350	375	400	425	450	475	500	525	550	575	600	625	650
Circumference (mm)		628	707	785	864	942	1021	1100	1178	1257	1335	1414	1492	1571	1649	1728	1806	1885	1963	2042
Cross-sectional area (m^2)		0.031	0.040	0.049	0.059	0.071	0.083	0.096	0.110	0.126	0.142	0.159	0.177	0.196	0.216	0.238	0.260	0.283	0.307	0.332
Vel. Pres.[y] (Pa)	Vel.[x] (m/s)	Air flow rate[w] (m^3/min)																		
18	5	9.42	11.9	14.7	17.8	21.2	24.9	28.9	33.1	37.7	42.6	47.7	53.2	58.9	64.9	71.3	77.9	84.8	92.0	99.5
26	6	11.3	14.3	17.7	21.4	25.4	29.9	34.6	39.8	45.24	51.1	57.3	63.8	70.7	77.9	85.5	93.5	102	110	119
35	7	13.2	16.7	20.6	24.9	29.7	34.8	40.4	46.4	52.78	59.6	66.8	74.4	82.5	90.9	99.8	109	119	129	139
46	8	15.1	19.1	23.6	28.5	33.9	39.8	46.2	53.0	60.32	68.1	76.3	85.1	94.2	104	114	125	136	147	159
59	9	17.0	21.5	26.5	32.1	38.2	44.8	52.0	59.6	67.86	76.6	85.9	95.7	106	117	128	140	153	166	179
72	10	18.8	23.9	29.5	35.6	42.4	49.8	57.7	66.3	75.40	85.1	95.4	106	118	130	143	156	170	184	199
88	11	20.7	26.2	32.4	39.2	46.7	54.8	63.5	72.9	82.94	93.6	105	117	130	143	157	171	187	202	219
104	12	22.6	28.6	35.3	42.8	50.9	59.7	69.3	79.5	90.48	102	115	128	141	156	171	187	204	221	239
122	13	24.5	31.0	38.3	46.3	55.1	64.7	75.0	86.1	98.02	111	124	138	153	169	185	203	221	239	259
142	14	26.4	33.4	41.2	49.9	59.4	69.7	80.8	92.8	106	119	134	149	165	182	200	218	238	258	279
163	15	28.3	35.8	44.2	53.5	63.6	74.7	86.6	99.4	113	128	143	159	177	195	214	234	254	276	299
185	16	30.2	38.2	47.1	57.0	67.9	79.6	92.4	106	121	136	153	170	188	208	228	249	271	295	319
209	17	32.0	40.6	50.1	60.6	72.1	84.6	98.1	113	128	145	162	181	200	221	242	265	288	313	338
235	18	33.9	42.9	53.0	64.1	76.3	89.6	104	119	136	153	172	191	212	234	257	280	305	331	358
261	19	35.8	45.3	56.0	67.7	80.6	94.6	110	126	143	162	181	202	224	247	271	296	322	350	378
290	20	37.7	47.7	58.9	71.3	84.8	99.5	115	133	151	170	191	213	236	260	285	312	339	368	398
319	21	39.6	50.1	61.9	74.8	89.1	105	121	139	158	179	200	223	247	273	299	327	356	387	418
351	22	41.5	52.5	64.8	78.4	93.3	110	127	146	166	187	210	234	259	286	314	343	373	405	438
383	23	43.4	54.9	67.7	82.0	97.5	114	133	152	173	196	219	245	271	299	328	358	390	423	458
417	24	45.2	57.3	70.7	85.5	102	119	139	159	181	204	229	255	283	312	342	374	407	442	478
453	25	47.1	59.6	73.6	89.1	106	124	144	166	188	213	239	266	295	325	356	390	424	460	498
490	26	49.0	62.0	76.6	92.7	110	129	150	172	196	221	248	276	306	338	371	405	441	479	518
528	27	50.9	64.4	79.5	96.2	115	134	156	179	204	230	258	287	318	351	385	421	458	497	538
568	28	52.8	66.8	82.5	99.8	119	139	162	186	211	238	267	298	330	364	399	436	475	515	557
609	29	54.7	69.2	85.4	103	123	144	167	192	219	247	277	308	342	377	413	452	492	534	577
652	30	56.5	71.6	88.4	107	127	149	173	199	226	255	286	319	353	390	428	467	509	552	597
696	31	58.4	74.0	91.3	110	131	154	179	205	234	264	296	330	365	403	442	483	526	571	617
742	32	60.3	76.3	94.2	114	136	159	185	212	241	272	305	340	377	416	456	499	543	589	637

Table 1a. (*Continued*)

Diameter (mm)		675	700	725	750	800	850	900	950	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500
Circumference (mm)		2121	2199	2278	2356	2513	2670	2827	2985	3142	3299	3456	3613	3770	3927	4084	4241	4398	4555	4712
Cross-sectional area (m^2)		0.358	0.385	0.413	0.442	0.503	0.567	0.636	0.709	0.785	0.866	0.950	1.039	1.131	1.227	1.327	1.431	1.539	1.651	1.767
Vel. Pres.[y] (Pa)	Vel.[x] (m/s)	Air flow rate[w] (m^3/min)																		
18	5	107	115	124	133	151	170	191	213	236	260	285	312	339	368	398	429	462	495	530
26	6	129	139	149	159	181	204	229	255	283	312	342	374	407	442	478	515	554	594	636
35	7	150	162	173	186	211	238	267	298	330	364	399	436	475	515	557	601	647	694	742
46	8	172	185	198	212	241	272	305	340	377	416	456	499	543	589	637	687	739	793	848
59	9	193	208	223	239	271	306	344	383	424	468	513	561	611	663	717	773	831	892	954
72	10	215	231	248	265	302	340	382	425	471	520	570	623	679	736	796	859	924	991	1060
88	11	236	254	272	292	332	375	420	468	518	571	627	686	746	810	876	945	1016	1090	1166
104	12	258	277	297	318	362	409	458	510	565	623	684	748	814	884	956	1031	1108	1189	1272

Table 1a. (*Continued*)

Diameter (mm)		675	700	725	750	800	850	900	950	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500
Circumference (mm)		2121	2199	2278	2356	2513	2670	2827	2985	3142	3299	3456	3613	3770	3927	4084	4241	4398	4555	4712
Cross-sectional area (m^2)		0.358	0.385	0.413	0.442	0.503	0.567	0.636	0.709	0.785	0.866	0.950	1.039	1.131	1.227	1.327	1.431	1.539	1.651	1.767
Vel. Pres.[y] (Pa)	Vel.[x] (m/s)	Air flow rate[w] (m^3/min)																		
122	13	279	300	322	345	392	443	496	553	613	675	741	810	882	957	1035	1116	1201	1288	1378
142	14	301	323	347	371	422	477	534	595	660	727	798	872	950	1031	1115	1202	1293	1387	1484
163	15	322	346	372	398	452	511	573	638	707	779	855	935	1018	1104	1195	1288	1385	1486	1590
185	16	344	369	396	424	483	545	611	680	754	831	912	997	1086	1178	1274	1374	1478	1585	1696
209	17	365	393	421	451	513	579	649	723	801	883	969	1059	1154	1252	1354	1460	1570	1684	1802
235	18	386	416	446	477	543	613	687	766	848	935	1026	1122	1221	1325	1434	1546	1663	1783	1909
261	19	408	439	471	504	573	647	725	808	895	987	1083	1184	1289	1399	1513	1632	1755	1882	2015
290	20	429	462	495	530	603	681	763	851	942	1039	1140	1246	1357	1473	1593	1718	1847	1982	2121
319	21	451	485	520	557	633	715	802	893	990	1091	1197	1309	1425	1546	1672	1804	1940	2081	2227
351	22	472	508	545	583	664	749	840	936	1037	1143	1254	1371	1493	1620	1752	1889	2032	2180	2333
383	23	494	531	570	610	694	783	878	978	1084	1195	1311	1433	1561	1694	1832	1975	2124	2279	2439
417	24	515	554	594	636	724	817	916	1021	1131	1247	1368	1496	1629	1767	1911	2061	2217	2378	2545
453	25	537	577	619	663	754	851	954	1063	1178	1299	1425	1558	1696	1841	1991	2147	2309	2477	2651
490	26	558	600	644	689	784	885	992	1106	1225	1351	1483	1620	1764	1914	2071	2233	2401	2576	2757
528	27	580	623	669	716	814	919	1031	1148	1272	1403	1540	1683	1832	1988	2150	2319	2494	2675	2863
568	28	601	647	694	742	844	953	1069	1191	1319	1455	1597	1745	1900	2062	2230	2405	2586	2774	2969
609	29	623	670	718	769	875	987	1107	1233	1367	1507	1654	1807	1968	2135	2310	2491	2679	2873	3075
652	30	644	693	743	795	905	1021	1145	1276	1414	1559	1711	1870	2036	2209	2389	2576	2771	2972	3181
696	31	666	716	768	822	935	1055	1183	1318	1461	1611	1768	1932	2104	2283	2469	2662	2863	3071	3287
742	32	687	739	793	848	965	1090	1221	1361	1508	1663	1825	1994	2171	2356	2548	2748	2956	3170	3393

[z] Pipe size relationships: Diameter (mm) = circumference (mm) ÷ 3.1416, Cross-sectional area (m^2) = 3.1416 × [diameter (mm)/1000]^2/4

[y] Velocity pressure based on pipe centerline velocity pressure reading at standard air (density = 1.2 kg/m^3) conditions @ 21°C and 101.3 kPa

[x] Velocity $(\text{m/s}) = 1.175\sqrt{\text{velocity pressure}\,(\text{Pa})}$

[w] Air flow rate (m^3/min) = velocity (m/s) × cross-sectional area (m^2) × 60

Table 1b. Air flow rates for air velocity or velocity pressure and round pipe sizes[z]—USC units

Diameter (in.)		8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Circumference (in.)		25.13	28.27	31.42	34.56	37.70	40.84	43.98	47.12	50.27	53.41	56.55	59.69	62.83	65.97	69.12	72.26	75.40	78.54	81.68
Cross-sectional area (ft^2)		0.349	0.442	0.545	0.660	0.785	0.922	1.069	1.227	1.396	1.576	1.767	1.969	2.182	2.405	2.640	2.885	3.142	3.409	3.687
Vel. Pres.[y] (in. H_2O)	Vel.[x] (ft/min)	Air flow rate[w] (ft^3/min)																		
0.08	1000	349	442	545	660	785	922	1069	1227	1396	1576	1767	1969	2182	2405	2640	2885	3142	3409	3687
0.11	1200	419	530	654	792	942	1106	1283	1473	1676	1892	2121	2363	2618	2886	3168	3462	3770	4091	4424
0.15	1400	489	619	764	924	1100	1290	1497	1718	1955	2207	2474	2757	3054	3367	3696	4039	4398	4772	5162
0.19	1600	559	707	873	1056	1257	1475	1710	1963	2234	2522	2827	3150	3491	3848	4224	4616	5027	5454	5899
0.24	1800	628	795	982	1188	1414	1659	1924	2209	2513	2837	3181	3544	3927	4330	4752	5193	5655	6136	6637
0.30	2000	698	884	1091	1320	1571	1844	2138	2454	2793	3153	3534	3938	4363	4811	5280	5770	6283	6818	7374
0.36	2200	768	972	1200	1452	1728	2028	2352	2700	3072	3468	3888	4332	4800	5292	5808	6348	6912	7499	8111
0.43	2400	838	1060	1309	1584	1885	2212	2566	2945	3351	3783	4241	4725	5236	5773	6336	6925	7540	8181	8849
0.51	2600	908	1149	1418	1716	2042	2397	2779	3191	3630	4098	4595	5119	5672	6254	6864	7502	8168	8863	9586
0.59	2800	977	1237	1527	1848	2199	2581	2993	3436	3910	4414	4948	5513	6109	6735	7391	8079	8796	9545	10324
0.68	3000	1047	1325	1636	1980	2356	2765	3207	3682	4189	4729	5301	5907	6545	7216	7919	8656	9425	10227	11061
0.77	3200	1117	1414	1745	2112	2513	2950	3421	3927	4468	5044	5655	6301	6981	7697	8447	9233	10053	10908	11798
0.87	3400	1187	1502	1854	2244	2670	3134	3635	4172	4747	5359	6008	6694	7418	8178	8975	9810	10681	11590	12536
0.97	3600	1257	1590	1963	2376	2827	3318	3848	4418	5027	5675	6362	7088	7854	8659	9503	10387	11310	12272	13273
1.08	3800	1326	1679	2073	2508	2985	3503	4062	4663	5306	5990	6715	7482	8290	9140	10031	10964	11938	12954	14011
1.20	4000	1396	1767	2182	2640	3142	3687	4276	4909	5585	6305	7069	7876	8727	9621	10559	11541	12566	13635	14748
1.32	4200	1466	1856	2291	2772	3299	3871	4490	5154	5864	6620	7422	8270	9163	10102	11087	12118	13195	14317	15485
1.45	4400	1536	1944	2400	2904	3456	4056	4704	5400	6144	6936	7775	8663	9599	10583	11615	12695	13823	14999	16223

Diameter (in.)		8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26
Circumference (in.)		25.13	28.27	31.42	34.56	37.70	40.84	43.98	47.12	50.27	53.41	56.55	59.69	62.83	65.97	69.12	72.26	75.40	78.54	81.68
Cross-sectional area (ft^2)		0.349	0.442	0.545	0.660	0.785	0.922	1.069	1.227	1.396	1.576	1.767	1.969	2.182	2.405	2.640	2.885	3.142	3.409	3.687
Vel. Pres.[y] (in. H_2O)	Vel.[x] (ft/min)	Air flow rate[w] (ft^3/min)																		
1.59	4600	1606	2032	2509	3036	3613	4240	4917	5645	6423	7251	8129	9057	10036	11064	12143	13272	14451	15681	16960
1.73	4800	1676	2121	2618	3168	3770	4424	5131	5890	6702	7566	8482	9451	10472	11545	12671	13849	15080	16362	17698
1.88	5000	1745	2209	2727	3300	3927	4609	5345	6136	6981	7881	8836	9845	10908	12026	13199	14426	15708	17044	18435
2.03	5200	1815	2297	2836	3432	4084	4793	5559	6381	7261	8197	9189	10239	11345	12507	13727	15003	16336	17726	19172
2.19	5400	1885	2386	2945	3564	4241	4977	5773	6627	7540	8512	9543	10632	11781	12989	14255	15580	16965	18408	19910
2.35	5600	1955	2474	3054	3696	4398	5162	5986	6872	7819	8827	9896	11026	12217	13470	14783	16157	17593	19090	20647
2.53	5800	2025	2562	3163	3828	4555	5346	6200	7118	8098	9142	10249	11420	12654	13951	15311	16734	18221	19771	21385
2.70	6000	2094	2651	3272	3960	4712	5531	6414	7363	8378	9458	10603	11814	13090	14432	15839	17311	18850	20453	22122
2.89	6200	2164	2739	3382	4092	4869	5715	6628	7609	8657	9773	10956	12207	13526	14913	16367	17889	19478	21135	22859
3.07	6400	2234	2827	3491	4224	5027	5899	6842	7854	8936	10088	11310	12601	13963	15394	16895	18466	20106	21817	23597

Table 1b. (*Continued*)

Diameter (in.)		27	28	29	30	32	34	36	38	40	42	44	46	48	50	52	54	56	58	60
Circumference (in.)		84.82	87.96	91.11	94.25	100.5	106.8	113.1	119.4	125.7	131.9	138.2	144.5	150.8	157.1	163.4	169.6	175.9	182.2	188.5
Cross-sectional area (ft^2)		3.976	4.276	4.587	4.909	5.585	6.305	7.069	7.876	8.727	9.621	10.56	11.54	12.57	13.64	14.75	15.90	17.10	18.35	19.63
Vel. Pres.[y] (in. H_2O)	Vel.[x] (ft/min)	Air flow rate[w] (ft^3/min)																		
0.08	1000	3976	4276	4587	4909	5585	6305	7069	7876	8727	9621	10559	11541	12566	13635	14748	15904	17104	18348	19635
0.11	1200	4771	5131	5504	5890	6702	7566	8482	9451	10472	11545	12671	13849	15080	16362	17698	19085	20525	22017	23562
0.15	1400	5567	5986	6422	6872	7819	8827	9896	11026	12217	13470	14783	16157	17593	19090	20647	22266	23946	25687	27489
0.19	1600	6362	6842	7339	7854	8936	10088	11310	12601	13963	15394	16895	18466	20106	21817	23597	25447	27367	29356	31416
0.24	1800	7157	7697	8256	8836	10053	11349	12723	14176	15708	17318	19007	20774	22619	24544	26546	28628	30788	33026	35343
0.30	2000	7952	8552	9174	9817	11170	12610	14137	15752	17453	19242	21118	23082	25133	27271	29496	31809	34208	36696	39270
0.36	2200	8747	9407	10091	10799	12287	13871	15551	17327	19199	21166	23230	25390	27646	29998	32446	34989	37629	40365	43197
0.43	2400	9543	10263	11009	11781	13404	15132	16965	18902	20944	23091	25342	27698	30159	32725	35395	38170	41050	44035	47124
0.51	2600	10338	11118	11926	12763	14521	16393	18378	20477	22689	25015	27454	30007	32673	35452	38345	41351	44471	47704	51051
0.59	2800	11133	11973	12843	13744	15638	17654	19792	22052	24435	26939	29566	32315	35186	38179	41294	44532	47892	51374	54978
0.68	3000	11928	12828	13761	14726	16755	18915	21206	23627	26180	28863	31678	34623	37699	40906	44244	47713	51313	55043	58905
0.77	3200	12723	13683	14678	15708	17872	20176	22619	25203	27925	30788	33790	36931	40212	43633	47194	50894	54734	58713	62832
0.87	3400	13519	14539	15596	16690	18989	21437	24033	26778	29671	32712	35901	39239	42726	46360	50143	54075	58154	62382	66759
0.97	3600	14314	15394	16513	17671	20106	22698	25447	28353	31416	34636	38013	41548	45239	49087	53093	57256	61575	66052	70686
1.08	3800	15109	16249	17430	18653	21223	23959	26861	29928	33161	36560	40125	43856	47752	51814	56043	60436	64996	69722	74613
1.20	4000	15904	17104	18348	19635	22340	25220	28274	31503	34907	38485	42237	46164	50265	54542	58992	63617	68417	73391	78540
1.32	4200	16700	17959	19265	20617	23457	26481	29688	33078	36652	40409	44349	48472	52779	57269	61942	66798	71838	77061	82467
1.45	4400	17495	18815	20183	21598	24574	27742	31102	34654	38397	42333	46461	50780	55292	59996	64891	69979	75259	80730	86394
1.59	4600	18290	19670	21100	22580	25691	29003	32515	36229	40143	44257	48573	53089	57805	62723	67841	73160	78679	84400	90321
1.73	4800	19085	20525	22017	23562	26808	30264	33929	37804	41888	46181	50684	55397	60319	65450	70791	76341	82100	88069	94248
1.88	5000	19880	21380	22935	24544	27925	31525	35343	39379	43633	48106	52796	57705	62832	68177	73740	79522	85521	91739	98175
2.03	5200	20676	22235	23852	25525	29042	32786	36757	40954	45379	50030	54908	60013	65345	70904	76690	82702	88942	95408	102102
2.19	5400	21471	23091	24769	26507	30159	34047	38170	42529	47124	51954	57020	62321	67858	73631	79639	85883	92363	99078	106029
2.35	5600	22266	23946	25687	27489	31276	35308	39584	44104	48869	53878	59132	64630	70372	76358	82589	89064	95784	102748	109956
2.53	5800	23061	24801	26604	28471	32393	36569	40998	45680	50615	55803	61244	66938	72885	79085	85539	92245	99205	106417	113883
2.70	6000	23856	25656	27522	29452	33510	37830	42412	47255	52360	57727	63355	69246	75398	81812	88488	95426	102625	110087	117810
2.89	6200	24652	26512	28439	30434	34627	39091	43825	48830	54105	59651	65467	71554	77911	84539	91438	98607	106046	113756	121737
3.07	6400	25447	27367	29356	31416	35744	40352	45239	50405	55851	61575	67579	73862	80425	87266	94387	101788	109467	117426	125664

[z] **Pipe size relationships: Diameter (in.) = circumference (in.) ÷ 3.1416, Cross-sectional area (ft^2) = 3.1416 × [diameter (in.)/12]^2/4**

[y] **Velocity pressure based on pipe centerline velocity pressure reading at standard air (density = 0.075 lb/ft^3) conditions @ 70 °F and 29.92-in. Hg**

[x] **Velocity $(\text{ft/min}) = 3650\sqrt{\text{velocity pressure } (\text{in. } H_2O)}$.**

[w] **Air flow rate (ft^3/min) = velocity (ft/min) × cross-sectional area (ft^2)**

Pressure. Moving air exerts a pressure on objects in the airstream, for example, seed cotton in a pipe. This pressure, called the velocity pressure, is proportional to the air density and the square of the velocity. The faster the air stream is moving, the more force it exerts on the material being conveyed. In the gin, air velocity is determined from the velocity pressure of the moving airstream. Velocity pressure is always exerted in the direction of flow and is always positive (Fig. 2).

