4

Compressed Air Distribution (Systems)

COMPRESSED AIR DISTRIBUTION SYSTEMS

When a compressed air distribution system is properly designed, installed, operated and maintained, it is a major source of industrial power, possessing many inherent advantages. Compressed air is safe, economical, adaptable and easily transmitted and provides labor saving power. The cost of a complete compressed air system and pneumatic tools is relatively small in comparison with the savings effected by their use.

Object of the Compressed Air Distribution System

The primary object of a compressed air distribution system is to transport the compressed air from its point of production (compressors) to its points of use (applications) in sufficient quantity and quality and at adequate pressure for efficient operation of air tools and other pneumatic devices. However, many other considerations come into the design of the system to ensure the efficiency and safety of the total system. These will be discussed in this chapter. These include:

- Air volume flow rate
- Air pressure requirements
- Type(s) and number of compressors
- Air quality
- Air system efficiency
- Air system safety
- Air system layout
- Air volume flow rate requirements

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Air Volume Flow Rate Requirements

The proper capacity to install is a vital and basic question and often misunderstood. The capacity rating of air compressors generally is published in terms of “free air,” which is at atmospheric conditions of pressure, temperature and relative humidity and not at the pressure, temperature and relative humidity required at the air tool or pneumatic device to be operated.

The Applications chapter of this book contains many illustrations of current uses of compressed air power. The air tools chapter also provides much useful information on applications of air powered tools and other pneumatic devices.

A study of air-operated devices in a typical manufacturing plant will show that some of these devices operate almost constantly while others operate infrequently but may require a relatively large volume of air while in use. It also will be found that the amount of air actually used by the individual devices will vary considerably in different applications. The total air requirement therefore should not be the total of the individual maximum requirement but the sum of the average air consumption of each. Sufficient controlled storage capacity of compressed air also is essential to meet short-term high volume demands.

Recommendations for efficient components for the compressed air system have been discussed in earlier chapters. This chapter deals with the compressed air distribution system which feeds the production operation. Proper design of the distribution system is essential to avoid energy waste and to ensure proper use of all pneumatic devices.

Determination of the average air consumption is facilitated by the use of the concept of load factor. Pneumatic devices generally are operated only intermittently and often are operated at less than full load capacity. The ratio of actual air consumption to the maximum continuous full load air consumption, each measured in cubic feet per minute of free air, is known as the load factor. It is essential that the best possible determination or estimate of load factor be used in arriving at the plant capacity needed.

Two items are involved in the load factor. The first is the time factor, which is the percentage of work time during which a device actually is in use. The second is the work factor, which is the percentage of the air required for maximum possible output of work per minute that is required for the work actually being performed by the device. For example, the air consumption of a grinder with full open throttle varies considerably, depending on how hard the operator applies the grinding wheel against the work piece. The work factor also is affected by the system operating pressure. For example, a system pressure of 125 psig will provide a work factor 22% higher than a system pressure of 100 psig. (See Table 4.10). The work factor therefore is the ratio (expressed as a percentage) of the air consumption under actual conditions of operation, to the air consumption when the tool is fully loaded. The load factor is the product of the time factor and the work factor. In one plant studied, the air actually consumed by 434 portable air tools on production work was only 15% of the total rated full time air consumption of all the tools.

In designing an entirely new compressed air distribution system, it is highly desirable to utilize experience with a similar plant. The established load factor can be used as the basis of a good estimate for the new system. A log of pressures throughout an existing facility will reveal trends, including peaks and lulls in demand and potential irregularities to be avoided in the new system. Another source of this type of information is the manufacturer of the air tools and pneumatic devices involved.

Table 4.1, shows the maximum air requirements of various tools and can be used for preliminary estimates. These figures are approximate and individual tools from different manufacturers may vary by more than 10% from the figures given. Since load factor may vary considerably from one plant to another, any general figures should be used with caution. For example, one manufacturer states that the compressor capacity should be about one third of the requirement of all the pneumatic tools. See Table 4.2. It is recommended that the manufacturer of each air tool, device or machine, be consulted as to recommended requirements. Table 4.1 should not be used for constant demand applications, including sandblasting requirements shown in Table 4.2.

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Table 4.1 Air Requirements of Various Tools

Tool	Free Air, cfm at 90 psig, 100% Load Factor
Grinders, 6" and 8" wheels	50
Grinders, 2" and 2 1/2" wheels	14-20
File and burr machines	18
Rotary sanders, 9" pads	53
Rotary sanders, 7" pads	30
Sand rammers and tampers,	
1" x 4" cylinder	25
1 1/4" x 5" cylinder	28
1 1/2" x 6" cylinder	39
Chipping hammers, weighing 10-13 lb	28-30
Heavy	39
Weighing 2-4 lb	12
Nut setters to 5/16" weighing 8 lb	20
Nut setters 1/2" to 3/4" weighing 18 lb	30
Sump pumps, 145 gal (a 50-ft head)	70
Paint spray, average	7
Varies from	2-20
Bushing tools (monument)	15-25
Carving tools (monument)	10-15
Plug drills	40-50
Riveters, 3/32"-1" rivets	12
Larger weighing 18-22 lb	35
Rivet busters	35-39
Wood borers to 1" diameter weighing 4 lb	40
2" diameter weighing 26 lb	80
Steel drills, rotary motors	
Capacity up to 1/4" weighing 1 1/4-4 lb	18-20
Capacity 1/4" to 3/8" weighing 6-8 lb	20-40
Capacity 1/2" to 3/4" weighing 9-14 lb	70
Capacity 7/8" to 1" weighing 25 lb	80
Capacity 1 1/4" weighing 30 lb	95
Steel drills, piston type	
Capacity 1/2" to 3/4" weighing 13-15 lb	45
Capacity 7/8" to 1 1/4" weighing 25-30 lb	75-80
Capacity 1 1/4" to 2" weighing 40-50 lb	80-90
Capacity 2" to 3" weighing 55-75 lb	100-110

Table 4.2 Cubic Feet of Air Per Minute Required By Sandblast

	Compressed Air Gage Pressure (psig)			
Nozzle Diameter	60	70	80	100
1/16"	4	5	5.5	6.5
3/32"	9	11	12	15
1/8"	17	19	21	26
3/16"	38	43	47	58
1/4"	67	76	85	103
5/16"	105	119	133	161
3/8"	151	171	191	232
1/2"	268	304	340	412

For tools used regularly on one operation, a study of active and inactive times may be made. Judgement may be exercised at this time as to the work factor to be applied if other than unity. If air requirements of a manufacturing process are evaluated on the basis of unit production in cubic feet of free air per piece produced, they may then be combined on the basis of total production to arrive at the average volume rate of air required.