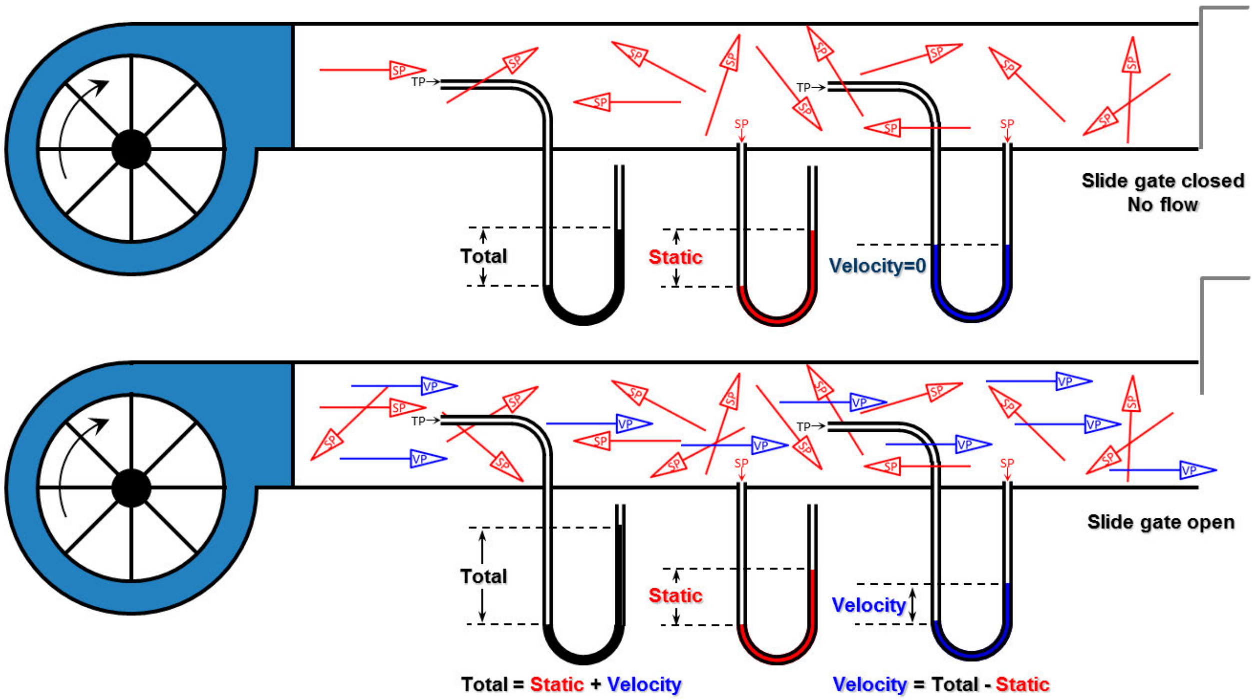


Figure 2. Total (TP), static (SP), and velocity pressure (VP) relationships illustrated.

The air pressure that the fan must supply to overcome the resistance to air flow through the ductwork and components is called static pressure. This static pressure is equal in all directions and can exist in the absence of air flow (Fig. 2). A common example of static pressure is the pressure inside an inflated balloon. In a pneumatic conveying system, static pressure is measured perpendicular to the air flow and is usually negative before the fan inlet and positive after the fan outlet.

The total pressure in a conveying pipe is the sum of the velocity and static pressures and is either positive or negative (Fig. 2).

The velocity, static, and total pressures for air flow are normally measured in units of Pascals (Pa) or inches of water column (in. H_2O) with a differential pressure gauge. In metric units, 10 Pa is approximately equivalent to 1 millimeter of water column (mm H_2O). In USC units, 27.7 in. H_2O is approximately equivalent to 1 pound per square inch (lb/in.^{2} or psi).

AIR FLOW REQUIREMENTS

Efficient operation of pneumatic conveying systems depends on several factors, and air velocity is the primary factor. Table 2 shows the recommended air velocities for efficient conveying of various materials in cotton gin systems. Typical air velocity in seed cotton conveying pipes should be approximately 23 m/s (4,500 ft/min), whereas it should be approximately 8.9 m/s (1,750 ft/min) for lint systems. If the conveying air velocity is too low, the conveyed material can settle out of the air stream and choke the conveying pipe, resulting in operational downtime. Excessively high air velocities should be avoided as they waste energy, create excessive wear on the machinery, and can damage the material being conveyed. Therefore, it is critical to maintain air velocities within recommended ranges throughout a gin's conveying pipes or ducts. When no material is conveyed, such as in the pipe from the burner inlet to the point where seed cotton is fed into the conveying system, lower air velocities can be used. The lower velocity reduces the static pressure and energy required.

Air flow rate is also an important factor and depends on the material conveyed, pipe or duct size, and the purpose of the gin system. Typical air requirements to convey materials in cotton gin systems are shown in Table 3. The recommended air volume for conveying seed cotton is 0.9 to 1.25 m^3 per kg (15-20 ft^3 per lb) of seed cotton for systems used strictly for conveying. This air volume might have to be increased for systems that handle wet, trashy cotton or when air leakage is greater than normal. Also, cotton drying systems can be designed to provide 1.25 to 2.5 m^3 of air per kg of cotton (20-40 ft^3/lb) to enhance drying effectiveness.

Table 2. Recommended material conveying air velocities for cotton gin systems

Material & System	Air velocity	
	(m/s)	(ft/min)
Seed cotton in telescope pipe	27.9 - 30.5	5,500 - 6,000
Seed cotton in conveying pipes	20.3 - 25.4	4,000 - 5,000
Seed cotton in tower dryers	10.2 - 12.7	2,000 - 2,500
Seed in small-pipe systems	20.3 - 25.4	4,000 - 5,000
Hulls and trash	20.3 - 25.4	4,000 - 5,000
Lint cotton	7.6 - 10.2	1,500 - 2,000

Table 3. Typical air requirements for conveying material in cotton gin systems

Seed Cotton System	Air requirement	
	(m^3 per kg material)	(ft^3 per lb material)
Unloading	1.25 - 1.56	20 - 25
Conveying	0.9 - 1.25	15 - 20
Drying	1.25 - 2.5	20 - 40
Lint Systems [z]	(m^3/min per m width)	(ft^3/min per ft width)
Gin Stand and Lint Cleaner Brushes	93	1000

[z] Machinery dependent

The following example demonstrates using Tables 1, 2, and 3 and Equations 1 and 2 to determine the required air flow rate and pipe size to convey a specified amount of material.

Example 2. For a seed cotton conveying system to handle 30 bales per hour, determine the required air flow rate and pipe size. Assume 590 kg (1,300 lb) of seed cotton per bale.

The mass of seed cotton conveyed per minute:

$$\frac{30\text{ bales}}{\text{hr}}\times\frac{590\text{ kg}}{\text{bale}}\times\frac{\text{hr}}{60\text{ min}}=295\text{ kg / min}$$

or

$$\frac{30\text{ bales}}{\text{hr}}\times\frac{1300\text{ lb}}{\text{bale}}\times\frac{\text{hr}}{60\text{ min}}=650\text{ lb / min}$$

From Table 3 for seed cotton conveying, use 1.25 m^3 per kg (20 ft^3 per lb) of seed cotton. The required air flow rate:

$$Q=\frac{295\text{ kg}}{\text{min}}\times\frac{1.25\text{ m}^3}{\text{kg}}=369\text{ m}^3\text{ / min}$$

or

$$Q=\frac{650\text{ lb}}{\text{min}}\times\frac{20\text{ ft}^3}{\text{lb}}=13000\text{ ft}^3\text{ / min}$$

The recommended air velocity for seed cotton conveying from Table 2 is 20.3 to 25.4 m/s (4,000-5,000 ft/min), selecting a value between the higher and lower velocities, we select 23 m/s (4,500 ft/min). Using Table 1, the needed pipe diameter is found by starting at 23 m/s (4,400 ft/min) in the left column. Then, move right to the air flow rate at or just below the required 369-m^3/min (13,000 ft^3/min) air flow rate, which is 358 m^3/min (12,695 ft^3/min) that corresponds to 575-mm (23-in.) pipe diameter. The actual air velocity for the required 369-m^3/min (13,000-ft^3/in) air flow rate with a 575-mm (23-in.) diameter pipe is determined from Equation 1 and the pipe area = 0.260 m^2 (2.885 ft^2) from Table 1.

$$V = \frac{369\ \text{m}^3/\text{min}}{0.260\ \text{m}^2 \times 60\ \text{s}/\text{min}} = 23.6\ \text{m}/\text{s}$$

or

$$V = \frac{13000\ \text{ft}^3/\text{min}}{2.885\ \text{ft}^2} = 4506\ \text{ft}/\text{min}$$

Machine Air Flow Requirements. Much of the equipment in a cotton gin, although designed for certain material capacities, also has recommended air flow requirements to operate efficiently. This is particularly true for equipment handling lint after the ginning point, such as gin stands, lint cleaners, and condensers. Insufficient air flow at machines can cause incomplete doffing and promote choke-ups that can lead to reduced efficiency, increased down time, and higher incidence of fires. Each manufacturer has recommended air flow rates for the cotton gin equipment.

AIR MEASUREMENTS

Many problems at the gin can be attributed to air flow issues. The only reliable way to determine the air flow of a system is to take measurements of air velocity and static pressure. Although pneumatic systems in gins vary greatly (no two installations are identical), procedures for making air measurements are basically the same and can be made with a relatively simple kit containing the following items (Fig. 3):

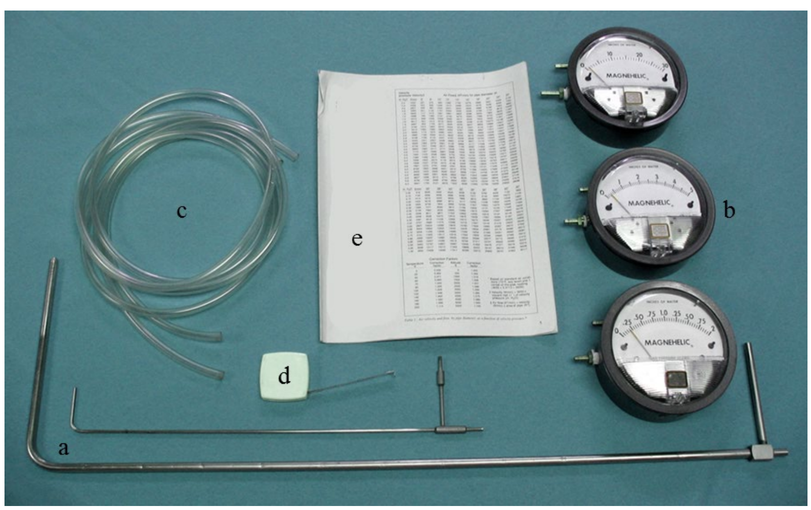


Figure 3. Items necessary for making air flow measurements in cotton gin pipes: a) 8.0-mm (5⁄16-in.) diameter and 3.2-mm (1⁄8-in.) diameter Pitot tubes; b) differential pressure gauges; c) plastic hoses to connect the Pitot tube to the gauges d) measuring tape for determining pipe diameters; and e) tables of air flow rates for various pipe diameters and velocity pressures.

1. 3.2-mm (1⁄8-in.) diameter by 305-mm (12-in.) long Pitot tube for pipes up to 600-mm (24-in.) diameter, and 8.0-mm (5⁄16-in.) diameter Pitot tube up to 1,500-mm (60-in.) long for larger diameter pipes. Duct diameter should be at least 30 times the Pitot tube diameter (never use an 8.0-mm (5⁄16-in.) diameter Pitot tube in a duct smaller than 240 mm [10 in.]);

2. Differential pressure gauges similar to the Magnehelic® differential pressure gauges (Dwyer Instruments, Michigan City, IN) shown in Fig. 3. Three gauges with ranges of 0 to 500 Pa (0-2 in. H_2O) for most velocity pressure measurements, and 0 to 1.5 kPa and 0 to 8 kPa (0-5 in. H_2O and 0-30 in. H_2O) for static pressure measurements are recommended. A liquid manometer (not shown in Figure) can be used instead of a pressure gauge, but it is more awkward to manage and more fragile than a gauge;
3. Two rubber or plastic hoses to connect the Pitot tube to the gauge;
4. Measuring tape for determining pipe diameters; and
5. Table 1 with air flow rates for air velocity or velocity pressure, and round pipe sizes.

Although not shown in Fig. 3, a tachometer for measuring fan speeds and an ammeter to check fan motor electrical current should also be included in an air measurement kit.

Velocity Pressure Measurement. Pressure measurements are ideally taken at least one and one-half pipe diameters upstream and eight and one-half pipe diameters downstream from any disturbance such as a fan, valve, or elbow (Fig. 4). This means that if the diameter of the pipe is 600 mm (24 in.), then the air measurement must be taken at least 0.9-m (3-ft) upstream and 5.1-m (17-ft) downstream from a disturbance to get a reliable measurement. This is to ensure that the air stream is uniform across the cross-section of the pipe. If this ideal situation is not possible, a traverse of air readings might have to be conducted to obtain an accurate measurement (see Appendix A).

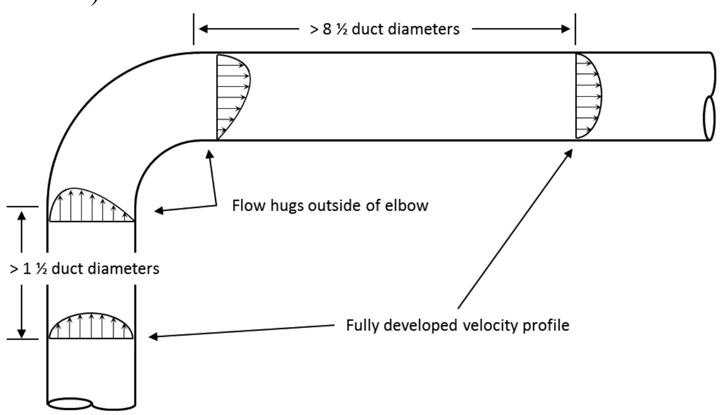


Figure 4. Distance up- and down-stream of disturbance for fully developed velocity profile and for reliable air flow measurements.

Use the following procedures for taking velocity pressure and static pressure readings:

1. Start all the fans for the system. If the system is a push/pull system, ensure that both fans are operating. Pressure measurements with a Pitot tube cannot be made in pipes that are loaded with seed cotton, lint, or trash. Stop the material flow, allow material to clear the system, and then take the measurements. Air measurements for drying systems should be taken with burners turned off. From time to time, systems can be operated without burners, thus air readings need to be taken "cold" to ensure that adequate air flow is maintained. Also, careful attention should be paid to not plug up or damage the Pitot tube, and to avoid dangerous situations while measuring.
2. Connect the gauge or manometer to the Pitot tube with the two lengths of flexible hose. The differential pressure gauge tap marked "high pressure" should be connected with tubing to the total pressure connection that is an extension of the main tube of the Pitot tube, and the "low pressure" tap on the gauge should be connected to the static pressure connection that branches off perpendicularly to the main tube of the Pitot tube (Fig. 5). The pressure gauge should be held vertically and should be zeroed with the set screw on the front.

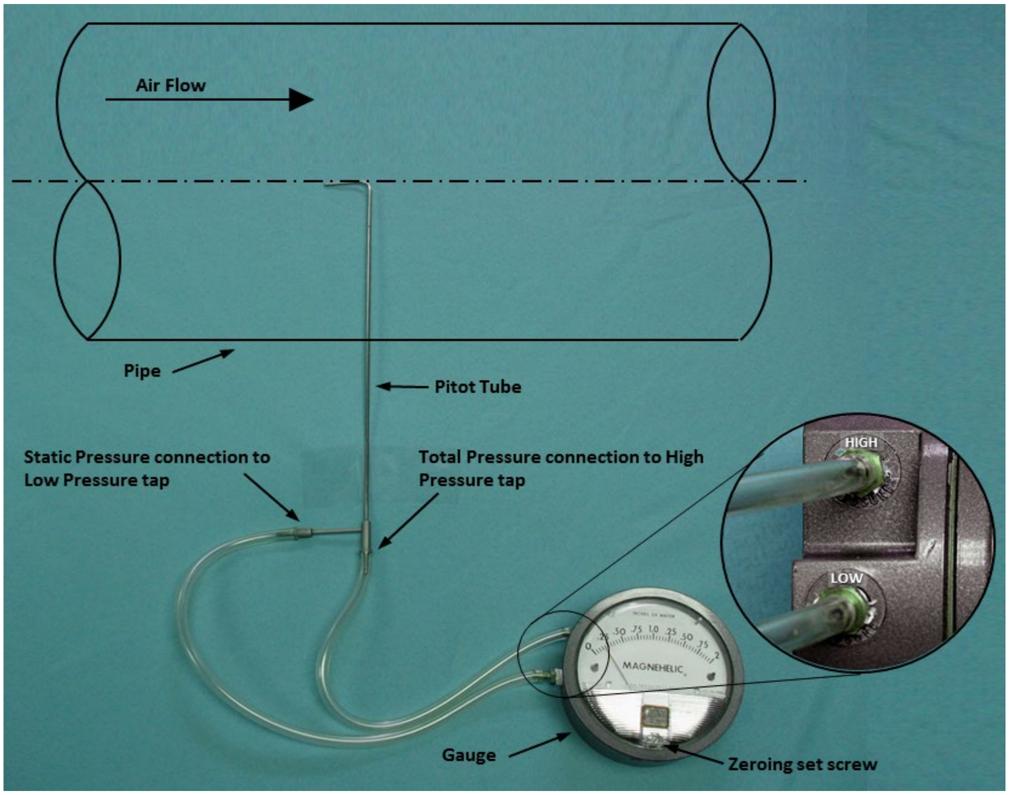


Figure 5. Measuring centerline velocity pressure with a Pitot tube and differential pressure gauge.

3. A hole approximately 1.5 mm (1⁄16 in.) larger than the Pitot tube should be used to allow insertion of the Pitot tube. When drilling holes in conveying pipes and ducts avoid making burs that will snag material. Insert the Pitot tube with tip pointed directly into the air flow (the static pressure connection pointing upstream and the stem at a right angle to the pipe ensures that the Pitot tube points directly into the air flow). Check for uniform air velocity by observing that the velocity pressure is near zero at the near wall, reaches a maximum at the pipe center, and falls to near zero at the far wall. If the velocity pressure does not follow this pattern, then the flow is not uniform and a different straight section of pipe should be used for the measurement or a traverse should be made.
4. Keeping the Pitot tube tip pointed into the air flow at the centerline of the pipe, determine the velocity pressure by reading the differential pressure gauge. Because readings usually fluctuate, take several measurements or readings and estimate the average.

After the centerline velocity pressure reading has been obtained (step 4), the air velocity (m/s [ft/min]) and air flow rate (m^3/min [ft^3/min]) that the fan is delivering can be estimated from Table 1. The lefthand column of this table gives velocity pressure, and the second column gives the air velocity that corresponds to the velocity pressure, regardless of pipe diameter. The body of the table gives the air flow rate for a given velocity pressure and various commonly used pipe diameters. The air velocity and flow rate for velocity pressures and pipe sizes also can be calculated from the equations given in the footnotes of Table 1, Equations 1 and 2, and pipe areas given in Table 1. The following example demonstrates determining air velocity and flow rate using Table 1.

Example 3: If the centerline velocity pressure reading for air flowing in a 300-mm (12in.) diameter pipe is 400 Pa (1.6 in. H_2O), what is the air velocity and air flow rate in the pipe?

In the far left column of Table 1, 400 Pa velocity pressure is approximately halfway between 383 and 417 Pa. In next column over, the same two rows show 23 and 24 m/s air velocity. So, the estimated air velocity for 400 Pa would be 23.5 m/s. Then, proceeding right to the column corresponding to 300-mm pipe diameter, the average of the air flow rates from the two rows (97.5 and 102 m^3/min) is 99.7 m^3/min.

For USC units, the method would be similar. In the far left column of Table 1, 1.6 in. H_2O velocity pressure is close to 1.59 in. H_2O. The same row in the next column over shows 4,600-ft/min air velocity. Then, proceeding right to the column for 12-in. pipe diameter, find 3,613-ft^3/min air flow rate.

Table 1 gives the velocity and air flow rate based on one centerline reading of velocity pressure for standard air density (1.2 kg/m^3 [0.075 lb/ft^3]) at sea level and standard conditions (21 °C and 101.3 kPa [70 °F and 29.92-in. Hg]). If velocity pressure readings are taken for air at other than standard temperature and elevation, for example, near Lubbock, Texas (990-m [3,250-ft] elevation) in summer (38 °C [100 °F]), then air temperature and elevation correction factors from Table 4 should be used to correct the air velocity and air flow rates.

To find and apply the correction factors, first look up the actual air temperature in Table 4 and find the corresponding correction factor in the column to the right. Next, look up the actual elevation in Table 4 and find the corresponding correction factor in the column to the right. Then, multiply each factor by the air velocity and air flow rate determined from the air measurement procedure detailed previously. The following example shows how to obtain air temperature and elevation correction factors and apply them to air velocity and flow rate measurements.

Table 4. Air velocity and air flow rate correction factors for Pitot tube measurements at nonstandard conditions

Air Temperature	Factor	Elevation[z]	Factor	Air Temperature	Factor	Elevation[z]	Factor
(°C)		(m)		(°F)		(ft)	
-15	0.937	0[y]	1.000	0	0.932	0[y]	1.000
-10	0.946	100	1.006	10	0.942	500	1.009
-5	0.955	200	1.012	20	0.952	1000	1.018
0	0.963	300	1.018	30	0.962	1500	1.028
5	0.972	400	1.024	40	0.971	2000	1.037
10	0.981	500	1.031	50	0.981	2500	1.046
15	0.989	600	1.036	60	0.990	3000	1.056
21[y]	1.000	700	1.042	70[y]	1.000	3500	1.066
25	1.006	800	1.048	80	1.009	4000	1.075
30	1.015	900	1.055	90	1.019	4500	1.086
35	1.023	1000	1.062	100	1.028	5000	1.095
40	1.032	1100	1.068	110	1.037	5500	1.106
45	1.040	1200	1.074	120	1.046		
50	1.048	1300	1.081	130	1.055		
55	1.056	1400	1.088	140	1.064		
60	1.064	1500	1.094	150	1.073		
65	1.072	1600	1.100	160	1.082		
70	1.080	1700	1.108	170	1.090		
75	1.088			180	1.098		
80	1.095			190	1.106		
85	1.102			200	1.114		
90	1.109			225	1.137		
95	1.117			250	1.157		
100	1.125			275	1.177		
110	1.141			300	1.197		
120	1.155			325	1.217		
140	1.184			350	1.236		
160	1.213			375	1.255		
180	1.240			400	1.273		
200	1.267			425	1.292		
220	1.294			450	1.310		
240	1.320			475	1.328		
260	1.345			500	1.345		

[z] Above mean sea level.

[y] Standard condition.

Example 4: If the system in **Example 3** above was handling air at 35 °C (95 °F) and was located at an elevation of 900 m (3,000 ft) above sea level, what are the correction factors for temperature and elevation and corrected air velocity and flow rate?

First, in Table 4, the temperature correction factor to the right of 35 °C is 1.023. For USC units, 95 °F falls between 90 and 100 °F. So, the temperature correction factor is between 1.019 and 1.028, approximately 1.023. Next, the elevation correction factor to the right of 900-m (3,000-ft) elevation is 1.055 (1.056). Then, multiplying the temperature and elevation factors by the measured air velocity and flow rate results in the actual velocity and flow rate:

$$\text{Air velocity} = 23.5\ \text{m/s} \times 1.023 \times 1.055 = 25.4\ \text{m/s}$$
$$\text{Air flow rate} = 99.7\ \text{m}^3/\text{min} \times 1.023 \times 1.055 = 107.6\ \text{m}^3/\text{min}$$

or

$$\text{Air velocity} = 4600\ \text{ft/min} \times 1.023 \times 1.056 = 4969\ \text{fpm}$$
$$\text{Air flow rate} = 3613\ \text{ft}^3/\text{min} \times 1.023 \times 1.056 = 3903\ \text{ft}^3/\text{min}$$

Static Pressure Measurement. To measure the static pressure, disconnect the total pressure tubing from the Pitot tube and differential pressure gauge, and leave the static pressure outlet connected (the static pressure outlet on the Pitot tube is perpendicular to the main tube, see Fig. 5). For measuring static pressure on the outlet side (push side) of the fan, the tubing from the Pitot tube static pressure outlet should be connected to high pressure tap on the differential pressure gauge. For static pressure measurements on the inlet side (suction side) of the fan, connect the tubing to the low pressure tap. Insert the Pitot tube into the pipe in the same manner as described in step 3 of the velocity pressure measurement procedure above and read the static pressure.

The static pressure reading also can be taken by removing the tubing from the Pitot tube and holding the end of the flexible hose over the test hole in the pipe to make a good seal. The two static pressure readings should be nearly the same.

For fans with piping on both the inlet and outlet, measure the static pressure on the opposite sides of the fan, and add the absolute value of the two static pressure readings to obtain the overall system static pressure. However, an outlet reading is all that is necessary on a push fan with no piping on the inlet, and only an inlet reading is necessary on a pull fan with no piping on the outlet. Air readings, air flow, and static pressure, can be taken at whatever point in the system where information is needed, such as at troublesome points in a piping system or at separator or machinery inlets and outlets to determine leakage.

FANS

Air flows from high-pressure to low-pressure areas. In a pneumatic conveying system, a fan is used to create the pressure differential or static pressure required to induce air flow. The fan creates pressure to overcome the flow resistance of the piping, elbows, and fittings; and accelerate and lift conveyed material. Two types of fans, centrifugal and axial flow, are used in cotton gins, but the majority are centrifugal fans.

Centrifugal Fans. The centrifugal fan resembles a paddle wheel and consists of an impeller or blast wheel that rotates within a castiron or sheet metal scroll-type housing (Fig. 6). Air enters through the inlet to the impeller parallel to the shaft, is accelerated by the blades, and discharges perpendicular to the impeller shaft. Pressure is generated by the centrifugal force on the rotating air and by the velocity of the air leaving the impeller tip. The fan housings used in ginning applications usually have a single air inlet and a single outlet. Generally, a centrifugal fan delivers lower air volumes at higher pressures up to approximately 7,500 Pa (30 in. H_2O). This type of fan is commonly used for unloading systems, push/pull pre-cleaning and drying systems, and for waste handling.

The main types of centrifugal fan impellers or blast wheels used in cotton gins are radial- or straight-blade and backward-inclined. Straight-blade wheels are usually open wheels with 6 to 12 flat blades (increased number of blades increases static pressure capability: 6 blades ≈ 4,500 Pa [18 in. H_2O], 10 blades ≈ 7,500 Pa

[30 in. H_2O]) and are rugged, simple, and designed to handle coarse and abrasive materials in the air stream, like gin waste (Fig. 7). They are generally noisier than other types of centrifugal fan impellers and power will rise continually to free delivery, which means they can overload. Straight-blade fans are also less efficient than other centrifugal fan designs, typically converting a maximum of 70% of the input mechanical energy to the energy of the moving air stream. Backward-inclined impellers have from 8 to 12 flat blades that are inclined away from the direction of rotation (Fig. 8). They are more efficient by approximately 80% and quieter than radial blade fan, but maximum static pressure is only approximately 2,500 Pa (10 in. H_2O). Their horsepower increases to maximum as air flow increases and then drops off toward free delivery, which means they are non-overloading. Backward-inclined wheels are designed for clean or lightly contaminated air with little fiber, such as burner push fans or battery condensers.

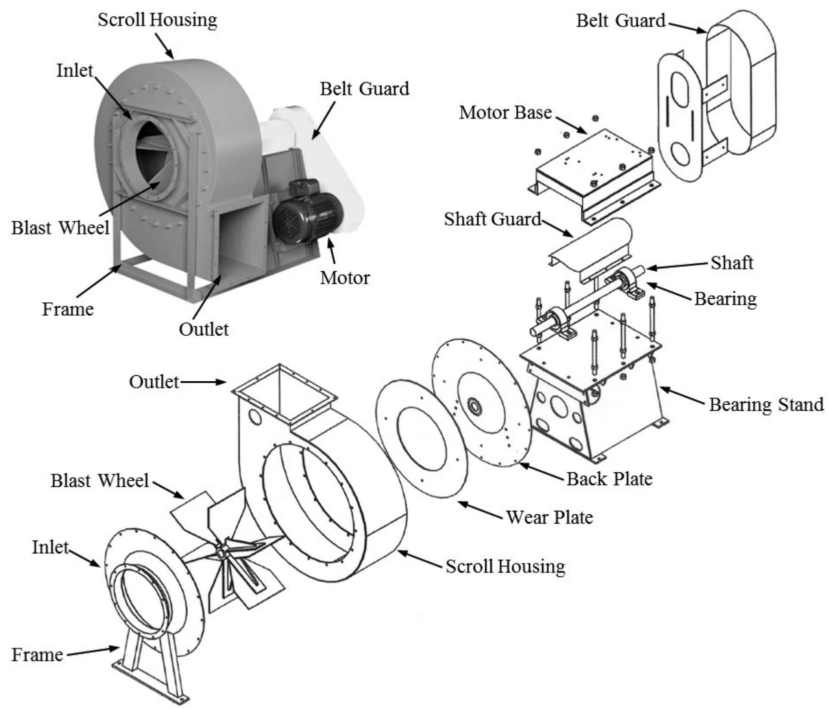


Figure 6. Centrifugal fan components (Exploded view courtesy of Smith Fans Inc., Lamesa, TX).

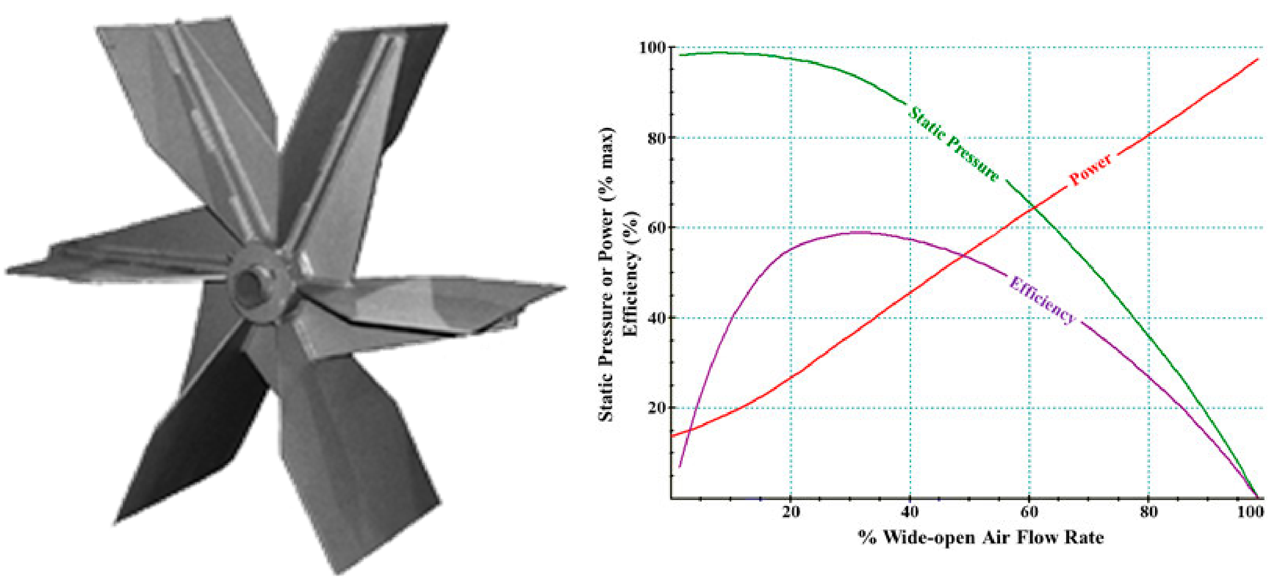


Figure 7. Radial or straight blade centrifugal fan wheel and typical performance curve.

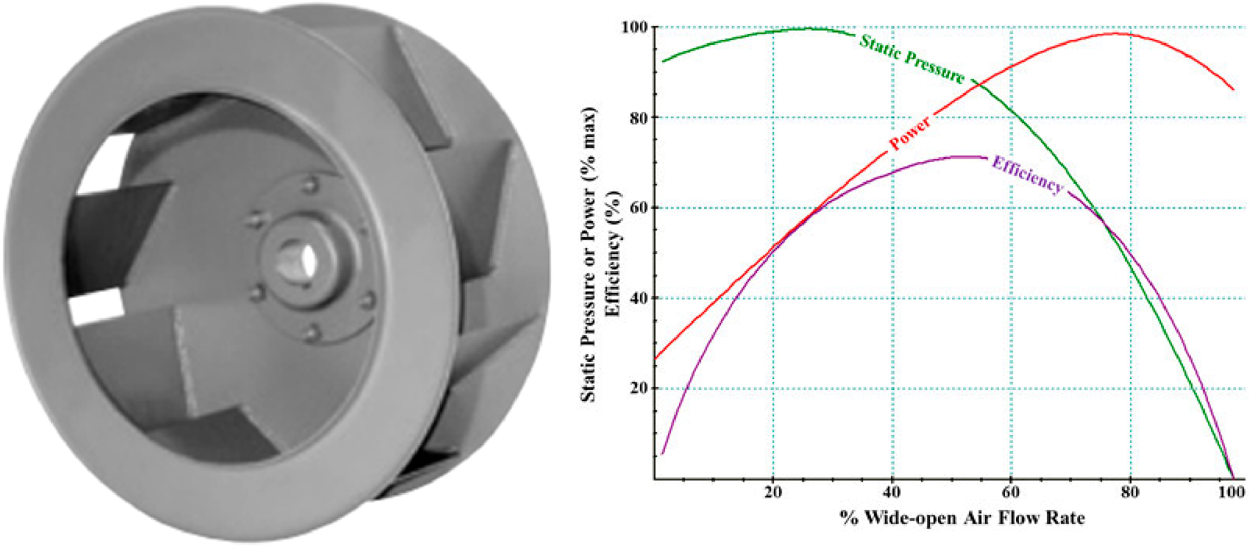


Figure 8. Backward inclined centrifugal fan wheel and typical performance curve.

There are other types of centrifugal fan wheels that can be used in gins for specific purposes (Fig. 9). The wool wheel [Fig. 9(a)] is a straight-blade wheel with a back plate that prevents stringy or fibrous materials from wrapping around the blades and is sometimes used to handle and fluff cotton lint, especially for roller ginning. Rembert fans [Fig. 9(b)] with perforated flat disks or cones permit the material to pass through the fan housing without damage from the wheel. This type of fan can be used for transferring seed cotton from one location to another without using a separator. Remberttype fans normally are only approximately 35% efficient. Both wool wheel and Rembert-type fans have fan curves similar to that in Fig. 7.