Many pieces of production equipment are actuated by pneumatic cylinders. These include automatic feed devices, chucks, vises, clamps, presses, intermittent motion devices, both reciprocating and rotary, door openers and many other devices. Such devices usually have low air consumption and are themselves inexpensive. They find increasing use in automated production processes. Air consumption for such cylinders is shown in Table 4.3. This table shows only the theoretical volume swept out by the piston during one full stroke, which must be converted into a flow rate of free air. Many cylinders contain air cushioning chambers which increase the volume somewhat over the tabled figures. In addition, in actual use the air pressure to the cylinder may be throttled to a pressure considerably below the system line pressure. If a limit switch cuts off the air supply when a certain force is exerted by the cylinder, the corresponding pressure should be calculated and used rather than full line pressure in converting the tabled figures to free air conditions. In many applications the full available piston stroke is not needed. In fact, a reduced length of stroke may be an advantage in reducing operating time. The air consumption for such cases is calculated using only the actual stroke.

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Table 4.3 Volume of Compressed Air in Cubic Feet Required per Stroke to Operate Air Cylinder

Piston Diameter in Inches	Length of Stroke in Inches*											
	1	2	3	4	5	6	7	8	9	10	11	12
1 1/4	.00139	.00278	.00416	.00555	.00694	.00832	.00972	.0111	.0125	.0139	.0153	.01665
1 7/8	.00158	.00316	.00474	.00632	.0079	.00948	.01105	.01262	.0142	.0158	.0174	.01895
2	.00182	.00364	.00545	.00727	.0091	.0109	.0127	.0145	.01636	.0182	.020	.0218
2 1/8	.00205	.0041	.00615	.0082	.0103	.0123	.0144	.0164	.0185	.0205	.0226	.0244
2 1/4	.0023	.0046	.0069	.0092	.0115	.0138	.0161	.0184	.0207	.0230	.0253	.0276
2 3/8	.00256	.00512	.00768	.01025	.0128	.01535	.01792	.02044	.0230	.0256	.0282	.0308
2 1/2	.00284	.00568	.00852	.01137	.0142	.0171	.0199	.0228	.0256	.0284	.0312	.0343
2 5/8	.00313	.00626	.0094	.01254	.01568	.0188	.0219	.0251	.0282	.0313	.0345	.0376
2 3/4	.00343	.00686	.0106	.0137	.0171	.0206	.0240	.0272	.0308	.0343	.0378	.0412
2 7/8	.00376	.00752	.0113	.01503	.01877	.0226	.0263	.0301	.0338	.0376	.0413	.045
3	.00409	.00818	.0123	.0164	.0204	.0246	.0286	.0327	.0368	.0409	.0450	.049
3 1/8	.00443	.00886	.0133	.0177	.0222	.0266	.0310	.0354	.0399	.0443	.0488	.0532
3 1/4	.0048	.0096	.0144	.0192	.024	.0288	.0336	.0384	.0432	.0480	.0529	.0575
3 3/8	.00518	.01036	.0155	.0207	.0259	.031	.0362	.0415	.0465	.0518	.037	.062
3 1/2	.00555	.0112	.0167	.0222	.0278	.0333	.0389	.0445	.050	.0556	.061	.0644
3 5/8	.00595	.0119	.0179	.0238	.0298	.0357	.0416	.0477	.0536	.0595	.0655	.0715
3 3/4	.0064	.0128	.0192	.0256	.032	.0384	.0447	.0512	.0575	.064	.0702	.0766
3 7/8	.0068	.01362	.0205	.0273	.0341	.041	.0477	.0545	.0614	.068	.075	.082
4	.00725	.0145	.0218	.029	.0363	.0435	.0508	.058	.0653	.0725	.0798	.087
4 1/8	.00773	.01547	.0232	.0309	.0386	.0464	.0541	.0618	.0695	.0773	.0851	.092
4 1/4	.0082	.0164	.0246	.0328	.041	.0492	.0574	.0655	.0738	.082	.0903	.0985
4 3/8	.0087	.0174	.0261	.0348	.0435	.0522	.0608	.0694	.0782	.087	.0958	.1042
4 1/2	.0092	.0184	.0276	.0368	.046	.0552	.0643	.0735	.0828	.092	.101	.1105
4 5/8	.0097	.0194	.0291	.0388	.0485	.0582	.0679	.0775	.0873	.097	.1068	.1163
4 3/4	.01025	.0205	.0308	.041	.0512	.0615	.0717	.0818	.0922	.1025	.1125	.123
4 7/8	.0108	.0216	.0324	.0431	.054	.0647	.0755	.0862	.097	.108	.1185	.1295
5	.0114	.0228	.0341	.0455	.0568	.0681	.0795	.091	.1023	.114	.125	.136
5 1/8	.01193	.0239	.0358	.0479	.0598	.0716	.0837	.0955	.1073	.1193	.1315	.1435
5 1/4	.0125	.0251	.0376	.0502	.0627	.0753	.0878	.100	.1128	.125	.138	.151
5 3/8	.0131	.0263	.0394	.0525	.0656	.0788	.092	.105	.118	.131	.144	.158
5 1/2	.01375	.0275	.0412	.055	.0687	.0825	.0962	.110	.1235	.1375	.151	.165
5 5/8	.0144	.0288	.0432	.0575	.072	.0865	.101	.115	.1295	.144	.1585	.173
5 3/4	.015	.030	.045	.060	.075	.090	.105	.120	.135	.150	.165	.180
5 7/8	.0157	.0314	.047	.0628	.0785	.094	.110	.1254	.142	.157	.1725	.188
6	.0164	.032	.0492	.0655	.082	.0983	.1145	.131	.147	.164	.180	.197

Air turbines may be used for starting gas turbines and for other purposes. Air consumption of turbines may be calculated by the usual methods of thermodynamics. For single-stage impulse turbines with converging nozzles, the air consumption may be found by applying Fliegner's equation to the nozzles. The air turbine manufacturer can supply the needed data.

Other devices have an air flow condition approximating simple throttling. A steady jet used for blowing chips from a tool would fall within this classification. Another device with approximately the same flow characteristics is the vibrator actuated by a steel ball propelled around a closed circular track by means of an air jet. Table 4.4 may be used for estimating air flow through such devices. These data are not intended for use in air measurements and should be used only for estimating system air requirements.

Table 4.4 How to Determine Compressor Size Required

			Load Factor	CFM required*		
		Number of Tools	(per cent of time tools actually operated)	Per Tool When Operating	Total if All Tools Operated Simultaneously	Total Actually Used (A x B x C ÷ 100
Type of Tool	Location	(A)	(B)	(C)	(D)	(E)
Blowguns, chucks and vises	Machine Shop	4	25	25	100	25
8-in. grinders	Cleaning	10	50	50	500	250
Chippers	Cleaning	10	50	30	300	150
Hoists	Cleaning	2	10	35	70	7
Small screwdrivers	Assembly	20	25	12	240	60
Large nutsetters	Assembly	2	25	30	60	15
Woodborer	Shipping	1	25	30	30	$7\ ^1/_2$
Screwdriver	Shipping	1	20	30	30	6
Hoist	Shipping	1	20	40	40	8
Total		47			1270	$528\ ^1/_2$

*Cfm is cubic feet of free air per minute.
Note: Total of column (E) determines required compressor sizes.