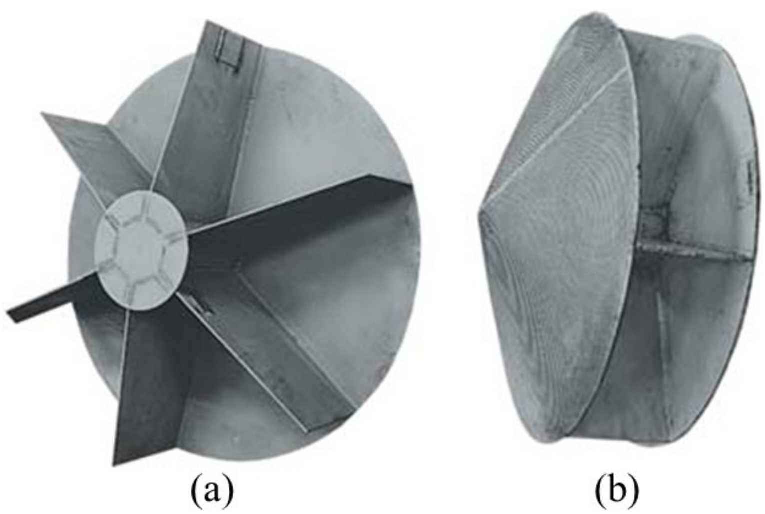


Figure 9. Wool wheel (a) and Rembert wheel (b) (Courtesy of Lummus Corporation, Savannah, GA).

Size designations for centrifugal fans can be confusing because each manufacturer uses their own descriptive terms. Many centrifugal fans are designated simply by a number, No. 30, or a number that can indicate size, 923 RBO with 23-in. diameter inlet. Table 5 shows data for some common modern gin fans. Maximum operating speeds of centrifugal fans vary usually from approximately 1,200 to 3,000 rpm, but for safety, the blast wheel speed should not exceed manufacturer's recommendations. It is a good practice to contact the manufacturer and request specific information on the fan in question.

Table 5. Common modern[z] gin fan data

Fan Model	Blast Wheel Type	Blast Wheel Dia. mm (in.)	Inlet Dia. mm (in.)	Outlet Dimensions mm (in.)	Max Speed rpm	Max Static Pressure Pa (in. H_2O)
Lummus Corporation - Savannah, GA						
913 RBO	Straight Blade	574 (22.6)	330 (13)	283×316 (11.1×12.4)	2822	4977 (20)
915 RBO	Straight Blade	663 (26.1)	381 (15)	325×363 (12.8×14.3)	2485	4977 (20)
919 RBO	Straight Blade	838 (33)	483 (19)	410×459 (16.1×18.1)	1951	4977 (20)
921 RBO	Straight Blade	927 (36.5)	533 (21)	452×508 (17.8×20)	1778	5972 (24)
923 RBO	Straight Blade	1016 (40)	584 (23)	499×559 (19.6×22)	1623	5972 (24)
926 RBO	Straight Blade	1146 (45.1)	660 (26)	562×630 (22.1×24.8)	1439	5972 (24)
929 RBO	Straight Blade	1283 (50.5)	737 (29)	626×703 (24.6×27.7)	1286	5972 (24)
933 RBO	Straight Blade	1461 (57.5)	838 (33)	711×800 (28×31.5)	1129	5972 (24)
200 BC 3	Backward Inclined	508 (20)	543 (21.38)	405×538 (16.1×21.4)	3169	3235 (13)
222 BC 3	Backward Inclined	565 (22.25)	603 (23.75)	449×598 (17.8×23.8)	2848	3235 (13)
245 BC 3	Backward Inclined	622 (24.5)	662 (26.06)	494×659 (19.6×26.2)	2587	3235 (13)
300 BC 3	Backward Inclined	762 (30)	803 (31.63)	605×808 (23.9×32)	2059	3484 (14)
365 BC 3	Backward Inclined	927 (36.5)	978 (38.50)	734×983 (29.1×39)	1727	3981 (16)
402 BC 3	Backward Inclined	1022 (40.25)	1078 (42.44)	808×1083 (32.1×42.9)	1566	3981 (16)
445 BC 3	Backward Inclined	1130 (44.5)	1191 (46.88)	894×1197 (35.4×47.4)	1416	3981 (16)
490 BC 3	Backward Inclined	1245 (49)	1311 (51.63)	981×1319 (38.8×52.2)	1286	3981 (16)
24B7	Vane-axial	610 (24)			2795	1742 (7)
36B7	Vane-axial	914 (36)			1966	1742 (7)
42B6	Vane-axial	1067 (42)			1610	1493 (6)
Kimbell Gin Machinery Co., Lubbock, TX						
KGM 35-40	Straight Blade	743 (29.25)	432 (17)	267×419 (10.5×16.5)	2351	5474 (22)
KGM 45-50	Straight Blade	908 (35.75)	533 (21)	318×521 (12.5×20.5)	2157	6968 (28)
KGM 55-60	Straight Blade	1016 (40)	610 (24)	406×559 (16×22)	2114	8461 (34)
KGM 60	Straight Blade	1118 (44)	635 (25)	514×660 (20.25×26)	1772	6968 (28)
KGM 70	Straight Blade	1118 (44)	699 (27.5)	572×686 (22.5×27)	1634	6470 (26)
KGM 80	Straight Blade	1283 (50.5)	762 (30)	635×749 (25×29.5)	1499	6968 (28)
Smith Fans Inc. - Lamesa, TX						
30	Straight Blade (MU)[y]	55.9 (22)	33.0 (13)	23.2×35.7 (9⅛×14)	2800	4484 (18)
45-50	Straight Blade (MH)[y]	95.9 (37.75)	53.3 (21)	30.5×54.3 (12×21⅜)	1800	6227 (25)
50C4	Straight Blade (LC)[y]	95.9 (37.75)	53.3 (21)	43.2×54.3 (17×21⅜)	1800	4982 (20)
60E	Straight Blade (MH)[y]	111.8 (44)	61.0 (24)	35.7×63.5 (14×25)	1550	6227 (25)
60C4	Straight Blade (MH & LC)	111.8 (44)	61.0 (24)	48.3×63.5 (19×25)	1550	4982 (20)
36BC	Backward Curved (BC & LC)[y]	91.4 (36)	101.6 (40)	71.8×99.1 (28.25×39)	1900	3487 (14)
40BC	Backward Curved (BC & LC)[y]	101.6 (40)	106.7 (42)	80.6×110.2 (31.75×43.38)	1700	3487 (14)
49BC	Backward Curved (BC & LC)[y]	124.5 (49)	127.0 (50)	97.8×124.5 (38.5×49)	1400	3487 (14)

[z] Data for older models available from Stedronsky (1964)

[y] MU = Moisture Unit, MH = Material and Trash Handling, LC = Lint Cleaner Pull, BC = Battery Condenser

Axial Fans. Axial-flow fans resemble propellers and consist of an impeller or bladed rotor that rotates inside a sheet metal housing in-line with the connected duct (Fig. 10). The air enters parallel to the shaft at the leading edge of the blade, is propelled by the blades, and the flow is essentially parallel to the shaft and swirls around the shaft. Pressure is created by the change in air velocity as it passes through the blades. The predominant axial fan used in cotton gins is the vane-axial. Vane-axial fans consist of an axial-flow impeller usually including 7 to 12 blades within a cylinder that contains a set of guide vanes located on the discharge side of the impeller. These guide vanes assist in recovering the energy used for tangential acceleration, and therefore vane-axial fans can operate against static pressures normally up to 1,500 Pa (6 in. H_2O) and as high as 2,500 Pa (10 in. H_2O). Axial-flow fans cannot handle material through the fan impeller. They are primarily used in low-pressure machinery, such as lint-cleaner condensers and battery condensers. An access door should be provided on both sides of the fan to facilitate inspection for lint fly buildup, especially on the fixed vanes. In recent years, many gins have replaced axial fans on their lint cleaner exhausts with centrifugal fans, allowing gins to use cyclones on those exhausts.

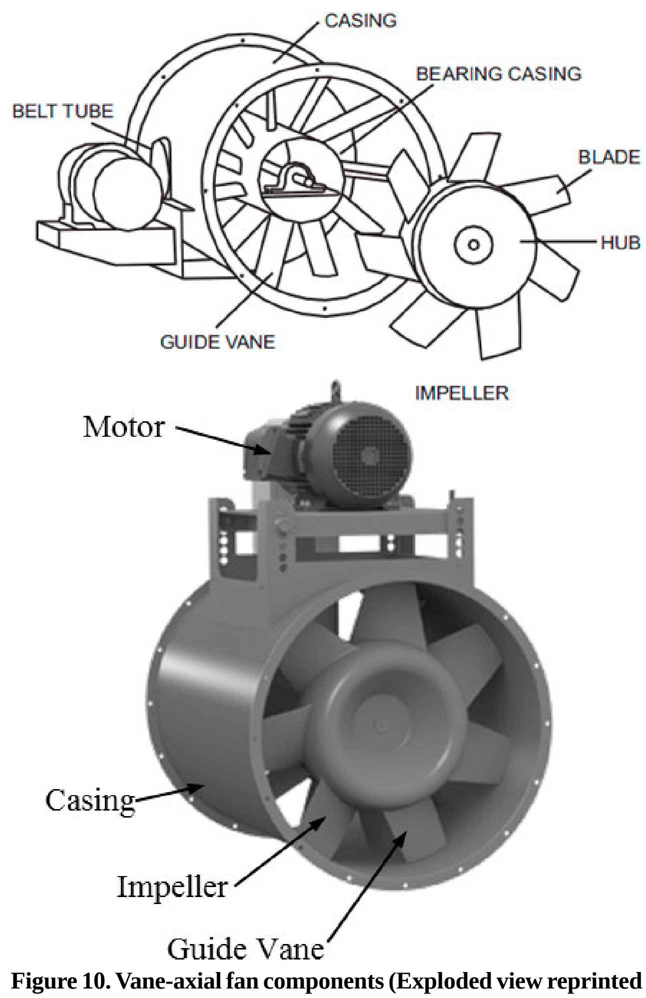


Figure 10. Vane-axial fan components (Exploded view reprinted with permission from AMCA Publication 201-02, Fans and Systems]).

Another type of axial fan used in gins, but to a lesser extent is a tube-axial fan. These fans consist of an axialflow impeller with four to seven helical blades rotating within a cylinder without the guide vanes for energy recovery present on vane-axial fans. Thus, tube-axial fans operate against lower resistance pressures than vane-axial fans, approximately 500 to 1,000 Pa (2-4 in. H_2O). These fans are sometimes referred to in the ginning industry as duct fans.

Axialflow fans are sized by the diameter of the cylinder housing and typically range from 300 mm to 1,500 mm (12-60 in.). Axial-flow fans generally require more horsepower as the flow rate decreases, with maximum horsepower required when flow is blocked completely (Fig. 11).

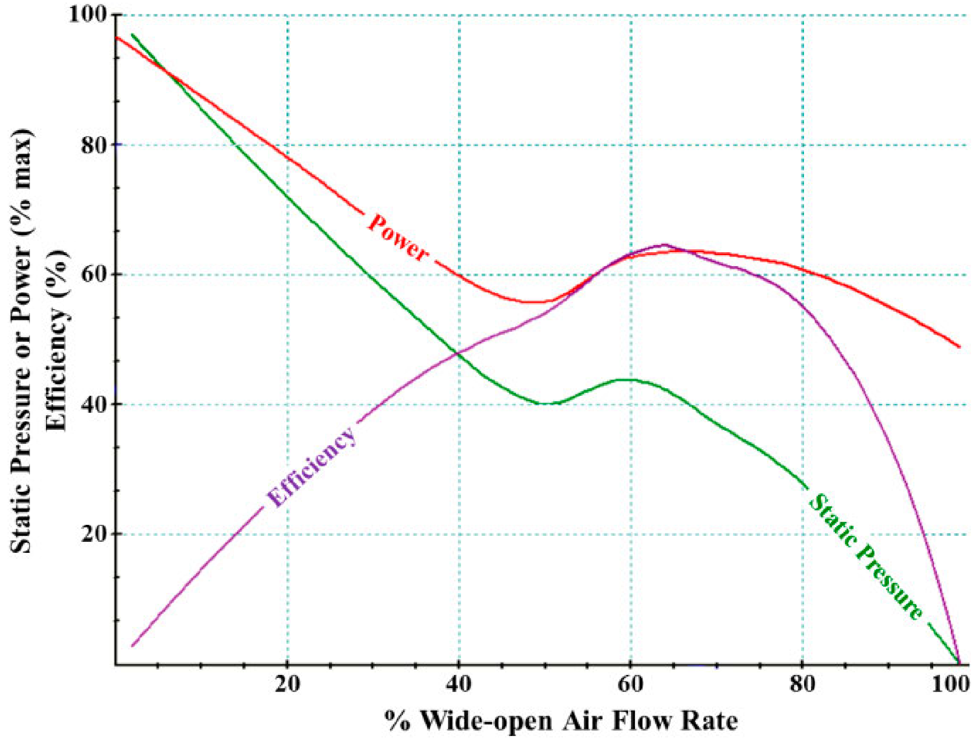


Figure 11. Typical vane-axial performance curve.

Fan Performance Tables and Curves. The amount of air that a fan will deliver depends on its design and construction, its speed, and the static pressure of the system. Fan manufacturers test their fans and prepare fan tables and curves that show air flow capabilities and horsepower requirements of the fans at various shaft speeds and static pressures. To select a fan for a given application, the required air flow rate, static pressure for the system, and fan performance rating tables or curves from the manufacturer are needed. It is good practice to have fan tables or curves on file for all fans used in the gin. A typical performance table is shown in Fig. 12 (Fig. 12a for SI Units and Fig. 12b for USC Units) and a typical performance curve for the same fan is shown in Fig. 13 (Fig. 13a for SI Units and Fig. 13b for USC Units). The following example demonstrates the use of these fan tables and curves.

Example 5: The power needed to drive a centrifugal fan can vary over a broad range, even when the fan is operated at a constant speed. For example, a gin system requires 325 m^3/min (11,500 ft^3/min) and has 4,000 Pa (16 in. H_2O) of static pressure.

From Fig. 12, moving across the top of the table sections the nearest static pressure is 3,981 Pa (16 in. H_2O). The closest or slightly higher air flow rate in the first column is 326 m^3/min (11,500 ft^3/min). The intersection of the static pressure column and flow rate row shows that the fan uses 34.21 kW (45.88 hp) at a speed of 1,350 rpm.

Similarly, from Fig. 13, the intersection of 325 m^3/min (11,500 ft^3/min) on the x-axis and 4,000 Pa (16 in. H_2O) of static pressure on the y-axis falls between the 1,200-rpm and 1,400-rpm green static-rpm curves at approximately 1,350 rpm. Next, proceed up from 325 m^3/min (11,500 ft^3/min) on the x-axis to 1,350 rpm between the red power-rpm curves. Then, move right to the corresponding power of approximately 35 kW (46 hp).

To size a motor, round up to a 37- or 45-kW (50- or 60-hp) motor to select a standard size motor and account for drive losses and variability in conditions.

This same fan at the same speed will operate differently if connected to another piping system having a static resistance of only 3,484 Pa (14 in. H_2O). From Fig. 12, under the new static pressure column and approximately the same wheel speed (1,359 rpm) the fan would deliver approximately 453-m^3/min (16,000-ft^3/min) flow rate and require 46.26 kW (62.04 hp).

Performance Data

RBO 923

Wheel Diameter: 1016 mm

Inlet Area: 0.268 sq. m
Inlet Diameter: 594 mm O.D.

Outlet Area: 0.269 sq. m
Outlet Dimension: 498 mm x 559 mm

Tip Speed (m/s): 0.0532 x RPM

Flow	OV	124-Pa SP		249-Pa SP		498-Pa SP		995-Pa SP		1493-Pa SP		1991-Pa SP		2448-Pa SP		2986-Pa SP		3484-Pa SP	
m^3/min	m/s	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW
102	6.30	273	0.41	351	0.71	476	1.36	663	2.75	810	4.24								
119	7.36	289	0.54	364	0.87	482	1.59	666	3.18	811	4.84	934	6.57	1043	8.40				
136	8.41	307	0.69	378	1.06	491	1.85	670	3.61	814	5.49	936	7.38	1044	9.34	1142	11.39		
159	9.81	332	0.92	399	1.36	507	2.27	677	4.21	818	6.32	939	8.50	1046	10.69	1144	12.96	1233	15.24
181	11.21	359	1.23	423	1.74	525	2.74	688	4.88	824	7.19	944	9.64	1050	12.12	1147	14.63	1236	17.15
204	12.61	388	1.62	447	2.17	545	3.29	702	5.65	832	8.11	949	10.76	1054	13.50	1150	16.29	1239	19.10
227	14.02	418	2.09	472	2.68	567	3.93	718	6.50	844	9.16	956	11.95	1060	14.94	1155	17.98	1243	21.08
249	15.41	450	2.66	499	3.28	590	4.65	736	7.43	859	10.35	967	13.30	1067	16.42	1160	19.66	1247	22.98
272	16.81	482	3.33	527	3.98	614	5.47	755	8.46	875	11.62	980	14.77	1077	18.06	1167	21.44	1253	25.00
294	18.22	514	4.10	556	4.79	638	6.38	776	9.62	892	12.97	995	16.37	1089	19.83	1177	23.40	1260	27.07
317	19.62	548	5.03	587	5.76	663	7.39	797	10.86	910	14.41	1011	18.06	1103	21.74	1189	25.50	1269	29.28
340	21.02	581	6.06	618	6.84	689	8.52	820	12.28	930	16.02	1029	19.92	1119	23.83	1202	27.72	1281	31.74
362	22.42	615	7.26	650	8.08	717	9.84	843	13.79	951	17.78	1047	21.83	1135	25.96	1218	30.19	1295	34.38
385	23.83	649	8.58	682	9.45	745	11.27	867	15.45	972	19.64	1066	23.88	1153	28.29	1234	32.73	1309	37.09
408	25.23	683	10.07	714	10.96	775	12.92	891	17.23	995	21.73	1087	26.18	1171	30.69	1251	35.41	1325	40.01
430	26.62	718	11.76	747	12.68	805	14.70	916	19.17	1018	23.94	1108	28.61	1191	33.35	1269	38.23	1342	43.11
453	28.03	753	13.62	781	14.61	835	16.64	941	21.24	1041	26.26	1129	31.13	1211	36.12	1287	41.14	1359	46.26
476	29.43	788	15.67	814	16.66	866	18.79	968	23.58	1065	28.80	1152	33.95	1232	39.11	1307	44.35	1378	49.70
498	30.83	823	17.92	848	18.96	898	21.19	995	26.07	1089	31.46	1175	36.91	1253	42.23	1327	47.69	1396	53.14
521	32.23	858	20.37	882	21.45	930	23.76	1023	28.78	1114	34.36	1199	40.10	1276	45.70	1348	51.29	1416	56.93