Regenerative desiccant type compressed air dryers require purge air which may be as much as 15% of the rated dryer capacity and this must be added to the estimate of air required at points of use.

An often neglected consideration is system leaks. Theoretically, a new system should have no leaks but experience shows that most systems have varying amounts of leakage sources.

Electronic leak detectors are available and should be used on a regularly scheduled basis. It also can be useful to determine how long an air compressor runs to maintain system pressure during a shutdown period when there is no actual usage of compressed air.

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Air Pressure Requirements

This is one of the more critical factors in the design of an efficient compressed air distribution system. One problem is that the variety of points of application may require a variety of operating pressure requirements. Equipment manufacturers should be consulted to determine the pressure requirement at the machine, air tool or pneumatic device. If these operating pressure requirements vary by more than 20%, consideration should be given to separate systems. In a typical plant with an air distribution system operating at a nominal 100 psig, an increase of one half per cent in the air compressor energy costs is required for each additional 1 psi in system pressure. Operating the complete system at 20% higher pressure to accommodate one point of use, would result in the air compressor(s) using 10% more energy and an increase in work factor as previously noted. This, obviously, is to be avoided.

Allowance also must be made for pressure drops through compressed air treatment equipment, including air dryers and filters.

An inadequately sized piping distribution system will cause excessive pressure drops between the air compressors and the points of use, requiring the compressor to operate at a much higher pressure than at the points of use. This also requires additional energy. For example, if the distribution piping size is only half of the ideal, the cross-sectional area is only one fourth, resulting in velocities four times the ideal and sixteen times the pressure drop. In an air distribution system where a given pipe diameter piping may be sufficient, it should be remembered that the installation labor cost will be the same for double the pipe diameter and only the material cost will increase. The savings in energy costs from reduced pressure drop will repay the difference in material costs in a very short time. and could provide for future capacity.

Air velocity through the distribution piping should not exceed 1800 ft. per minute (30 ft. per sec.). One recommendation, to avoid moisture being carried beyond drainage drop legs in compressor room header upstream of dryer(s), is that the velocity should not exceed 1200 ft. per minute (20 ft. per sec.). Branch lines having an air velocity over 2000 ft. per minute, should not exceed 50 ft. in length. The system should be designed so that the operating pressure drop between the air compressor and the point(s) of use should not exceed 10% of the compressor discharge pressure.

Pressure loss in piping due to friction at various operational pressures is tabulated in Tables 4.5, 4.6, 4.7, 4.8 and 4.9 can be used to determine pipe sizes required for the system being designed. These tables are based upon non-pulsating flow in a clean, smooth pipe.

Table 4.5 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.96	10.0	1.53	0.43	0.10								
20	3.94	39.7	5.99	1.71	0.39	0.18							
30	5.89	....	13.85	3.86	0.88	0.40							
40	7.86	....	24.7	6.85	1.59	0.71	0.19						
50	9.84	....	38.6	10.7	2.48	1.10	0.30						
60	11.81	....	55.5	15.4	3.58	1.57	0.43						
70	13.75	....	....	21.0	4.87	2.15	0.57						
80	15.72	....	....	27.4	6.37	2.82	0.75						
90	17.65	....	....	34.7	8.05	3.57	0.57	0.37					
100	19.60	....	....	42.8	9.95	4.40	1.18						
125	19.4	....	....	46.2	12.4	6.90	1.83	0.14					
150	29.45	....	....	....	22.4	9.90	2.64	0.32					
175	34.44	....	....	....	30.8	13.40	3.64	0.43					
200	39.40	....	....	....	39.7	17.60	4.71	0.57					
250	49.20	....	....	....	....	27.5	7.37	0.89	0.21				
300	58.90	....	....	....	....	39.6	10.55	1.30	0.31				
350	68.8	....	....	....	....	54.0	14.4	1.76	0.42				
400	78.8	....	....	....	....	....	18.6	2.30	0.53				
450	88.4	....	....	....	....	....	23.7	2.90	0.70				
500	98.4	....	....	....	....	....	29.7	3.60	0.85				
600	118.1	....	....	....	....	....	42.3	5.17	1.22				
700	137.5	....	....	....	....	....	57.8	7.00	1.67				
800	157.2	....	....	....	....	....	....	9.16	2.18				
900	176.5	....	....	....	....	....	....	11.6	2.76				
1,000	196.0	....	....	....	....	....	....	14.3	3.40				
1,500	294.5	....	....	....	....	....	....	32.3	7.6	0.87	0.29		
2,000	394.0	....	....	....	....	....	....	57.5	13.6	1.53	0.36		
2,500	492	....	....	....	....	....	....	....	21.3	2.42	0.57	0.17	
3,000	589	....	....	....	....	....	....	....	30.7	3.48	0.81	0.24	
3,500	688	....	....	....	....	....	....	....	41.7	4.68	1.07	0.33	
4,000	788	....	....	....	....	....	....	....	54.5	6.17	1.44	0.44	
4,500	884	....	....	....	....	....	....	....	....	7.8	1.83	0.55	0.21
5,000	984	....	....	....	....	....	....	....	....	9.7	2.26	0.67	0.27
6,000	1,181	....	....	....	....	....	....	....	....	13.9	3.25	0.98	0.38
7,000	1,375	....	....	....	....	....	....	....	....	18.7	4.43	1.34	0.51
8,000	1,572	....	....	....	....	....	....	....	....	24.7	5.80	1.73	0.71
9,000	1,765	....	....	....	....	....	....	....	....	31.3	7.33	2.20	0.87
10,000	1,960	....	....	....	....	....	....	....	....	38.6	9.05	2.72	1.06
11,000	2,165	....	....	....	....	....	....	....	....	46.7	10.9	3.29	1.28
12,000	2,362	....	....	....	....	....	....	....	....	55.5	13.0	3.90	1.51
13,000	2,560	....	....	....	....	....	....	....	....	....	15.2	4.58	1.77
14,000	2,750	....	....	....	....	....	....	....	....	....	17.7	5.32	2.07
15,000	2,945	....	....	....	....	....	....	....	....	....	20.3	6.10	2.36
16,000	3,144	....	....	....	....	....	....	....	....	....	23.1	6.95	2.70
18,000	3,530	....	....	....	....	....	....	....	....	....	29.2	8.80	3.42
20,000	3,940	....	....	....	....	....	....	....	....	....	36.2	10.8	4.22
22,000	4,330	....	....	....	....	....	....	....	....	....	43.7	13.2	5.12
24,000	4,724	....	....	....	....	....	....	....	....	....	51.9	15.6	5.92
26,000	5,120	....	....	....	....	....	....	....	....	....	....	18.3	7.15
28,000	5,500	....	....	....	....	....	....	....	....	....	....	21.3	8.3
30,000	5,890	....	....	....	....	....	....	....	....	....	....	24.4	9.5

In psi in 1000 ft of pipe, 60 lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