Flow	OV	3981-Pa SP		4479-Pa SP		4977-Pa SP		5474-Pa SP		5972-Pa SP		6470-Pa SP		6968-Pa SP		7465-Pa SP		7963-Pa SP	
m^3/min	m/s	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW	RPM	kW
170	10.51	1318	18.66	1396	21.19	1470	23.80												
184	11.38	1319	19.98	1397	22.63	1471	25.35	1541	28.12	1609	31.01								
198	12.26	1321	21.36	1398	24.12	1472	26.96	1542	29.85	1609	32.80	**1673**	**35.79**	**1735**	**38.87**				
212	13.14	1323	22.77	1400	25.67	1473	28.62	1543	31.63	1610	34.70	**1674**	**37.81**	**1736**	**41.00**	**1795**	**44.18**	**1853**	**47.50**
227	14.02	1325	24.17	1402	27.25	1475	30.36	1545	33.53	1611	36.67	**1675**	**39.90**	**1736**	**43.14**	**1796**	**46.51**	**1853**	**49.86**
241	14.89	1327	25.54	1404	28.82	1477	32.10	1546	35.38	1613	38.74	**1676**	**42.04**	**1737**	**45.41**	**1796**	**48.83**	**1854**	**52.37**
269	16.64	1333	28.32	1410	32.00	1482	35.63	1551	39.28	1616	42.85	**1680**	**46.55**	**1740**	**50.15**	**1799**	**53.86**	**1856**	**57.59**
297	18.39	1340	31.16	1416	35.12	1488	39.10	1556	43.06	1621	47.05	**1684**	**51.06**	**1745**	**55.10**	**1803**	**59.07**	**1859**	**63.03**
326	20.15	1350	34.21	1424	38.38	1494	42.56	1562	46.86	**1627**	**51.19**	**1689**	**55.51**	**1750**	**59.97**	**1808**	**64.33**	**1864**	**68.66**
354	21.89	1363	37.53	1434	41.83	1503	46.28	1570	50.85	**1634**	**55.43**	**1695**	**59.98**	**1755**	**64.69**	**1813**	**69.42**	**1869**	**74.17**
382	23.65	1379	41.16	1448	45.66	1515	50.30	1579	54.94	**1642**	**59.78**	**1703**	**64.67**	**1762**	**69.62**	**1819**	**74.58**	**1874**	**79.54**
411	25.40	1398	45.14	1465	49.85	1530	54.67	1592	59.48	**1653**	**64.47**	**1712**	**69.51**	**1770**	**74.68**	**1826**	**79.86**	**1881**	**85.17**
439	27.15	1418	49.33	1484	54.35	1547	59.35	1607	64.32	**1667**	**69.57**	**1724**	**74.72**	**1780**	**80.01**	**1835**	**85.42**	**1889**	**90.94**
467	28.91	1438	53.62	1503	58.95	1565	64.24	**1625**	**69.60**	**1683**	**75.00**	**1739**	**80.39**	**1793**	**85.78**	**1847**	**91.42**	**1899**	**97.02**
496	30.65	1461	58.37	1524	63.90	1585	69.50	**1644**	**75.14**	**1701**	**80.80**	**1755**	**86.30**	**1809**	**92.06**	**1861**	**97.78**		
524	32.41	1484	63.27	1546	69.09	1606	75.00	**1664**	**80.93**	**1720**	**86.86**	**1774**	**92.76**	**1826**	**98.61**	**1877**	**104.55**		
552	34.16	1508	68.49	1569	74.58	**1628**	**80.77**	**1685**	**86.99**	**1740**	**93.20**	**1793**	**99.36**	**1845**	**105.61**	**1895**	**111.78**		
580	35.91	1533	74.08	1593	80.41	**1651**	**86.85**	**1707**	**93.35**	**1761**	**99.83**	**1813**	**106.25**	**1864**	**112.74**				
609	37.66	1559	80.07	1618	86.64	**1675**	**93.30**	**1730**	**100.04**	**1783**	**106.77**	**1834**	**113.44**	**1885**	**120.38**				
637	39.42	1586	86.46	**1644**	**93.29**	**1700**	**100.18**	**1754**	**107.13**	**1806**	**114.07**	**1856**	**120.97**	**1906**	**128.15**				

MAXIMUM RPM: CLASS 22 = 1623 CLASS 32 = 1910
Performance shown is for installation type B & D: Free or ducted inlet, ducted outlet.
Power ratings (BHP) do not include drive losses.
Performance ratings do not include the effects of appurtenances in the airstream.

Underlined figures indicate maximum static efficiency.
Regular face type = Class 22
Bold face type = Class 3

Figure 12a. 923 RBO straight blade, centrifugal fan (class 22) performance table—SI units. Adapted from Industrial Fans. 2000 Twin City Fan & Blower Bulletin 902; reprinted with permission.

Performance Data

RBO 923

Wheel Diameter: 40”

Inlet Area: 2.88 sq. ft.
Inlet Diameter: $23^3/_8$" O.D.

Outlet Area: 2.90 sq. ft.
Outlet Dimension: $19^5/_8$” x 22”

Tip Speed (FPM): 10.47 x RPM

Flow	OV	0.5” SP		1” SP		2” SP		4” SP		6” SP		8” SP		10” SP		12” SP		14” SP	
CFM	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
3600	1241	273	0.55	351	0.95	476	1.82	663	3.69	810	5.69								
4200	1448	289	0.72	364	1.17	482	2.13	666	4.27	811	6.49	934	8.81	1043	11.27				
4800	1655	307	0.92	378	1.42	491	2.48	670	4.84	814	7.36	936	9.90	1044	12.53	1142	15.28		
5600	1931	332	1.24	399	1.82	507	3.05	677	5.64	818	8.48	939	11.40	1046	14.34	1144	17.38	1233	20.44
6400	2207	359	1.65	423	2.33	525	3.68	688	6.55	824	9.64	944	12.93	1050	16.25	1147	19.62	1236	23.00
7200	2483	388	2.17	447	2.91	545	4.41	702	7.58	832	10.87	949	14.43	1054	18.10	1150	21.85	1239	25.62
8000	2759	418	2.80	472	3.59	567	5.27	718	8.72	844	12.29	956	16.03	1060	20.03	1155	24.11	1243	28.27
8800	3034	450	3.57	499	4.40	590	6.24	736	9.97	859	13.88	967	17.84	1067	22.02	1160	26.36	1247	30.82
9600	3310	482	4.47	527	5.34	614	7.34	755	11.34	875	15.58	980	19.81	1077	24.22	1167	28.75	1253	33.52
10400	3586	514	5.50	556	6.43	638	8.55	776	12.90	892	17.39	995	21.95	1089	26.59	1177	31.38	1260	36.30
11200	3862	548	6.75	587	7.72	663	9.91	797	14.57	910	19.32	1011	24.22	1103	29.15	1189	34.20	1269	39.27
12000	4138	581	8.13	618	9.17	689	11.43	820	16.47	930	21.48	1029	26.71	1119	31.95	1202	37.17	1281	42.56
12800	4414	615	9.73	650	10.83	717	13.19	843	18.49	951	23.84	1047	29.27	1135	34.81	1218	40.49	1295	46.11
13600	4690	649	11.51	682	12.67	745	15.11	867	20.72	972	26.34	1066	32.03	1153	37.94	1234	43.89	1309	49.74
14400	4966	683	13.50	714	14.70	775	17.32	891	23.1	995	29.14	1087	35.11	1171	41.16	1251	47.48	1325	53.66
15200	5241	718	15.77	747	17.00	805	19.71	916	25.71	1018	32.10	1108	38.36	1191	44.72	1269	51.27	1342	57.81
16000	5517	753	18.27	781	19.59	835	22.31	941	28.49	1041	35.22	1129	41.75	1211	48.44	1287	55.17	1359	62.04
16800	5793	788	21.02	814	22.34	866	25.20	968	31.62	1065	38.62	1152	45.53	1232	52.45	1307	59.48	1378	66.65
17600	6069	823	24.03	848	25.42	898	28.41	995	34.96	1089	42.19	1175	49.50	1253	56.63	1327	63.96	1396	71.26
18400	6345	858	27.31	882	28.76	930	31.86	1023	38.60	1114	46.08	1199	53.78	1276	61.28	1348	68.78	1416	76.34

Flow	OV	16” SP		18” SP		20” SP		22” SP		24” SP		26” SP		28” SP		30” SP		32” SP	
CFM	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
6000	2069	1318	25.03	1396	28.42	1470	31.92												
6500	2241	1319	26.79	1397	30.35	1471	34.00	1541	37.71	1609	41.59								
7000	2414	1321	28.65	1398	32.34	1472	36.16	1542	40.03	1609	43.98	**1673**	**47.99**	**1735**	**52.12**				
7500	2586	1323	30.54	1400	34.43	1473	38.38	1543	42.42	1610	46.54	**1674**	**50.70**	**1736**	**54.98**	**1795**	**59.25**	**1853**	**63.70**
8000	2759	1325	32.41	1402	36.54	1475	40.71	1545	44.96	1611	49.17	**1675**	**53.51**	**1736**	**57.85**	**1796**	**62.37**	**1853**	**66.86**
8500	2931	1327	34.25	1404	38.65	1477	43.05	1546	47.44	1613	51.95	**1676**	**56.38**	**1737**	**60.90**	**1796**	**65.48**	**1854**	**70.23**
9500	3276	1333	37.98	1410	42.91	1482	47.78	1551	52.67	1616	57.46	**1680**	**62.43**	**1740**	**67.25**	**1799**	**72.23**	**1856**	**77.23**
10500	3621	1340	41.78	1416	47.09	1488	52.43	1556	57.75	1621	63.09	**1684**	**68.47**	**1745**	**73.89**	**1803**	**79.21**	**1859**	**84.52**
11500	3966	1350	45.88	1424	51.47	1494	57.07	1562	62.84	**1627**	**68.65**	**1689**	**74.44**	**1750**	**80.42**	**1808**	**86.27**	**1864**	**92.08**
12500	4310	1363	50.33	1434	56.09	1503	62.06	1570	68.19	**1634**	**74.33**	**1695**	**80.43**	**1755**	**86.75**	**1813**	**93.10**	**1869**	**99.47**
13500	4655	1379	55.19	1448	61.23	1515	67.45	1579	73.68	**1642**	**80.16**	**1703**	**86.73**	**1762**	**93.36**	**1819**	**100.02**	**1874**	**106.67**
14500	5000	1398	60.53	1465	66.85	1530	73.32	1592	79.77	**1653**	**86.46**	**1712**	**93.21**	**1770**	**100.15**	**1826**	**107.10**	**1881**	**114.21**
15500	5345	1418	66.15	1484	72.88	1547	79.59	1607	86.25	**1667**	**93.29**	**1724**	**100.20**	**1780**	**107.30**	**1835**	**114.55**	**1889**	**121.95**
16500	5690	1438	71.91	1503	79.05	1565	86.15	**1625**	**93.34**	**1683**	**100.57**	**1739**	**107.81**	**1793**	**115.03**	**1847**	**122.60**	**1899**	**130.10**
17500	6034	1461	78.27	1524	85.69	1585	93.20	**1644**	**100.76**	**1701**	**108.35**	**1755**	**115.73**	**1809**	**123.46**	**1861**	**131.13**		
18500	6379	1484	84.84	1546	92.65	1606	100.58	**1664**	**108.53**	**1720**	**116.48**	**1774**	**124.39**	**1826**	**132.24**	**1877**	**140.21**		
19500	6724	1508	91.84	1569	100.01	**1628**	**108.31**	**1685**	**116.66**	**1740**	**124.98**	**1793**	**133.24**	**1845**	**141.63**	**1895**	**149.90**		
20500	7069	1533	99.34	1593	107.83	**1651**	**116.47**	**1707**	**125.18**	**1761**	**133.87**	**1813**	**142.48**	**1864**	**151.19**				
21500	7414	1559	107.37	1618	116.18	**1675**	**125.12**	**1730**	**134.15**	**1783**	**143.18**	**1834**	**152.12**	**1885**	**161.43**				
22500	7759	1586	115.95	**1644**	**125.10**	**1700**	**134.34**	**1754**	**143.66**	**1806**	**152.97**	**1856**	**162.22**	**1906**	**171.85**				

MAXIMUM RPM: CLASS 22 = 1623 CLASS 32 = 1910

Performance shown is for installation type B & D: Free or ducted inlet, ducted outlet.
Power ratings (BHP) do not include drive losses.
Performance ratings do not include the effects of appurtenances in the airstream.

Underlined figures indicate maximum static efficiency.
Regular face type = Class 22
Bold face type = Class 32

Figure 12b. 923 RBO straight blade, centrifugal fan (class 22) performance table—USC units. Adapted from Industrial Fans. 2000 Twin City Fan & Blower Bulletin 902.

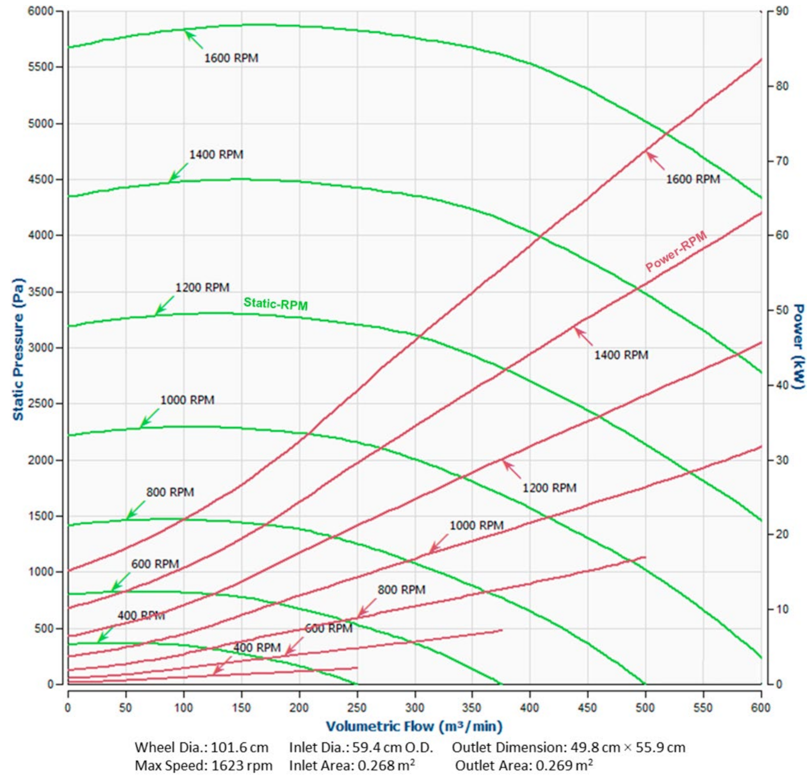


Figure 13a. 923 RBO straight blade, centrifugal fan performance curve—SI units. Produced with Twin City Fan Selector v10. 2017 Twin City Fan & Blower, Minneapolis, MN; reprinted with permission.

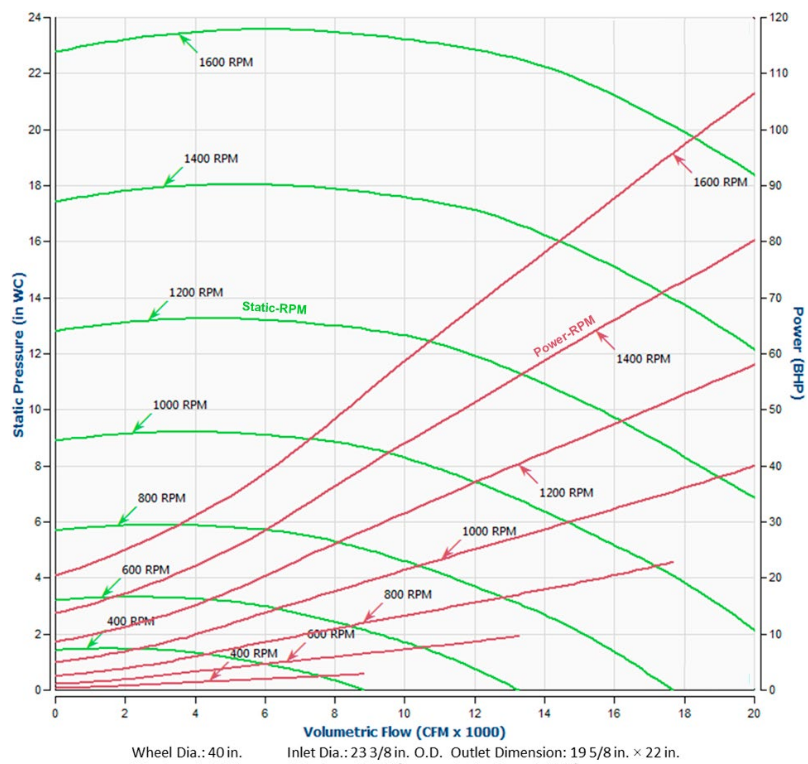


Figure 13b. 923 RBO straight blade, centrifugal fan performance curve—USC units. Produced with Twin City Fan Selector v10. 2017 Twin City Fan & Blower, Minneapolis, MN; reprinted with permission.

Maximum Fan Speed. Before increasing the speed of a fan, it is necessary to determine maximum operating speed of the fan and ensure that it is not exceeded. Fans have structural elements that are affected by fan speed. These elements can fail catastrophically if the maximum safe speed of the fan is exceeded. The maximum speed varies based on the fan blade type, configuration, and precision of manufacturing. The maximum fan speed is usually included along with other fan characteristics in the fan table or curve, or contact the manufacturer to determine the maximum operating speed. If maximum fan speed information is not available, one rule of thumb is that the maximum safe speed for many fans used in gins is 91-m/s (300-ft/s) tip speed. Fan tip speed is calculated from the following equation:

$$TS = \pi \times D \times \omega / 60 \qquad \text{(Eq. 4)}$$

where:

TS = fan tip speed, m/s (ft/s)

π = 3.1416

D = fan blade wheel diameter, mm/1000 or m (in./12 or ft)

ω = fan shaft rotational speed, rpm.

Example 6: The fan in Fig. 12 has a wheel diameter of 1,016 mm (40 in.) and a maximum fan speed of 1,623 rpm. What is the fan tip speed?

Using Equation 4:

$$TS = 3.1416 \times 1016 \text{ mm} / 1000 \times 1623 / 60 = 86 \text{ m/s}$$

or

$$TS = 3.1416 \times 40 \text{ in.} / 12 \times 1623 / 60 = 283 \text{ ft/s}$$

In addition, small increases in speed have large penalties in power. Before increasing speed, first calculate the new power required using the fan laws discussed later in this section to ensure that the fan's motor and electric service have enough capacity.

Finally, when fan speed is increased, expect the fan to be louder, to have increased noise from the air system fittings, and have more duct and fitting leakage.

Series and Parallel Fans. There are situations where multiple fans are used in a system to greatly increase the air flow rate or overcome large static pressures. If two fans are used on the same system, the combined performance depends on their relative placement (Fig. 14). When two fans are placed parallel to each other, the air volume is doubled, but the static pressure remains the same; however, if they are placed in series, the air volume remains the same and the static pressure is doubled. Large-diameter fan wheels operating at low speeds are generally more efficient and produce less noise than small-diameter fan wheels at high speeds.