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Table 4.6 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.55	7.90	1.21	0.34									
20	3.10	31.4	4.72	1.35	0.31								
30	4.65	70.8	10.9	3.04	0.69	0.31							
40	6.20	....	19.5	5.40	1.25	0.56							
50	7.74	....	30.5	8.45	1.96	0.87							
60	9.29	....	43.8	12.16	2.82	1.24	0.34						
70	10.82	....	59.8	16.6	3.84	1.70	0.45						
80	12.40	....	78.2	21.6	5.03	2.22	0.59						
90	13.95	....	....	27.4	6.35	2.82	0.75						
100	15.5	....	....	33.8	7.85	3.74	0.93						
125	19.4	....	....	46.2	12.4	5.45	1.44						
150	23.2	....	....	76.2	17.7	7.82	2.08						
175	27.2	....	....	....	24.8	10.6	2.87						
200	31.0	....	....	....	31.4	13.9	3.72	0.45					
250	38.7	....	....	....	49.0	21.7	5.82	0.70					
300	46.5	....	....	....	70.6	31.2	8.35	1.03					
350	54.2	....	....	....	....	42.5	11.4	1.39	0.33				
400	62.0	....	....	....	....	55.5	14.7	1.82	0.42				
450	69.7	....	....	....	....	....	18.7	2.29	0.55				
500	77.4	....	....	....	....	....	23.3	2.84	0.67				
600	92.9	....	....	....	....	....	33.4	4.08	0.96				
700	108.2	....	....	....	....	....	45.7	5.52	1.32				
800	124.0	....	....	....	....	....	59.3	7.15	1.72				
900	139.5	....	....	....	....	....	....	9.17	2.18				
1,000	155	....	....	....	....	....	....	11.3	2.68				
1,500	232	....	....	....	....	....	....	25.5	6.0	0.69			
2,000	310	....	....	....	....	....	....	45.3	10.7	1.21	0.29		
2,500	387	....	....	....	....	....	....	70.9	16.8	1.91	0.45		
3,000	465	....	....	....	....	....	....	....	24.2	2.74	0.64	0.19	
3,500	542	....	....	....	....	....	....	....	32.8	3.70	0.85	0.26	
4,000	620	....	....	....	....	....	....	....	43.0	4.87	1.14	0.34	
4,500	697	....	....	....	....	....	....	....	54.8	6.15	1.44	0.43	
5,000	774	....	....	....	....	....	....	....	67.4	7.65	1.78	0.53	0.21
6,000	929	....	....	....	....	....	....	....	....	11.0	2.57	0.77	0.29
7,000	1,082	....	....	....	....	....	....	....	....	14.8	3.40	1.06	0.40
8,000	1,240	....	....	....	....	....	....	....	....	19.5	4.57	1.36	0.54
9,000	1,395	....	....	....	....	....	....	....	....	24.7	5.78	1.74	0.69
10,000	1,550	....	....	....	....	....	....	....	....	30.5	7.15	2.14	0.84
11,000	1,710	....	....	....	....	....	....	....	....	36.8	8.61	2.60	1.01
12,000	1,860	....	....	....	....	....	....	....	....	43.8	10.3	3.08	1.19
13,000	2,020	....	....	....	....	....	....	....	....	51.7	12.0	3.62	1.40
14,000	2,170	....	....	....	....	....	....	....	....	60.2	14.0	4.20	1.63
15,000	2,320	....	....	....	....	....	....	....	....	68.5	16.0	4.82	1.84
16,000	2,480	....	....	....	....	....	....	....	....	78.2	18.2	5.48	2.13
18,000	2,790	....	....	....	....	....	....	....	....	....	23.0	6.95	2.70
20,000	3,100	....	....	....	....	....	....	....	....	....	28.6	8.55	3.33
22,000	3,410	....	....	....	....	....	....	....	....	....	34.5	10.4	4.04
24,000	3,720	....	....	....	....	....	....	....	....	....	41.0	12.3	4.69
26,000	4,030	....	....	....	....	....	....	....	....	....	48.2	14.4	5.6
28,000	4,350	....	....	....	....	....	....	....	....	....	55.9	16.8	6.3
30,000	4,650	....	....	....	....	....	....	....	....	....	64.2	19.3	7.5

In psi in 1000 ft of pipe, 80 lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 4.7 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.28	6.50	.99	0.28									
20	2.56	25.9	3.90	1.11	0.25	0.11							
30	3.84	58.5	9.01	2.51	0.57	0.26							
40	5.12	....	16.0	4.45	1.03	0.46							
50	6.41	....	25.1	9.96	1.61	0.71	0.19						
60	7.68	....	36.2	10.0	2.32	1.02	0.28						
70	8.96	....	49.3	13.7	3.16	1.40	0.37						
80	10.24	....	64.5.	17.8	4.14	1.83	0.49						
90	11.52	....	82.8	22.6	5.23	2.32	0.62						
100	12.81	....	....	27.9	6.47	2.86	0.77						
125	15.82	....	....	48.6	10.2	4.49	1.19						
150	19.23	....	....	62.8	14.6	6.43	1.72	0.21					
175	22.40	....	....	....	19.8	8.72	2.36	0.28					
200	25.62	....	....	....	25.9	11.4	3.06	0.37					
250	31.64	....	....	....	40.4	17.9	4.78	0.58					
300	38.44	....	....	....	58.2	25.8	6.85	0.84	0.20				
350	44.80	....	....	....	....	35.1	9.36	1.14	0.27				
400	51.24	....	....	....	....	45.8	12.1	1.50	0.35				
450	57.65	....	....	....	....	58.0	15.4	1.89	0.46				
500	63.28	....	....	....	....	71.6	19.2	2.34	0.55				
600	76.88	....	....	....	....	....	27.6	3.36	0.79				
700	89.60	....	....	....	....	....	37.7	4.55	1.09				
800	102.5	....	....	....	....	....	49.0	5.89	1.42				
900	115.3	....	....	....	....	....	62.3	7.6	1.80				
1,000	128.1	....	....	....	....	....	76.9	9.3	2.21				
1,500	192.3	....	....	....	....	....	....	21.0	4.9	0.57			
2,000	256.2	....	....	....	....	....	....	37.4	8.8	0.99	0.24		
2,500	316.4	....	....	....	....	....	....	58.4	13.8	1.57	0.37		
3,000	384.6	....	....	....	....	....	....	84.1	20.0	2.26	0.53		
3,500	447.8	....	....	....	....	....	....	....	27.2	3.04	0.70	0.22	
4,000	512.4	....	....	....	....	....	....	....	35.5	4.01	0.94	0.28	
4,500	576.5	....	....	....	....	....	....	....	45.0	5.10	1.19	0.36	
5,000	632.8	....	....	....	....	....	....	....	55.6	6.3	1.47	0.44	0.17
6,000	768.8	....	....	....	....	....	....	....	80.0	9.1	2.11	0.64	0.24
7,000	896.0	....	....	....	....	....	....	....	....	12.2	2.88	0.87	0.33
8,000	1,025	....	....	....	....	....	....	....	....	16.1	3.77	1.12	0.46
9,000	1,153	....	....	....	....	....	....	....	....	20.4	4.77	1.43	0.57
10,000	1,280	....	....	....	....	....	....	....	....	25.1	5.88	1.77	0.69
11,000	1,410	....	....	....	....	....	....	....	....	30.4	7.10	2.14	0.83
12,000	1,540	....	....	....	....	....	....	....	....	36.2	8.5	2.54	0.98
13,000	1,668	....	....	....	....	....	....	....	....	42.6	9.8	2.98	1.15
14,000	1,795	....	....	....	....	....	....	....	....	49.2	11.5	3.46	1.35
15,000	1,923	....	....	....	....	....	....	....	....	56.6	13.2	3.97	1.53
16,000	2,050	....	....	....	....	....	....	....	....	64.5	15.0	4.52	1.75
18,000	2,310	....	....	....	....	....	....	....	....	81.5	19.0	5.72	2.22
20,000	2,560	....	....	....	....	....	....	....	....	....	23.6	7.0	2.74
22,000	2,820	....	....	....	....	....	....	....	....	....	28.5	8.5	3.33
24,000	3,080	....	....	....	....	....	....	....	....	....	33.8	10.0	3.85
26,000	3,338	....	....	....	....	....	....	....	....	....	39.7	11.9	4.65
28,000	3,590	....	....	....	....	....	....	....	....	....	46.2	13.8	5.40
30,000	3,850	....	....	....	....	....	....	....	....	....	53.0	15.9	6.17

In psi in 1000 ft of pipe, 100 lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

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Table 4.8 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Equivalent Cu ft Compressed Air Per Min	Nominal Diameter, In.											
		1/2	3/4	1	1 1/4	1 1/2	2	3	4	6	8	10	12
10	1.05	5.35	0.82	0.23									
20	2.11	21.3	3.21	0.92	0.21								
30	3.16	48.0	7.42	2.07	0.47	0.21							
40	4.21	....	13.2	3.67	0.85	0.38							
50	5.26	....	20.6	5.72	1.33	0.59							
60	6.32	....	29.7	8.25	1.86	0.84	0.23						
70	7.38	....	40.5	11.2	2.61	1.15	0.31						
80	8.42	....	53.0	14.7	3.41	1.51	0.40						
90	9.47	....	68.0	18.6	4.30	1.91	0.51						
100	10.50	....	....	22.9	5.32	2.36	0.63						
125	13.15	....	....	39.9	8.4	3.70	0.98						
150	15.79	....	....	51.6	12.0	5.30	1.41	0.17					
175	18.41	....	....	....	16.3	7.2	1.95	0.24					
200	21.05	....	....	....	21.3	9.4	2.52	0.31					
250	26.30	....	....	....	33.2	14.7	3.94	0.48					
300	31.60	....	....	....	47.3	21.2	5.62	0.70					
350	36.80	....	....	....	....	28.8	7.7	0.94	0.22				
400	42.10	....	....	....	....	37.6	10.0	1.23	0.28				
450	47.30	....	....	....	....	47.7	12.7	1.55	0.37				
500	52.60	....	....	....	....	58.8	15.7	1.93	0.46				
600	63.20	....	....	....	....	....	22.6	2.76	0.65				
700	73.80	....	....	....	....	....	30.0	3.74	0.89				
800	84.20	....	....	....	....	....	40.2	4.85	1.17				
900	94.70	....	....	....	....	....	51.2	6.2	1.48				
1,000	105.1	....	....	....	....	....	63.2	7.7	1.82				
1,500	157.9	....	....	....	....	....	....	17.2	4.1	0.47			
2,000	210.5	....	....	....	....	....	....	30.7	7.3	0.82	0.19		
2,500	263.0	....	....	....	....	....	....	48.0	11.4	1.30	0.31		
3,000	316	....	....	....	....	....	....	69.2	16.4	1.86	0.43		
3,500	368	....	....	....	....	....	....	....	22.3	2.51	0.57	0.18	
4,000	421	....	....	....	....	....	....	....	29.2	3.30	0.77	0.23	
4,500	473	....	....	....	....	....	....	....	37.0	4.2	0.98	0.24	
5,000	526	....	....	....	....	....	....	....	45.7	5.2	1.21	0.36	
6,000	632	....	....	....	....	....	....	....	65.7	7.5	1.74	0.52	0.20
7,000	738	....	....	....	....	....	....	....	....	10.0	2.37	0.72	0.27
8,000	842	....	....	....	....	....	....	....	....	13.2	3.10	0.93	0.38
9,000	947	....	....	....	....	....	....	....	....	16.7	3.93	1.18	0.47
10,000	1,051	....	....	....	....	....	....	....	....	20.6	4.85	1.46	0.57
11,000	1,156	....	....	....	....	....	....	....	....	25.0	5.8	1.76	0.68
12,000	1,262	....	....	....	....	....	....	....	....	29.7	7.0	2.09	0.81
13,000	1,368	....	....	....	....	....	....	....	....	35.0	8.1	2.44	0.95
14,000	1,473	....	....	....	....	....	....	....	....	40.3	9.7	2.85	1.11
15,000	1,579	....	....	....	....	....	....	....	....	46.5	10.9	3.26	1.26
16,000	1,683	....	....	....	....	....	....	....	....	53.0	12.4	3.72	1.45
18,000	1,893	....	....	....	....	....	....	....	....	66.9	15.6	4.71	1.83
20,000	2,150	....	....	....	....	....	....	....	....	....	19.4	5.8	2.20
22,000	2,315	....	....	....	....	....	....	....	....	....	23.4	7.1	2.74
24,000	2,525	....	....	....	....	....	....	....	....	....	27.8	8.4	3.17
26,000	2,735	....	....	....	....	....	....	....	....	....	32.8	9.8	3.83
28,000	2,946	....	....	....	....	....	....	....	....	....	37.9	16.4	4.4
30,000	3,158	....	....	....	....	....	....	....	....	....	43.5	13.1	5.1

In psi in 1000 ft of pipe, 125 lb gage initial pressure. For longer or shorter lengths of pipe the friction loss is proportional to the length, i.e., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 4.9 Loss of Air Pressure Due to Friction