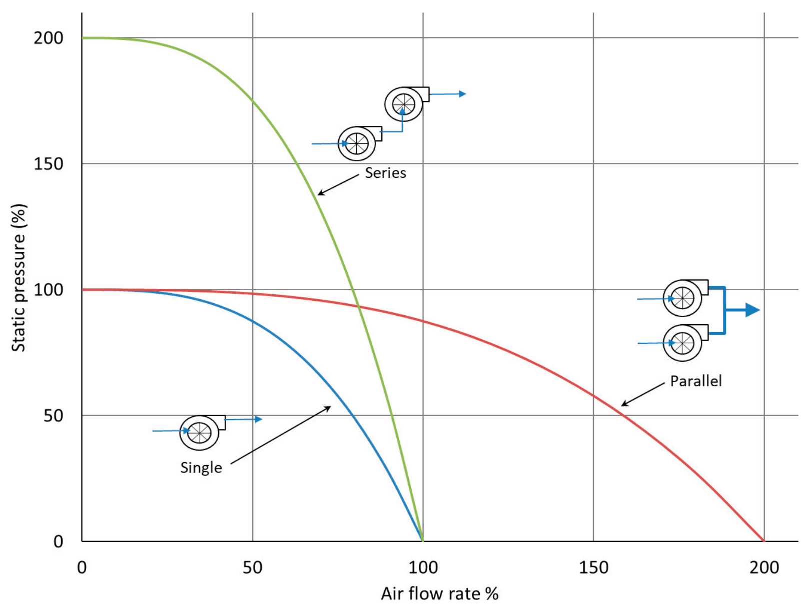


Figure 14. Static Pressure and air flow relationship for single, series, and parallel fans.

Fan Performance Temperature and Elevation Correction. Manufacturers base fan performance ratings on handling air at standard conditions; air at sea level having temperature of 21 °C (70 °F), barometric pressure of 101.3 kPa (29.92 in. Hg), and density of 1.2 kg/m^3 (0.075 lb/ft^3). When the conditions vary significantly from standard conditions, the static pressure and power requirements for the fan will change. For example, air at higher elevation and/or higher temperatures is less dense. Thus, the static pressure and power requirements for the fan will be lower than for the same fan at standard conditions. Correction factors for adjusting required fan static pressure and power for variations in temperature and elevation are given in Table 6, and the following examples demonstrate the proper use of the factors.

Table 6. Correction factors for adjusting estimated pressure and horsepower requirements of fans operating at nonstandard conditions

Air Temperature (°C)	Factor	Elevation[z] (m)	Factor	Air Temperature (°F)	Factor	Elevation[z] (ft)	Factor
-15	0.88	0[y]	1.00	0	0.87	0[y]	1.00
-10	0.90	100	1.01	10	0.89	500	1.02
-5	0.91	200	1.03	20	0.91	1000	1.04
0	0.93	300	1.04	30	0.93	1500	1.06
5	0.94	400	1.05	40	0.94	2000	1.08
10	0.96	500	1.07	50	0.96	2500	1.10
15	0.98	600	1.08	60	0.98	3000	1.12
21[y]	1.00	700	1.09	70[y]	1.00	3500	1.14
25	1.01	800	1.10	80	1.02	4000	1.16
30	1.03	900	1.12	90	1.04	4500	1.18
35	1.05	1000	1.13	100	1.06	5000	1.20
40	1.07	1100	1.14	110	1.08	5500	1.22
45	1.08	1200	1.16	120	1.09		
50	1.09	1300	1.17	130	1.11		
55	1.11	1400	1.18	140	1.13		
60	1.13	1500	1.20	150	1.15		
65	1.15	1600	1.21	160	1.17		
70	1.17	1700	1.22	170	1.19		
75	1.18			180	1.21		
80	1.20			190	1.23		
85	1.22			200	1.25		
90	1.24			225	1.28		
95	1.25			250	1.34		
100	1.27			275	1.39		
110	1.30			300	1.43		
120	1.34			325	1.48		
140	1.40			350	1.53		
160	1.47			375	1.58		
180	1.54			400	1.62		
200	1.61			425	1.67		
220	1.68			450	1.72		
240	1.75			475	1.77		
260	1.81			500	1.81		

[z] **Above mean sea level.**

[y] **Standard condition.**

Example 7: The fan in Example 5 was selected to deliver 326 m^3/min (11,500 ft^3/min) of air at 3981 Pa (16 in. H_2O) static pressure requiring a wheel speed of 1,350 rpm and 34.21 kW (45.88 hp) power at standard conditions. Suppose, that the actual operating conditions are at an elevation of 750 m (2,500 ft) and the air has been heated to 60 °C (140 °F). What are the performance characteristics of the fan under the actual operating conditions (assuming the fan speed remains the same)?

Because the fan speed was not changed, the fan will continue to deliver 326 m^3/min (11,500 ft^3/min) of air. This air, however, will be less dense than standard air because of the higher temperature and elevation. This change in density will affect static pressure and the horsepower requirement. From Table 6, the correction factor for 60 °C (140 °F) air temperature is 1.13 and the factor for 750-m (2,500-ft) elevation is approximately 1.10. The new operating static pressure:

$$\text{Static pressure} = 3981\ \text{Pa} / (1.13 \times 1.10) = 3203\ \text{Pa}$$

or

$$\text{Static pressure} = 16\ \text{in.}H_2O / (1.13 \times 1.10) = 12.9\ \text{in.}\ H_2O$$

The new power requirement:

$$\text{Power} = 34.21\ \text{kW} / (1.13 \times 1.10) = 27.5\ \text{kW}$$

or

$$\text{Power} = 45.88\ \text{hp} / (1.13 \times 1.10) = 36.9\ \text{hp}$$

A 30-kW (40-hp) or slightly higher [37-kW (50-hp)] motor would be appropriate.

Example 8: What should the power requirements be if the fan will run without the burner on?

Without the burner to heat the air, it is assumed that the air would be at standard temperature: 21 °C (70 °F). The fan would still be operated at 750-m (2,500-ft) elevation. Thus, only the elevation correction factor from Example 7 would be used to correct the fan power. The fan power requirement for elevation only:

$$\text{Power} = 34.21\ \text{kW} / 1.10 = 31.1\ \text{kW}$$

or

$$\text{Power} = 45.88\ \text{hp} / 1.10 = 41.7\ \text{hp}$$

In this case, a 37-kW (50-hp) motor would be necessary.

Fan Laws. The performance of a fan changes when its speed is changed. The faster a fan is operated, the more air it will deliver on a specific system, and the greater the static pressure it will develop. Output power is equal to the product of the air flow rate and the total pressure developed by the fan; therefore, power requirements increase significantly as fan speed is increased. If it becomes necessary to speed up fans, resulting changes in performance can be predicted from the following fan laws:

1. Fan capacity or air flow rate varies directly with the fan wheel speed.

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1} \text{ or } Q_2 = Q_1 \times \frac{N_2}{N_1} \text{ or } N_2 = N_1 \times \frac{Q_2}{Q_1} \qquad \text{(Eq. 5)}$$

2. Fan static pressure varies as the square of the fan wheel speed.

$$\frac{SP_2}{SP_1} \times \left(\frac{N_2}{N_1}\right)^2 \text{ or } SP_2 = SP_1 \times \left(\frac{N_2}{N_1}\right)^2 = SP_1 \times \text{speed ratio}^2 \qquad \text{(Eq. 6)}$$

3. Fan power required varies as the cube of the fan wheel speed.

$$\frac{P_2}{P_1} \times \left(\frac{N_2}{N_1}\right)^3 \text{ or } P_2 = P_1 \times \left(\frac{N_2}{N_1}\right)^3 = P_1 \times \text{speed ratio}^3 \qquad \text{(Eq. 7)}$$

where:

subscript 1 denotes current conditions and subscript 2 denotes new conditions under consideration;
Q = air flow rate, m^3/min (ft^3/min);
N = fan wheel speed (revolutions per minute, rpm);
speed ratio = the ratio of new fan speed to current or old speed, $\frac{N_2}{N_1}$;
SP = static pressure, Pa (in. H_2O); and
P = fan power, kW (hp).

Many ginners do not fully understand the effects of fan speed, which is described in the fan law equations (Equations 5-7). Figure 15 illustrates the relationship between fan speed, air flow, static pressure, and power. If fan speed is increased by 25%, the air flow rate also increases by 25%; but the static pressure increases by 56% and the power consumed increases 95%. Even small increases in fan speed can require the installation of a more powerful motor.

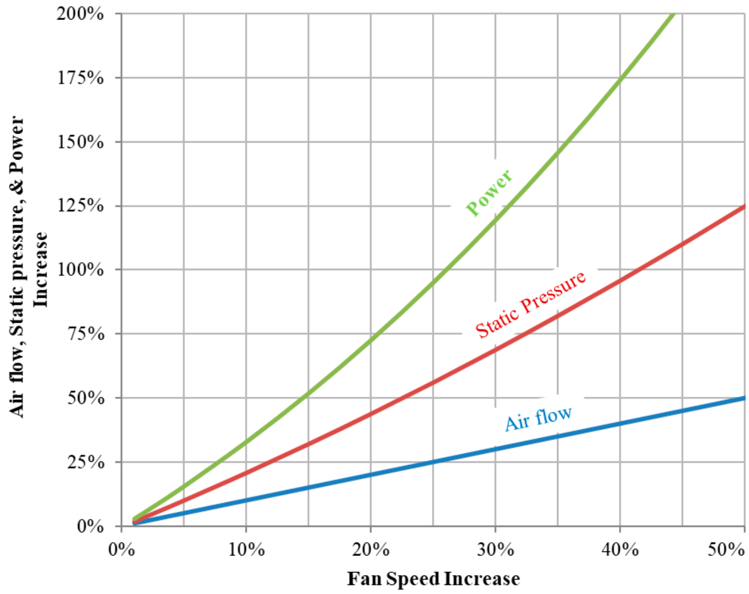


Figure 15. Air flow, static pressure, power, and fan speed relationships based on fan laws.

Motor current (in amps) can be substituted for power in Equation 7 with only a slight error if the power factor and voltage remain constant. Current is more convenient for ginners to use, because motor current is easier to measure than actual power. The results will be accurate enough for field calculations. The following example illustrates the use of the fan laws to predict fan performance.

Example 9. A fan currently delivers 142 m^3/min (5,000 ft^3/min) of air at 1,600-rpm impeller speed using 12 kW (16 hp) to a gin system with 2,488-Pa (10-in. H_2O) static pressure. The ginner desires to increase the air flow rate by 10% (new air flow rate = 156 m^3/min [5,500 ft^3/min]). Determine the new wheel speed, static pressure, and power required for the increase.

From Equation 5, the new speed:

$$\text{speed ratio} = \frac{N_2}{N_1} = \frac{Q_2}{Q_1} = \frac{156\ m^3/\min}{142\ m^3/\min} = 1.10$$

or

$$\text{speed ratio} = \frac{N_2}{N_1} = \frac{Q_2}{Q_1} = \frac{5500\ ft^3/\min}{5000\ ft^3/\min} = 1.10$$

$$N_2 = 1600\ \text{rpm} \times 1.10 = 1760\ \text{rpm}$$

From Equation 6, the new system static pressure:

$$SP_2 = 2488\ \text{Pa} \times 1.10^2 = 2488\ \text{Pa} \times 1.10 \times 1.10 = 3010\ \text{Pa}$$

or

$$SP_2 = 10\ \text{in.}H_2O \times 1.10^2 = 10\ \text{in.}H_2O \times 1.10 \times 1.10 = 12.1\ \text{in.}H_2O$$

From Equation 7, the new power requirement:

$$P_2 = 12\ \text{kW} \times 1.10^3 = 12\ \text{kW} \times 1.10 \times 1.10 \times 1.10 = 16.0\ \text{kW}$$

or

$$P_2 = 16\ \text{hp} \times 1.10^3 = 16\ \text{hp} \times 1.10 \times 1.10 \times 1.10 = 21.3\ \text{hp}$$

Care should be taken to choose a fan that will provide sufficient air flow for the system when the fan is operated within the speed ranges recommended by the manufacturer. Centrifugal fan performance varies widely due to differences in design and construction and how it is utilized in an air handling system. The efficiency of centrifugal fans can vary from 35 to 75% as a result of these differences. Fans for a given system should be selected so that they will operate at their most efficient point. If the system is changed, for example, a different dryer installed or pipes rerouted, the same fan could possibly be used. The fan speed could be varied to meet air flow requirements according to the fan laws, if the maximum safe operating speed specified by the manufacturer is not exceeded. However, the fan might not be operating at its most efficient point on the new system, and the electricity savings might justify purchase of a new fan. To obtain maximum efficiency from a fan, it is essential that the fan application be designed from accurate rating tables.

PRESSURE LOSSES

The static pressure needed for a required air flow depends on air velocity; size, shape, and frictional characteristics of the piping system; and the nature of the material conveyed. This static pressure requirement, called static pressure drop or loss, is caused by friction between the pipe walls and the air. Frictional losses in fittings, such as elbows, transitions, and valves, can be significantly larger than a comparable length of pipe. Static pressure loss also occurs for any piece of equipment through which the air flows, including inclined cleaners, tower driers, separators, and cyclones (Table 7). Conveying material also increases the static pressure requirement due to friction between the material and air, energy losses due to collisions between particles and the pipe walls, and overcoming gravity to lift and suspend material in the air stream. Additional static pressure also is required to accelerate the conveyed material when it is fed into the air stream and to lift the material vertically.

Table 7. Typical static pressure loss for gin components

Gin Components	Pressure Loss	
	Pa	in. H_2O
Burner	373-622	1.5-2.5
Blowbox	249-747	1-3
Inclined Cleaner	995-493	4-6
Separator	747-1244	3-5
Tower Drier	124 per shelf	0.5 per shelf
Cyclone 1D3D 2D2D 1D2D	1120 995 622	4.5 4 2.5
Battery Condenser	995-1244	4-5
Rock Trap	500	2
Lint Cleaner Condenser	373	1.5
Airjet Lint Cleaner	498	2

Straight Pipe Losses. The static pressure loss or friction loss due to air flow only in pipes can be determined from a friction chart, like that shown in Fig. 16 (Fig. 16a for SI Units and Fig. 16b for USC Units), which is based on standard air flowing through round galvanized pipes. Using Fig. 16, the friction loss can be estimated if two of the following variables are known: air flow rate, air velocity, and pipe diameter. The following example demonstrates use of Fig. 16.

Example 10. What is the friction loss of a 15-m (50-ft) long section of 500-mm (20-in.) diameter pipe with 23-m/s (4,500-ft/min) air velocity?

From Fig. 16, at the intersection of the 23-m/s (4,500-ft/min) air velocity diagonal and the 500-mm (20-in.) duct diameter diagonal, proceed horizontally to the left friction loss axis to read approximately 9.3 Pa/m (1.2 in. H_2O/100 ft). Thus, the friction loss for a 15-m (50-ft) section:

$$\text{Friction loss} = \frac{9.3\ \text{Pa}}{\text{m}} \times 15\ \text{m} = 139.5\ \text{Pa}$$

or

$$\text{Friction loss} = \frac{1.2\ \text{in.H}_2\text{O}}{100\ \text{ft}} \times 50\ \text{ft} = 0.6\ \text{in.H}_2\text{O}$$

Figure 16 shows that smaller pipes and higher air velocities have greater frictional resistance or static pressure loss than do larger pipes or lower air velocities and therefore require more power to move the air. When conveying air only, such as in the pipe from a burner to a blowbox, larger pipe diameters can be used to reduce static pressure requirements. Because no material is conveyed, lower air velocities are acceptable.

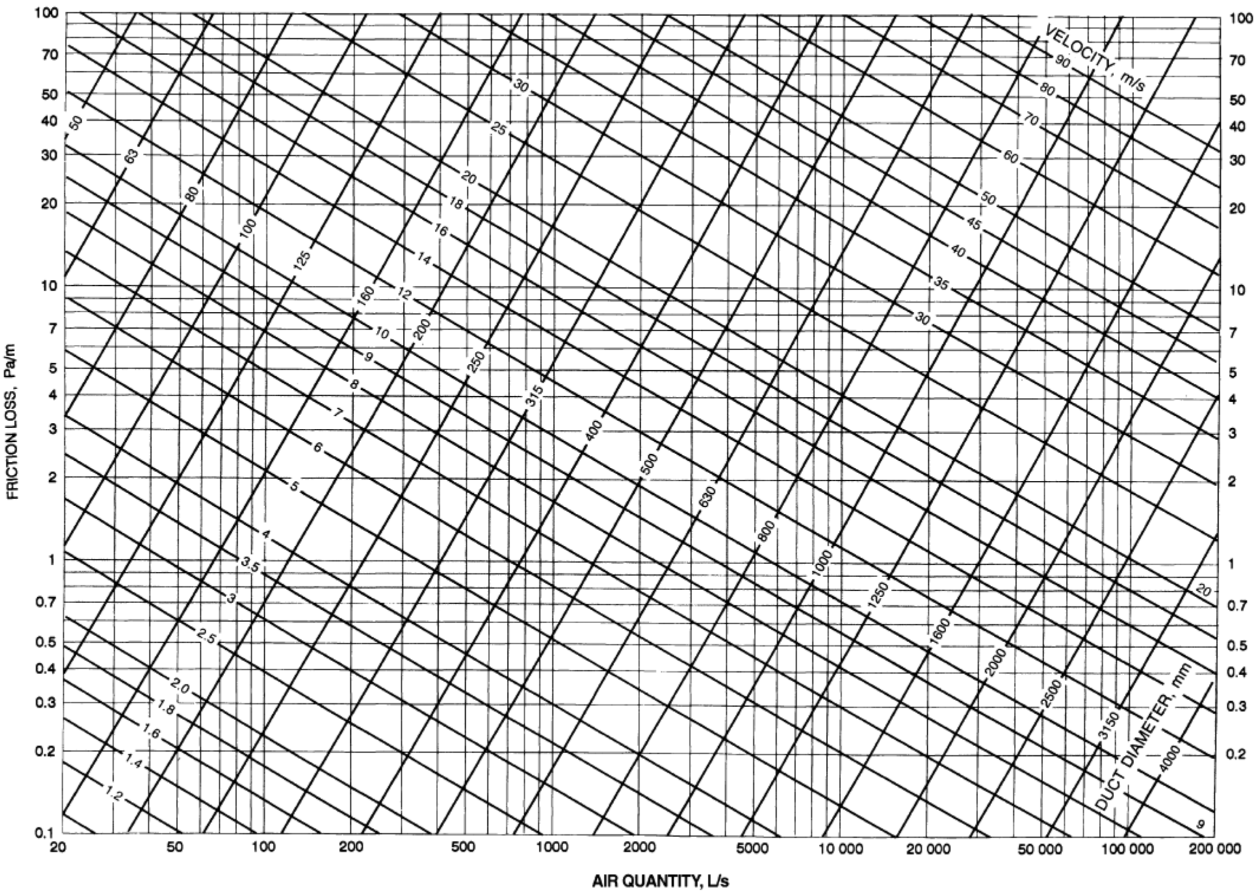


Figure 16a. Friction loss per meter length of round, galvanized pipe. Based on sea level-standard air conditions @ 21 °C and 101.3 kPa (1 L/s = 0.06 m³/min). Reprinted with permission from ASHRAE Handbook ©ASHRAE, www.ashrae.org. 2017. ASHRAE Handbook—Fundamentals (SI).