Cu ft Free Air Per Min	Nominal Diameter, In.											
	1/2	3/4	1	1 1/4	1 1/2	1 3/4	2	3	4	6	8	10	12
5	12.7	1.2	0.5										
10	50.7	7.8	2.2	0.5									
15	114.1	17.6	4.9	1.1									
20	202	30.4	8.7	2.0	0.9								
25	316	50.0	13.6	3.2	1.4	0.7							
30	456	70.4	19.6	4.5	2.0	1.1							
35	811	95.9	26.2	6.2	2.7	1.4							
40	....	125.3	34.8	8.1	3.6	1.9							
45	....	159	44.0	10.2	4.5	2.4	1.2						
50	....	196	54.4	12.6	5.6	2.9	1.4						
60	....	282	78.3	18.2	8.0	4.2	2.2						
70	....	385	106.6	24.7	10.9	5.7	2.9						
80	....	503	139.2	32.3	14.3	7.5	3.8						
90	....	646	176.2	40.9	18.1	9.5	4.8						
100	....	785	217.4	50.5	22.3	11.7	6.0						
110	....	950	263	61.2	27.0	14.1	7.2						
120	....	....	318	72.7	32.2	16.8	8.6						
130	....	....	369	85.3	37.8	19.7	10.1	1.2					
140	....	....	426	98.9	43.8	22.9	11.7	1.4					
150	....	....	490	113.6	50.3	26.3	13.4	1.6					
160	....	....	570	129.3	57.2	29.9	15.3	1.9					
170	....	....	628	145.8	64.6	33.7	17.6	2.1					
180	....	....	705	163.3	72.6	37.9	19.4	2.4					
190	....	....	785	177	80.7	42.2	21.5	2.6					
200	....	....	870	202	89.4	46.7	23.9	2.9					
220	....	....	....	244	108.2	56.5	28.9	3.5					
240	....	....	....	291	128.7	67.3	34.4	4.2					
260	....	....	....	341	151	79.0	40.3	4.9					
280	....	....	....	395	175	91.6	46.8	5.7					
300	....	....	....	454	201	105.1	53.7	6.6					
320	....	....	....	....	....	....	61.1	7.5					
340	....	....	....	....	....	....	69.0	8.4	2.0				
360	....	....	....	....	....	....	77.3	9.5	2.2				
380	....	....	....	....	....	....	86.1	10.5	2.5				
400	....	....	....	....	....	....	94.7	11.7	2.7				
420	....	....	....	....	....	....	105.2	12.9	3.1				
440	....	....	....	....	....	....	115.5	14.1	3.4				
460	....	....	....	....	....	....	125.6	15.4	3.7				
480	....	....	....	....	....	....	137.6	16.8	4.0				
500	....	....	....	....	....	....	150.0	18.3	4.3				
525	....	....	....	....	....	....	165.0	20.2	4.8				
550	....	....	....	....	....	....	181.5	22.1	5.2				
575	....	....	....	....	....	....	197	24.2	5.7				
600	....	....	....	....	....	....	215	26.3	6.2				
625	....	....	....	....	....	....	233	28.5	6.8				
650	....	....	....	....	....	....	253	30.9	7.3				
675	....	....	....	....	....	....	272	33.3	7.9				
700	....	....	....	....	....	....	294	35.8	8.5				
750	....	....	....	....	....	....	337	41.4	9.7				
800	....	....	....	....	....	....	382	46.7	11.1				
850	....	....	....	....	....	....	433	52.8	12.5				
900	....	....	....	....	....	....	468	59.1	14.0				
950	....	....	....	....	....	....	541	65.9	15.7				
1,000	....	....	....	....	....	....	600	73.0	17.3	1.9			
1,050	....	....	....	....	....	....	658	80.5	19.1	2.1			

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Table 4.9 Loss of Air Pressure Due to Friction (continued)

Cu ft Free Air Per Min	Nominal Diameter, In.												
	1/2	3/4	1	1 1/4	1 1/2	1 3/4	2	3	4	6	8	10	12
1,100	....	....	....	....	....	....	723	88.4	21.0	2.4			
1,200	....	....	....	....	....	....	850	105.2	25.0	2.8			
1,300	....	....	....	....	....	....	....	123.4	29.3	3.3			
1,400	....	....	....	....	....	....	....	....	33.9	3.8			
1,500	....	....	....	....	....	....	....	....	39.0	4.4			
1,600	....	....	....	....	....	....	....	....	44.3	5.1			
1,700	....	....	....	....	....	....	....	....	50.1	5.7			
1,800	....	....	....	....	....	....	....	....	56.1	6.4			
1,900	....	....	....	....	....	....	....	....	62.7	7.1	1.6		
2,000	....	....	....	....	....	....	....	....	69.3	7.8	1.8		
2,100	....	....	....	....	....	....	....	....	76.4	8.7	2.0		
2,200	....	....	....	....	....	....	....	....	83.6	9.5	2.2		
2,300	....	....	....	....	....	....	....	....	91.6	10.4	2.4		
2,400	....	....	....	....	....	....	....	....	99.8	11.3	2.6		
2,500	....	....	....	....	....	....	....	....	108.2	12.3	2.9		
2,600	....	....	....	....	....	....	....	....	117.2	13.3	3.1		
2,700	....	....	....	....	....	....	....	....	126	14.3	3.3		
2,800	....	....	....	....	....	....	....	....	136	15.4	3.6		
2,900	....	....	....	....	....	....	....	....	146	16.5	3.9		
3,000	....	....	....	....	....	....	....	....	156	17.7	4.1		
3,200	....	....	....	....	....	....	....	....	177	20.1	4.7		
3,400	....	....	....	....	....	....	....	....	200	22.7	5.3		
3,600	....	....	....	....	....	....	....	....	224	25.4	5.6	1.8	
3,800	....	....	....	....	....	....	....	....	250	28.4	6.6	2.0	
4,000	....	....	....	....	....	....	....	....	277	31.4	7.3	2.2	
4,200	....	....	....	....	....	....	....	....	305	34.6	8.1	2.4	
4,400	....	....	....	....	....	....	....	....	335	38.1	8.9	2.7	
4,600	....	....	....	....	....	....	....	....	366	41.5	9.7	2.9	
4,800	....	....	....	....	....	....	....	....	399	45.2	10.5	3.2	
5,000	....	....	....	....	....	....	....	....	433	49.1	11.5	3.4	
5,250	....	....	....	....	....	....	....	....	477	54.1	12.6	3.4	
5,500	....	....	....	....	....	....	....	....	524	59.4	13.9	4.2	1.6
5,750	....	....	....	....	....	....	....	....	....	64.9	15.2	4.6	1.8
6,000	....	....	....	....	....	....	....	....	....	70.7	16.5	5.0	1.9
6,500	....	....	....	....	....	....	....	....	....	82.9	19.8	5.9	2.3
7,000	....	....	....	....	....	....	....	....	....	96.2	22.5	6.8	2.6
7,500	....	....	....	....	....	....	....	....	....	110.5	25.8	7.8	3.0
8,000	....	....	....	....	....	....	....	....	....	125.7	29.4	8.8	3.6
9,000	....	....	....	....	....	....	....	....	....	159	37.2	10.2	4.4
10,000	....	....	....	....	....	....	....	....	....	196	45.9	13.8	5.4
11,000	....	....	....	....	....	....	....	....	....	237	55.5	16.7	6.5
12,000	....	....	....	....	....	....	....	....	....	282	66.1	19.8	7.7
13,000	....	....	....	....	....	....	....	....	....	332	77.5	23.3	9.0
14,000	....	....	....	....	....	....	....	....	....	387	89.9	27.0	10.5
15,000	....	....	....	....	....	....	....	....	....	442	103.2	31.0	12.0
16,000	....	....	....	....	....	....	....	....	....	503	117.7	35.3	13.7
18,000	....	....	....	....	....	....	....	....	....	636	148.7	44.6	17.4
20,000	....	....	....	....	....	....	....	....	....	....	184	55.0	21.4
22,000	....	....	....	....	....	....	....	....	....	....	222	66.9	26.0
24,000	....	....	....	....	....	....	....	....	....	....	264	79.3	30.1
26,000	....	....	....	....	....	....	....	....	....	....	310	93.3	36.3
28,000	....	....	....	....	....	....	....	....	....	....	360	108.0	42.1
30,000	....	....	....	....	....	....	....	....	....	....	413	123.9	48.2

*To determine the pressure drop in psi, the factor listed in the table for a given capacity and pipe diameter should be divided by the ratio of compression (from free air) at entrance of pipe, multiplied by the actual length of the pipe in feet, and divided by 1000.