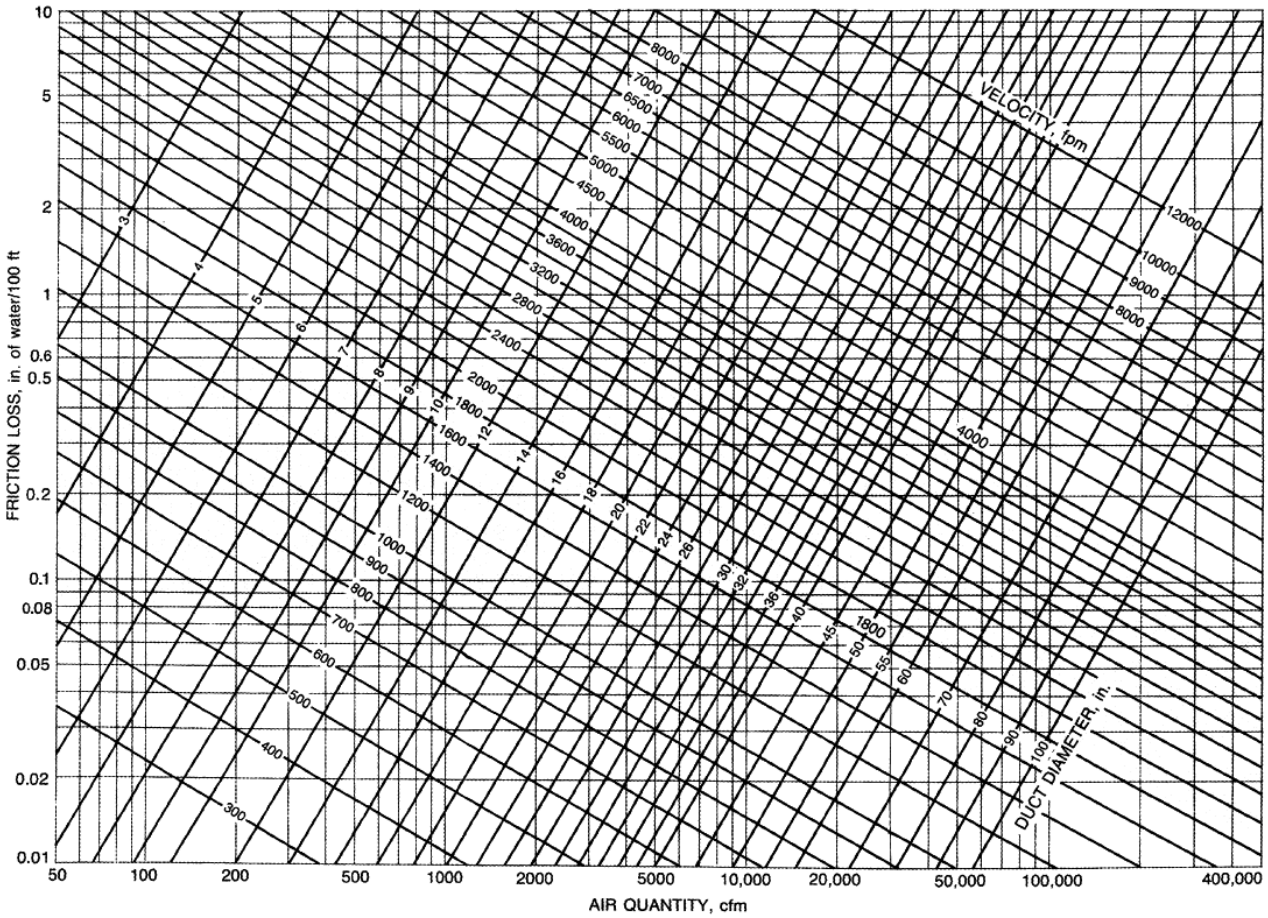
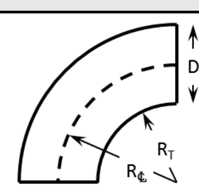


Figure 16b. Friction loss in round, galvanized, piper per 100 ft of length. Based on sea level-standard air conditions @ 70 °F and 29.92 in. Hg. Reprinted with permission from ASHRAE Handbook ©ASHRAE, www.ashrae.org. 2009. ASHRAE Handbook—Fundamentals (I-P).

Elbow Losses. Elbows in a piping system greatly increase the system's static pressure. More abrupt elbows with shorter radii have greater losses than do longer radius elbows. The losses due to round elbows are often expressed in terms of the equivalent length of straight pipe that would cause the same amount of static pressure loss. Equivalent length is usually expressed in terms of Le/D, where Le is the equivalent length and D is the diameter of the pipe. Table 8 shows Le/D for 90° elbows (round duct). If the centerline radius or throat radius of an elbow is known, then the equivalent straight pipe length of the elbow can be determined by multiplying Le/D from Table 8 by the pipe diameter.

Table 8. Pressure loss of 90° elbows[z] in terms of equivalent length (Le) in pipe diameters (D)

	Miter	Short Radius		Long Radius		
Centerline Radius Ratio ($R_{℄}/D$)		**0.75**	**1**	**1.5**	**2**	**2.5**
Throat Radius Ratio (R_T/D)		**0.25**	**0.5**	**1**	**1.5**	**2**
Equivalent Length (Le/D)	**65**	**23**	**17**	**12**	**10**	**8.5**



[z] **For bends less than 90°, the equivalent straight-pipe loss is proportional to the bend angle/90.**

Example 11. Find the Le/D from Table 8 and calculate the equivalent length of a 400-mm diameter (16in. diameter) 90° elbow with centerline radius ratio equal to 1½.

From Table 8, the Le/D for a pipe with 1½ centerline radius ratio is 12. Equivalent length is then

Le = 12*400 mm = 4800 mm or 4.8 m

or

Le = 12*16 in. = 192 in. or 16 ft

The elbow has the same resistance to air flow as 4.8 m (16 ft) of straight pipe. A system containing five such elbows will have an increased resistance equal to that of 60 diameters or 24 m (80 ft) of straight pipe.

As seen in Table 8, elbows add considerable pressure loss to a system and the loss is greater for shorter radius elbows. Also, for bends of less than 90° the equivalent straightpipe resistance is in proportion to the bend. For example, a 60° bend will have two-thirds of the equivalent resistance of a 90° bend. Therefore, equipment should be positioned and systems designed to minimize elbows.

PNEUMATIC SYSTEM DESIGN

In cotton gins, air and material are generally directed through 18- to 24-gauge sheet metal pipes that vary in diameter from approximately 200 to 900 mm (8-36 in.) Piping systems must be planned carefully to achieve desired results and properly maintained to avoid excessive horsepower consumption. Useful rules to follow are:

1. Make piping as simple and direct as possible, eliminating unnecessary elbows and valves.
2. Keep pipe joints as airtight and rigid as possible to minimize air leakage, reduce horsepower requirements, and save money for the ginner. However, some leakage is unavoidable. Therefore, air flow calculations should make provisions for normal leakage. Up to 35% leakage can occur through vacuum droppers. Tower dryers, rock and boll traps, and cylinder cleaners also frequently leak. Table 9 shows typical leakage values for gin machinery. Provisions must be made when sizing the fan to account for such leakage.
3. Properly maintain flights on vacuum droppers and flashings on separators to minimize leakage.
4. Reduce fan speeds to reduce excessive air flow and lower power consumption. Cutting holes in pipes to relieve pressures or to increase air flow rates is unsatisfactory.
5. Excessively high conveying velocities should be avoided to protect seed quality and reduce static pressure losses and energy consumption. Table 2 shows recommended ranges of conveying air velocities for various cotton gin materials and systems. Air velocities used for conveying seed cotton and seed should be maintained within recommended ranges to avoid cracking or shattering seed when the material impacts sharpturn elbows, separator reels, and other objects.
6. It is good practice to use long radius elbows in lines conveying seed cotton and seed. This consideration is especially important for small-pipe, high-pressure seed handling systems.
7. Avoid elbows immediately before the fan inlet—allow at least a short section of straight pipe. Similarly, install a long section of straight pipe on a centrifugal fan outlet, or orient the elbow to complement the rotation of the fan. Also, a long straight section of pipe before a cylinder cleaner inlet will produce a more even distribution of seed cotton across the cleaner.
8. Consider using a control system that reduces air flow rates when little or no cotton is flowing and increases air flow rates at maximum ginning rates to promote efficient operation and possibly reduce electricity use.

Table 9. Typical air leakage for gin machinery

Machinery	Leakage	
	m^3/min	ft^3/min
Blowbox	**14.1**	**500**
Incline Cleaners	**42.5-99.1**	**1500-3500**
Separators	**42.5-70.8**	**1500-2500**
Vacuum Droppers	**14.2-28.3**	**500-1000**
Airjet Lint Cleaners	**28.3**	**1000**
Condensers (Lint Cleaner & Battery Condenser)	**56.6-85.0**	**2000-3000**

The number, size, and operating rate of gin stands will determine the required flow rate of seed cotton in the gin. This flow rate and the layout of gin machinery will determine the dimensions and arrangement of the piping. The air pressures needed to overcome the resistance of equipment and pipes and the quantities of cotton needed to supply gins will determine the required fan capacity.

The following example illustrates air system design, bringing together many of the principles previously discussed.

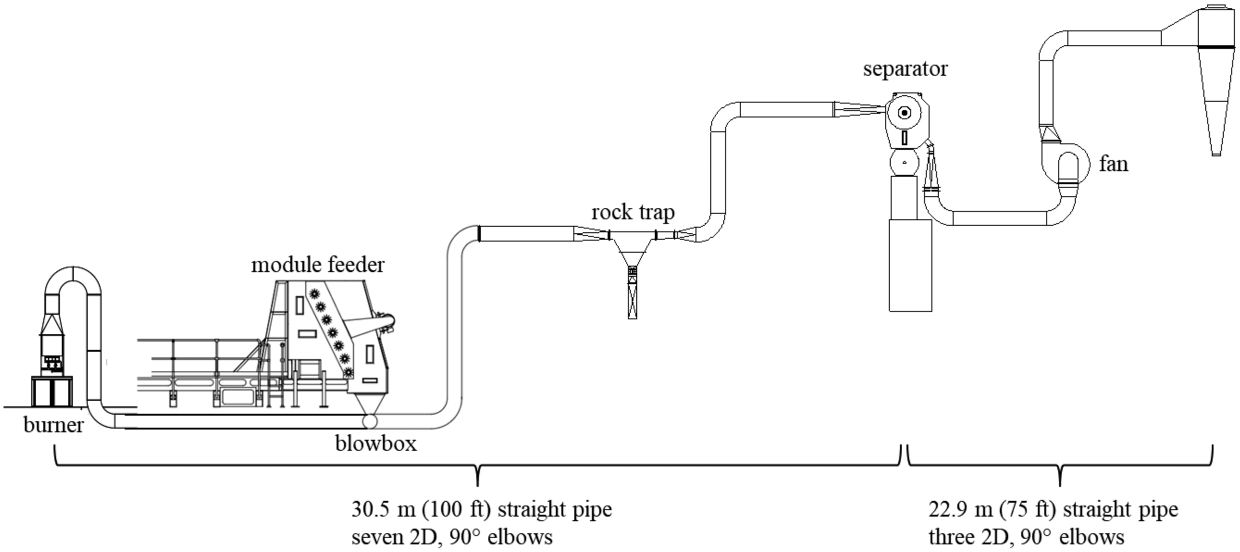


Figure 17. Unloading system diagram for Example 12.

Example 12: Determine pipe size and fan characteristics for an unloading system, shown in Fig. 17, to process 30 bales per hour of machine-picked cotton. Assume 590 kg (1,300 lb) of seed cotton per bale.

The mass of seed cotton conveyed per minute:

$$\frac{30 \text{ bales}}{\text{hr}} \times \frac{590 \text{ kg}}{\text{bale}} \times \frac{\text{hr}}{60 \text{ min}} = 295 \text{ kg / min}$$

or

$$\frac{30 \text{ bales}}{\text{hr}} \times \frac{1300 \text{ lb}}{\text{bale}} \times \frac{\text{hr}}{60 \text{ min}} = 650 \text{ lb / min}$$

From Table 3 for seed cotton unloading and conveying, use the air requirement of 1.25 m^3 per kg (20 ft^3 per lb) of seed cotton. The required air flow rate to convey the seed cotton from the module feeder to the separator:

$$\frac{295 \text{ kg}}{\text{min}} \times \frac{1.25 \text{ m}^3}{\text{kg}} = 369 \text{ m}^3 \text{ / min}$$

or

$$\frac{650 \text{ lb}}{\text{min}} \times \frac{20 \text{ ft}^3}{\text{lb}} = 13000 \text{ ft}^3 \text{ / min}$$

Determine the pipe diameter needed to convey the seed cotton at recommended velocity from the module feeder to the separator. The recommended air velocity for seed cotton conveying in Table 2 is 20.3 to 25.4 m/s (4,000-5,000 ft/min), so use approximately 23 m/s (4,500 ft/min). Using Table 1, move right on the row with 23-m/s (4,400-ft/min) air velocity to find an air flow rate nearest the required rate of 369 m^3/min (13000 ft^3/min), which is 358 m^3/min (12695 ft^3/min) in the column for 575-mm (23-in.) diameter pipe with 0.260-m^2 (2.885-ft^2) cross-sectional area. Calculate the actual air velocity in the selected pipe using Equation 1 to ensure that the conveying velocity will be within the recommended range from Table 2 (20.3-25.4 m/s [4,000-5,000 ft/min]):

$$\frac{369\ \text{m}^3/\text{min}}{0.260\ \text{m}^2 \times 60\ \text{s}/\text{min}} = 23.6\ \text{m}/\text{s}$$

or

$$\frac{13000\ \text{ft}^3/\text{min}}{2.885\ \text{ft}^2} = 4506\ \text{ft}/\text{s}$$

The next larger size pipe [600 mm (24 in.)] would also suffice and the actual conveying velocity [21.7 m/s (4,137 ft/min)] would be near the bottom of the recommended velocity range.

To ensure there is sufficient air flow at the module feeder, leakage at the separator must be included in the air flow rate for the section from the separator to the cyclone. On the suction side of the fan, leakage will be into the system. Table 9 shows the typical leakage for a separator is 57 m^3/min (2,000 ft^3/min). The air flow rate for the pipe from the separator to the cyclone:

$$369\ \text{m}^3/\text{min} + 57\ \text{m}^3/\text{min} = 426\ \text{m}^3/\text{min}$$

or

$$13000\ \text{ft}^3/\text{min} + 2000\ \text{ft}^3/\text{min} = 15000\ \text{ft}^3/\text{min}$$

Using this larger air flow rate, determine the pipe diameter needed to convey the dust and fine trash from the separator to the cyclone. The recommended air velocity for trash conveying is the same as seed cotton, 20.3 to 25.4 m/s (4,000-5,000 ft/min). Using Table 1, move right on the row with 23-m/s (4,400-ft/min) air velocity to find an air flow rate nearest the required rate of 426 m^3/min (15,000 ft^3/min), which is 423 m^3/min (14,999 ft^3/min) in the column for 625-mm (25-in.) diameter pipe with 0.307-m^2 (3.409-ft^2) cross-sectional area. Calculate the actual air velocity in the selected pipe using Equation 1 to ensure that the conveying velocity will be within the recommended range from Table 2 (20.3-25.4 m/s [4,000-5,000 ft/min]):

$$\frac{426\ \text{m}^3/\text{min}}{0.307\ \text{m}^2 \times 60\, s/min} = 23.1\ \text{m}/\text{s}$$

or

$$\frac{15000\ \text{ft}^3/\text{min}}{3.409\ \text{ft}^2} = 4400\ \text{ft}/\text{s}$$

Determine the straight pipe friction pressure loss for the two sections of pipe. For the 575-mm (23-in.) diameter section of pipe from the burner to separator, in Fig. 16 move up from the x-axis at flow rate of 369 m^3/min or 6,150 L/s (13,000 ft^3/min) to the intersection with 575-mm (23-in.) pipe diameter. Check that this point agrees with the intersection of 575-mm (23-in.) pipe diameter and 23.6-m/s (4,506-ft/min) actual air velocity. Moving left to the y-axis, the corresponding friction loss per length of pipe is approximately 8.5 Pa/m (0.95 in. H_2O/100 ft). The straight pipe friction pressure loss for the 30.5-m (100-ft) section of pipe from the burner to separator:

$$30.5\ \text{m} \times 8.5\ \text{Pa}/\text{m} = 259\ \text{Pa}$$

or

$$100\ \text{ft} \times 0.95\ \text{in. H}_2\text{O}/100\ \text{ft} = 0.95\ \text{in. H}_2\text{O}$$

For the 625-mm (25-in.) diameter section of pipe from the separator to the cyclone, in Fig. 16 move up from the x-axis at flow rate of 426 m^3/min or 7,100 L/s (15,000 ft^3/min) to the intersection with 625-mm (25-in.) pipe diameter. Check that this point agrees with the intersection of 625-mm (25-in.) pipe diameter and 23.1-m/s (4,400-ft/min) actual air velocity. Moving left to the y-axis, the corresponding friction loss per length of pipe is approximately 7.5 Pa/m (0.80 in. H_2O/100 ft). The straight pipe friction pressure loss for the 22.9-m (75-ft) section of pipe from the burner to separator:

$$22.9\ \text{m} \times 7.5\ \text{Pa}/\text{m} = 172\ \text{Pa}$$

or

$$75 \text{ ft} \times 0.80 \text{ in. } H_2O / 100 \text{ ft} = 0.60 \text{ in. } H_2O$$

Determine pressure loss for the 90°, 2D radius elbows. From Table 8, a 2D centerline radius ratio elbow has equivalent length 10D. The equivalent length of the seven 575-mm (23-in.) diameter elbows between the burner and separator:

$$7 \text{ elbows} \times 10 \times 575 \text{ mm} = 40,250 \text{ mm} = 40.25 \text{ m}$$
or
$$7 \text{ elbows} \times 10 \times 23 \text{ in.} = 1610 \text{ in.} = 134 \text{ ft}$$

The elbow pressure loss between the burner and separator using the equivalent length and friction loss per length of pipe found above:

$$40.25 \text{ m} \times 8.5 \text{ Pa / m} = 342 \text{ Pa}$$
or
$$134 \text{ ft} \times 0.95 \text{ in. } H_2O / 100 \text{ ft} = 1.27 \text{ in.} H_2O$$

Similarly, the equivalent length of the three 625-mm (25-in.) diameter elbows between the separator and cyclone:

$$3 \text{ elbows} \times 10 \times 625 \text{ mm} = 18,750 \text{ mm} = 18.75 \text{ m}$$
or
$$3 \text{ elbows} \times 10 \times 25 \text{ in.} = 750 \text{ in.} = 62.5 \text{ ft}$$