Piping from the header to points of use should connect to the top or side of the header to avoid being filled with condensate for which drainage drop legs from the bottom of the header should be installed. Properly located and maintained compressed air dryers should prevent condensate in headers. Headers and piping also should have an ample number of tapped connections to allow evaluation of air pressure at points throughout the system.

Air tools generally are rated at 90 psig. They can operate at lower or higher pressure but at the expense of efficiency. Torque wrenches will vary in torque output depending on the air pressure at the tool affecting the quality of the work piece. Similarly, paint spray may be too sparse or too dense if the air pressure at the paint gun fluctuates significantly.

Ideally, the pressure throughout an air distribution system should remain in steady state. This is not possible due to the variations in air flow requirements, pressure losses in the system and the types of controls used. This will be discussed later under air system efficiency.

The air pressure at a point of use will be the air pressure at the compressor discharge less the pressure drop due to friction from the flow rate between these two points, and the pressure drop through equipment such as compressed air dryers and filters. In addition, the use of flexible hose and quick disconnect fittings between piping and the tool may cause a significant pressure drop which must be accounted for. A point of use requiring compressed air at 100 psig requires an air compressor rated at 110 psig or higher, depending on the distribution system, controlled storage capacity and the type of treatment used for the air quality required at the point of use. Remember that the higher the pressure at the air compressor, the higher the operational energy costs. It follows then, that when procuring new production equipment requiring compressed air power, it should be specified to have the lowest possible efficient operating pressure.

Consideration also should be given to the potential addition of equipment creating additional air demand on the system, which could result in a fall in the system pressure, particularly if there is marginal air compressor capacity.

Artificial Demand in a Compressed Air System

When a compressed air system operates at a pressure higher than required, not only is more energy consumed in compressing the air, but end uses consume more air and leakage rates also increase. This increase may be referred to as Artificial Demand.

A compressed air system should be operated with compressor and system controls set to achieve the lowest practical pressure.

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Intermittent Demand in a Compressed Air System

Compressed Air Systems are dynamic, meaning that, conditions of flow rate and pressures throughout the system are not static but constantly changing.

The steadiest conditions usually occur in process type applications, where the demand for compressed air is relatively constant and/or changes are gradual. This simplifies the necessary controls for air flow and pressure.

In many industrial plants, demand can vary widely as a variety of tools are used and as isolated demand events occur. Often, a demand event occurs at some considerable distance from the compressor(s) supplying the compressed air. This often is aggravated by an initial distribution, sized for a given flow rate and distance, having been extended due to plant expansion. The original distribution pipe size has not been increased but the length and the flow rate have been increased. Pressure drop throughout the extended distribution system can vary erratically. This is because the compressor controls are sensing discharge pressure, an increase being interpreted as a reduction in demand and a decrease being interpreted as an increase in demand.

In some cases, a specific piece of machinery is installed, requiring a relatively large amount of compressed air but only for a relatively short period of time. If the total demand is measured over an hour or a day, the average flow rate in cubic feet per minute is well within the capacity capability of the compressor(s). However, during the time when the demand event occurs, the flow rate may exceed the capacity of the compressor(s), dryer(s) and filter(s).

Air Receiver Capacity

Theoretically, if the distribution system volume (air receivers plus distribution piping) was large enough, the air compressors would see a constant discharge pressure and there would be no artificial demand. Obviously a grossly oversized system volume is not practical but it demonstrates that many problems can be eliminated with adequately sized and located air receivers and sufficiently large diameter distribution piping. The basic problem is in getting compressed air from Point A to Point B, at the required flow rate and pressure. The air pressure, the length of the distribution piping and the air velocity due to its diameter, will determine the pressure drop. Excessive length and too small a pipe diameter can create significant problems. Hysteresis also will cause delays in response time of controls, further aggravating the problems.

An adequately sized secondary receiver, located close to points of high and/or intermittent demand, can provide the required flow rate(s) without significant pressure drop in the system and the air compressor(s) has adequate time to replenish the pressure in the secondary air receiver. A restriction orifice at the inlet to a secondary air receiver can limit the rate of flow to replenish the receiver in the available time, without depleting air supply needed at other points in the system. In some cases, a check valve prior to the inlet to the receiver will ensure the availability of the required intermittent demand flow rate and pressure, as the secondary receiver will not then be supplying air to other demand events which may occur prior to the receiver. This is particularly so for situations of high intermittent demand.

The size of an air receiver can be calculated as follows:

$$V = T\, x\, \frac{C\, x\, P_a}{P_1 - P_2}$$

where:

V	=	Receiver volume, ft³
T	=	Time allowed (minutes) for pressure drop $P_1 - P_2$ to occur.
C	=	Air demand, cfm of free air
P_a	=	Absolute atmospheric pressure, psia
P_1	=	Initial receiver pressure, psig
P_2	=	Final receiver pressure, psig

The formula assumes the receiver volume to be at ambient temperature and that no air is being supplied to the air receiver by the compressor(s). If the compressor(s) is running while air is being drawn from the receiver, the formula should be modified so that C is replaced by $C - S$, where S is the compressor capacity, cfm of free air. The initial formula also can be used with a known receiver size, to determine the time to restore the air receiver pressure. In this case, C is replaced by S, which is the compressor capacity, cfm of free air.

In a compressed air distribution system, the Supply Side consists of the air compressors, air aftercoolers, dryers and associated filters, and a primary air receiver. The primary air receiver provides storage volume of compressed air and tends to isolate the compressors from the dynamics of the system. The compressor capacity controls respond to the pressure seen at the discharge connection of the compressor package. Multiple compressors should be discharging into a common header, which will be at slightly higher pressure than the primary air receiver pressure, due to pressure drop through the dryers and filters (presuming the dryers and filters are between the compressors and the primary air receiver).

In the past, mainly with reciprocating compressors, rules of thumb for sizing a primary air receiver have been from 1 gallon per cfm to 3 gallons per cfm of compressor capacity. This is no longer regarded as good practice and the recommended primary receiver size will vary with the type of compressor(s), its type of capacity control, anticipated or actual supply side and demand side events.