The elbow pressure loss between the separator and cyclone using the equivalent length and friction loss per length of pipe found above:

$$18.75 \text{ m} \times 7.5 \text{ Pa / m} = 141 \text{ Pa}$$
or
$$62.5 \text{ ft} \times 0.80 \text{ in. } H_2O / 100 \text{ ft} = 0.50 \text{ in.} H_2O$$

Determine the total static pressure drop for the system. Summing the pressure loss for the straight pipe, elbows, and friction loss for the other components from Table 7:

Burner to separator:			
	Burner	373 Pa	(1.5 in. H_2O)
369 m^3/min (13000 ft^3/min)	Straight pipe: 30.5 m (100 ft)	259 Pa	(0.95 in. H_2O)
575-mm (23-in.) diameter	Elbows: 7	342 Pa	(1.27 in. H_2O)
	Blowbox at module feeder	500 Pa	(2 in. H_2O)
	Rock trap	500 Pa	(2 in. H_2O)
Separator to cyclone:			
	Separator	1000 Pa	(4 in. H_2O)
426 m^3/min (15000 ft^3/min)	Straight pipe: 22.9 m (75 ft)	172 Pa	(0.60 in. H_2O)
625-mm (25-in.) diameter	Elbows: 3	141 Pa	(0.50 in. H_2O)
	Cyclone	1120 Pa	(4.5 in. H_2O)
	Total system static pressure	4407 Pa	(17.3 in. H_2O)

Determine the fan wheel speed and motor power. Assume selected fan in Fig. 12. The calculated static pressure, 4,407 Pa (17.3 in. H_2O), is between 3,981 Pa (16 in. H_2O) and 4,479 Pa (18 in. H_2O) in Fig. 12. Also, the fan air flow rate of 426 m^3/min (15,000 ft^3/min) is between 411- and 439-m^3/min (14,500-15,500 ft^3/min) air flow rates listed in the far left column. Interpolation is necessary to estimate the fan speed and power requirements for the system. For this example, round up to the slightly higher value for air flow rate and interpolate between the two static pressures. In the 439-m^3/min (15,500-ft^3/min) air flow rate row, the fan speed and power listed under the 3,981-Pa (16-in. H_2O) static pressure column are 1,418-rpm fan speed and 49.33 kW (66.15 hp) and the fan speed and power listed under the 4,479-Pa (18-in. H_2O) static pressure column are 1,484-rpm fan speed and 54.35 kW (72.88 hp). To interpolate between the two static pressure values, use the following equation:

$$y = y_1 + (x - x_1)\frac{(y_2 - y_1)}{(x_2 - x_1)} \qquad \text{(Eq. 8)}$$

where:

(x_1, y_1) and (x_2, y_2) are the known values

y is the unknown value at x

For this problem the x, x_1, and x_2 and y, y_1, and y_2 values are as follows:

Static pressure	Fan speed	Power
x_1 = 3,981 Pa (16 in. H_2O)	y_1 = 1,418 rpm	y_1 = and 49.33 kW (66.15 hp)
x = 4,407 Pa (17.3 in. H_2O)	y = ?	y = ?
x_2 = 4,479 Pa (18 in. H_2O)	y_2 = 1,484 rpm	y_2 = 54.35 kW (72.88 hp)

Solving for fan speed:

$$y = 1418\text{ rpm} + (4407\text{ Pa} - 3981\text{ Pa})\frac{(1484\text{ rpm} - 1418\text{ rpm})}{(4479\text{ Pa} - 3981\text{ Pa})} = 1474\text{ rpm}$$

or

$$y = 1418\text{ rpm} + (17.3\text{ in.H}_2\text{O} - 16\text{ in.H}_2\text{O})\frac{(1484\text{ rpm} - 1418\text{ rpm})}{(18\text{ in.H}_2\text{O} - 16\text{ in.H}_2\text{O})} = 1461\text{ rpm}$$

Solving for power:

$$y = 49.33\text{ kW} + (4407\text{ Pa} - 3981\text{ Pa})\frac{(54.35\text{ kW} - 49.33\text{ kW})}{(4479\text{ Pa} - 3981\text{ Pa})} = 53.62\text{ kW}$$

or

$$y = 66.15\text{ hp} + (17.3\text{ in.H}_2\text{O} - 16\text{ in.H}_2\text{O})\frac{(72.88\text{ hp} - 66.15\text{ hp})}{(18\text{ in.H}_2\text{O} - 16\text{ in.H}_2\text{O})} = 70.52\text{ hp}$$

The estimated fan speed and power would be approximately 1,470 rpm and 55 kW (75 hp). For a more accurate estimate, two-way interpolation between the values in Fig. 12 is required. Also, the air flow rate in the pipe between the burner and module feeder would be less than 369 m^3/min (13000 ft^3/min) due to leakage at the module feeder blowbox. In that section no material is conveyed. Therefore, the air velocity can be reduced in that section by increasing pipe diameter, which would reduce pressure drop and fan power requirements. Preferably, work with the fan manufacturer who has the technical knowledge and experience to size and select the correct fan.

Pneumatic conveying systems are used extensively in the cotton gin, consume more than 50% of the electrical energy used at the gin, and play a critical role in proper drying of seed cotton. Pneumatic systems must be designed correctly and maintained properly to ensure efficient gin operation, reduce energy consumption, and limit downtime.

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APPENDIX A. DUCT TRAVERSE

If the ideal location for a center-line Pitot air measurement in a long-straight conveying duct at least 1.5 pipe diameters upstream and 8.5 pipe diameters downstream from any obstruction or disturbance such as a fan, valve, or elbow (Fig. 4) cannot be identified or if uniform air velocity is not observed as described earlier in the Air Measurements section, then the air velocity must be averaged across the cross-section of the duct to obtain an accurate velocity measurement. This is accomplished using a duct traverse of air readings, which is a set of air measurements using a designated pattern of measurement points taken across the duct cross-section.

For round ducts, traverse measurements should be taken in two plains that pass through the centerline of the duct and are at 90º to each other (Fig. A1). Depending on the reference, there are many different criteria for determining the number of traverse points. The ACGIH (1998) recommends 10 points in each plane for round ducts larger than 150 mm (6 in.) and 20 points in each plane for large round ducts greater than 1200 mm (48 in.). Table A1 shows the distance from the round duct wall to the measurement location for a 10-point traverse.

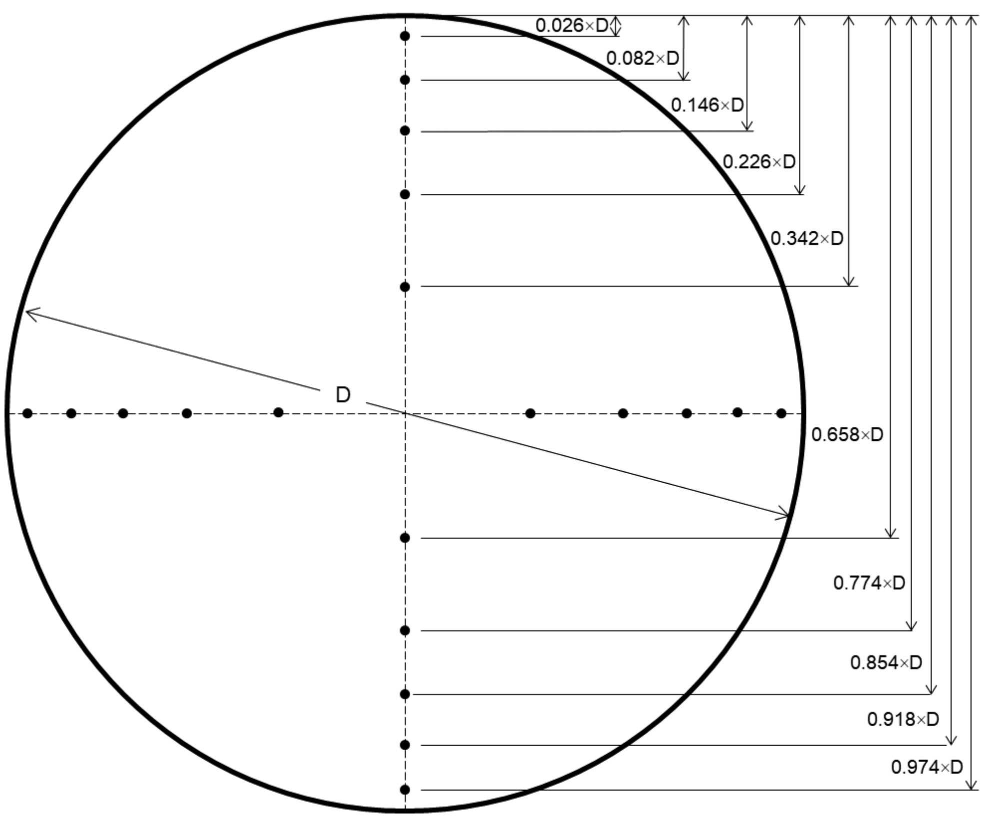

Figure A1. Locations for a 10-point duct traverse in a round duct with diameter D.

Table A1a. Distance from wall of round duct to point of duct traverse measurement for 10-point traverse—SI units.

Duct Dia. (mm)	Traverse Point									
	1	2	3	4	5	6	7	8	9	10
	Distance from duct wall (mm)									
D	0.026×D	0.082×D	0.146×D	0.226×D	0.342×D	0.658×D	0.774×D	0.854×D	0.918×D	0.974×D
200	5	16	29	45	68	132	155	171	184	195
225	6	18	33	51	77	148	174	192	207	219
250	7	21	37	57	86	165	194	214	230	244
275	7	23	40	62	94	181	213	235	252	268
300	8	25	44	68	103	197	232	256	275	292
325	8	27	47	73	111	214	252	278	298	317
350	9	29	51	79	120	230	271	299	321	341
375	10	31	55	85	128	247	290	320	344	365
400	10	33	58	90	137	263	310	342	367	390
425	11	35	62	96	145	280	329	363	390	414
450	12	37	66	102	154	296	348	384	413	438
475	12	39	69	107	162	313	368	406	436	463
500	13	41	73	113	171	329	387	427	459	487
525	14	43	77	119	180	345	406	448	482	511
550	14	45	80	124	188	362	426	470	505	536
575	15	47	84	130	197	378	445	491	528	560
600	16	49	88	136	205	395	464	512	551	584
625	16	51	91	141	214	411	484	534	574	609
650	17	53	95	147	222	428	503	555	597	633
675	18	55	99	153	231	444	522	576	620	657
700	18	57	102	158	239	461	542	598	643	682
725	19	59	106	164	248	477	561	619	666	706
750	20	62	110	170	257	494	581	641	689	731
775	20	64	113	175	265	510	600	662	711	755
800	21	66	117	181	274	526	619	683	734	779
850	22	70	124	192	291	559	658	726	780	828
900	23	74	131	203	308	592	697	769	826	877
950	25	78	139	215	325	625	735	811	872	925
1000	26	82	146	226	342	658	774	854	918	974
1050	27	86	153	237	359	691	813	897	964	1023
1100	29	90	161	249	376	724	851	939	1010	1071
1150	30	94	168	260	393	757	890	982	1056	1120
1200	31	98	175	271	410	790	929	1025	1102	1169

Table A1b. Distance from wall of round duct to point of duct traverse measurement for 10-point traverse—USC units

Duct Dia. (in.)	Traverse Point									
	1	2	3	4	5	6	7	8	9	10
	Distance from duct wall (in.)									
D	0.026×D	0.082×D	0.146×D	0.226×D	0.342×D	0.658×D	0.774×D	0.854×D	0.918×D	0.974×D
8	1/4	5/8	1 1/8	1 3/4	2 3/4	5 1/4	6 1/4	6 7/8	7 3/8	7 3/4
9	1/4	3/4	1 3/8	2	3 1/8	5 7/8	7	7 5/8	8 1/4	8 3/4
10	1/4	7/8	1 1/2	2 1/4	3 3/8	6 5/8	7 3/4	8 1/2	9 1/8	9 3/4
11	1/4	7/8	1 5/8	2 1/2	3 3/4	7 1/4	8 1/2	9 3/8	10 1/8	10 3/4
12	1/4	1	1 3/4	2 3/4	4 1/8	7 7/8	9 1/4	10 1/4	11	11 3/4
13	3/8	1 1/8	1 7/8	3	4 1/2	8 1/2	10	11 1/8	11 7/8	12 5/8
14	3/8	1 1/8	2	3 1/8	4 3/4	9 1/4	10 7/8	12	12 7/8	13 5/8
15	3/8	1 1/4	2 1/4	3 3/8	5 1/8	9 7/8	11 5/8	12 3/4	13 3/4	14 5/8
16	3/8	1 1/4	2 3/8	3 5/8	5 1/2	10 1/2	12 3/8	13 5/8	14 3/4	15 5/8
17	1/2	1 3/8	2 1/2	3 7/8	5 7/8	11 1/8	13 1/8	14 1/2	15 5/8	16 1/2
18	1/2	1 1/2	2 5/8	4 1/8	6 1/8	11 7/8	13 7/8	15 3/8	16 1/2	17 1/2
19	1/2	1 1/2	2 3/4	4 1/4	6 1/2	12 1/2	14 3/4	16 1/4	17 1/2	18 1/2
20	1/2	1 5/8	2 7/8	4 1/2	6 7/8	13 1/8	15 1/2	17 1/8	18 3/8	19 1/2
21	1/2	1 3/4	3 1/8	4 3/4	7 1/8	13 7/8	16 1/4	17 7/8	19 1/4	20 1/2
22	5/8	1 3/4	3 1/4	5	7 1/2	14 1/2	17	18 3/4	20 1/4	21 3/8
23	5/8	1 7/8	3 3/8	5 1/4	7 7/8	15 1/8	17 3/4	19 5/8	21 1/8	22 3/8
24	5/8	2	3 1/2	5 3/8	8 1/4	15 3/4	18 5/8	20 1/2	22	23 3/8
25	5/8	2	3 5/8	5 5/8	8 1/2	16 1/2	19 3/8	21 3/8	23	24 3/8
26	5/8	2 1/8	3 3/4	5 7/8	8 7/8	17 1/8	20 1/8	22 1/4	23 7/8	25 3/8
27	3/4	2 1/4	4	6 1/8	9 1/4	17 3/4	20 7/8	23	24 3/4	26 1/4
28	3/4	2 1/4	4 1/8	6 3/8	9 5/8	18 3/8	21 5/8	23 7/8	25 3/4	27 1/4
29	3/4	2 3/8	4 1/4	6 1/2	9 7/8	19 1/8	22 1/2	24 3/4	26 5/8	28 1/4
30	3/4	2 1/2	4 3/8	6 3/4	10 1/4	19 3/4	23 1/4	25 5/8	27 1/2	29 1/4
32	7/8	2 5/8	4 5/8	7 1/4	11	21	24 3/4	27 3/8	29 3/8	31 1/8
34	7/8	2 3/4	5	7 5/8	11 5/8	22 3/8	26 3/8	29	31 1/4	33 1/8
36	7/8	3	5 1/4	8 1/8	12 1/4	23 3/4	27 7/8	30 3/4	33	35 1/8
38	1	3 1/8	5 1/2	8 5/8	13	25	29 3/8	32 1/2	34 7/8	37
40	1	3 1/4	5 7/8	9	13 5/8	26 3/8	31	34 1/8	36 3/4	39
42	1 1/8	3 1/2	6 1/8	9 1/2	14 3/8	27 5/8	32 1/2	35 7/8	38 1/2	40 7/8
44	1 1/8	3 5/8	6 3/8	10	15	29	34	37 5/8	40 3/8	42 7/8
46	1 1/4	3 3/4	6 3/4	10 3/8	15 3/4	30 1/4	35 5/8	39 1/4	42 1/4	44 3/4
48	1 1/4	3 7/8	7	10 7/8	16 3/8	31 5/8	37 1/8	41	44 1/8	46 3/4

The same equipment as described in the Air Measurements section should be used for the duct traverse. Velocity pressure measurements should be taken at each traverse location in both plains. Then air velocity should be calculated for each measurement using the follow equations for standard air [density = 1.2 kg/m^3 (0.075 lb/ft^3)] conditions at 21 °C and 101.3 kPa (70 °F and 29.92-in. Hg):

$$\text{air velocity}\left(\text{m / s}\right)=1.29\sqrt{\text{velocity pressure}\left(\text{Pa}\right)} \qquad \text{(Eq. A1)}$$

or

$$\text{air velocity}\left(\text{ft / min}\right)=4005\sqrt{\text{velocity pressure}\left(\text{in. H}_2\text{O}\right)} \qquad \text{(Eq. A2)}$$

Note: Do not use Table 1 to determine air velocity as it is for centerline measurements only.

Velocity measurements should then be averaged to obtain the average air velocity for the duct. Air flow rate can then be calculated using Equations 1 and 2, shown earlier. Also, air velocity and air flow can be corrected for conditions (temperature and elevation) that differ from standard conditions in the same manner as shown the Air Measurements section.

Duct traverses for rectangular ducts are done similarly to those for round ducts. However, the duct should be divided into at least 16 small rectangles of equal areas and measure the velocity pressure at the center of each (Fig. A2). Enough traverse points should be used so that measurement locations are no more than 150 mm (6 in.) apart.

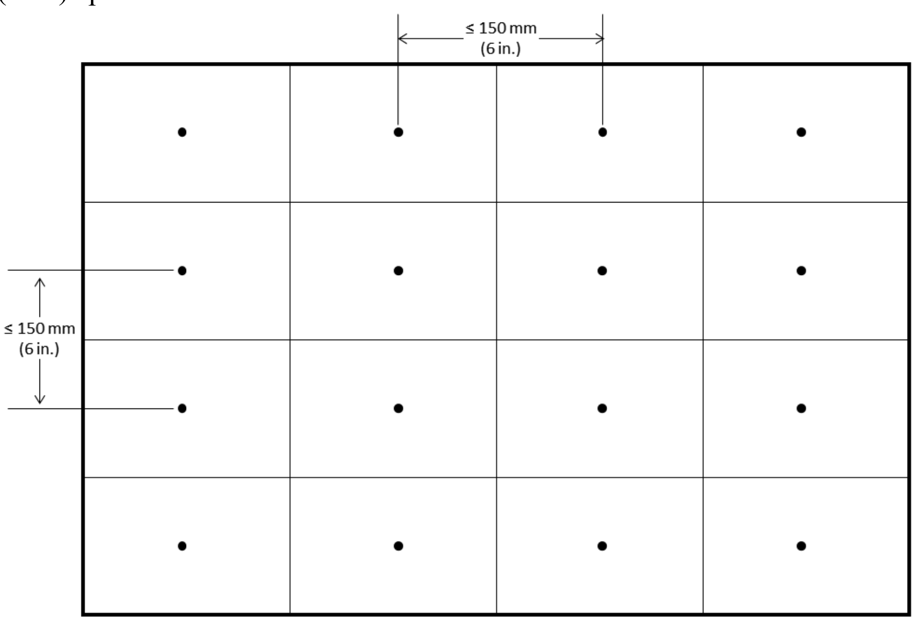

Figure A2. Example locations for a 16-point duct traverse in a rectangular duct. Each rectangle is equal area.