Many oil injected rotary screw compressors are equipped with capacity control by inlet valve modulation designed to match the output from the air compressor with the demand from the points of use. On this basis, it has been stated that an air receiver is not needed. At best, this is misleading. An air receiver near the discharge of a rotary screw compressor will shield the compressor control system from pressure fluctuations from the demand side, downstream of the receiver and can allow the compressor to be unloaded for a longer period of time, during periods of light demand. The addition of an over-run timer can stop the compressor if it runs unloaded for a pre-set time, saving additional energy.

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Some oil injected rotary screw compressors are sold with load/unload capacity control, which is claimed to be the most efficient. This also can be misleading, since an adequate receiver volume is essential to obtain any real savings in energy.

At the moment an oil injected rotary screw compressor unloads, the discharge pressure of the air end is that in the sump/separator vessel, the Unload Pressure Set Point. At the same time, the inlet valve closes but that is as inlet valve modulation to zero capacity. At this point, the power is approximately 70% of full load power. It is not until the pressure in the sump separator vessel has been bled down to the fully unloaded pressure that the fully unloaded power of approximately 25% is reached.

Rapid bleed down of pressure would cause excessive foaming of the oil and increased oil carry-over downstream. To prevent this, from 30 – 100 seconds bleed-down time is normal and, in some cases, the compressor will re-load before the fully unloaded condition is reached. For this reason, adequate primary storage volume is essential to benefit from Load/Unload Capacity Control.

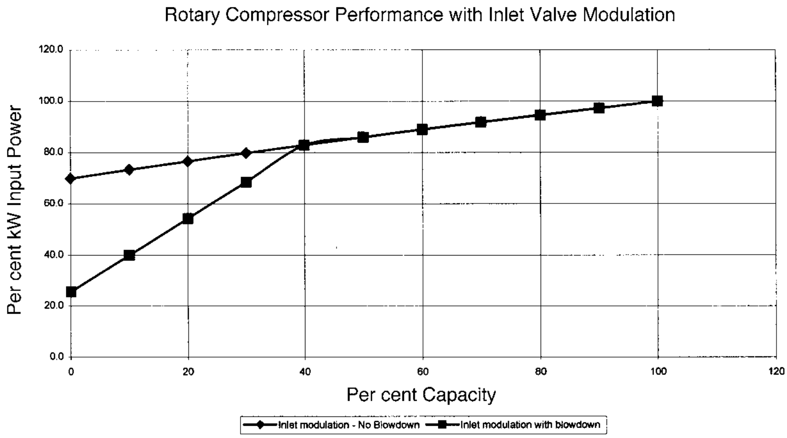

Figure 4.1A Shows Inlet Valve Modulation from 100% to 40% capacity and unloading at that point.

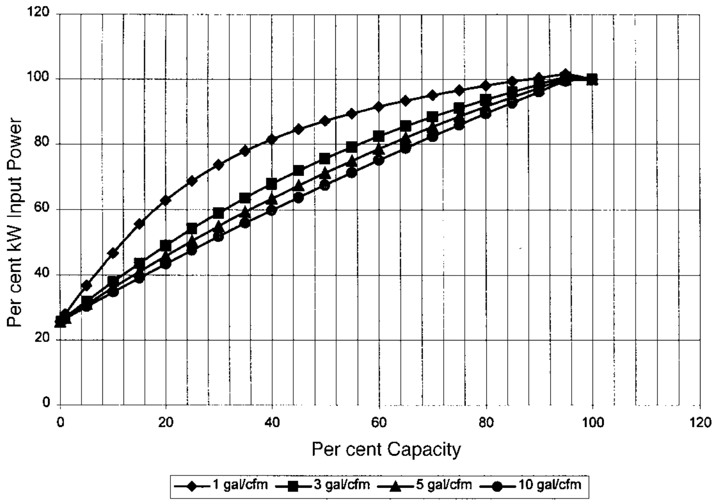

Figure 4.1B Shows Average Power *v* Percent Capacity with various sizes of primary receiver.

One solution, sometimes proposed, is to eliminate modulation and have the compressors operate in a load/unload mode. Certain factors must be recognized before making such a change. The standard full capacity, full load pressure, often has the compressor running at around 110% of motor nameplate rating, or using 10% of the available 15% continuous overload service factor. The remaining 5% is meant to cover tolerances and items such as increased pressure drop through the air/oil separator before it is required to be changed.

If the discharge pressure is allowed to rise by an additional 10 psi without the capacity being reduced by inlet valve modulation, the bhp will increase by 5% and the motor could be overloaded. A reduction in discharge pressure may be necessary to operate in this mode.

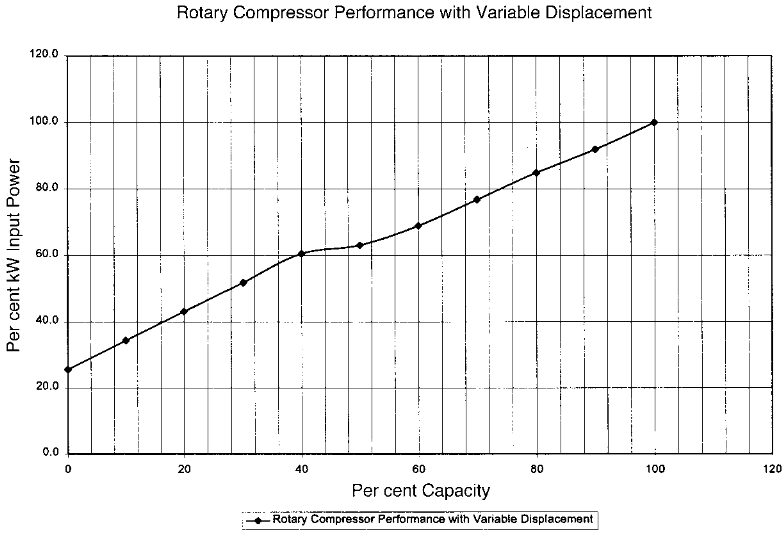

Figure 4.1C Shows Average Power *v* Percent Capacity with Variable Displacement Capacity Control (Slide/Spiral/Turn Valve) from 100% to 50% capacity followed by Inlet Valve Modulation to 40% capacity, then unloading. With this type of control, the inlet pressure to the air end does not change, hence the pressure ratio remains essentially constant. The effective length of the rotors is reduced.

In each of the above types of capacity control, the compressors are essentially the same, running at constant speed. Only the method of control changes.

Variable Speed may be achieved by variable frequency AC drive, or by switched reluctance DC drive. Each of these has its specific electrical characteristics, including inverter and other losses.

Air end displacement is directly proportional to rotor speed but air end efficiency depends upon male rotor tip speed. Most variable speed drive (VSD) package designs involve full capacity operation above the optimum rotor tip speed, at reduced air end efficiency and increased input power, when compared with a constant speed compressor of the same capacity, operating at or near its optimum rotor tip speed. While energy savings can be realized at all reduced capacities, the best energy savings are realized in applications where the running hours are long, with a high proportion in the mid to low capacity range.

Some designs stop the compressor when a lower speed of around 20% is reached, while others may unload at 40-50%, with an unloaded power of 10-15%. The appropriate amount of storage volume should be considered for each of these scenarios.

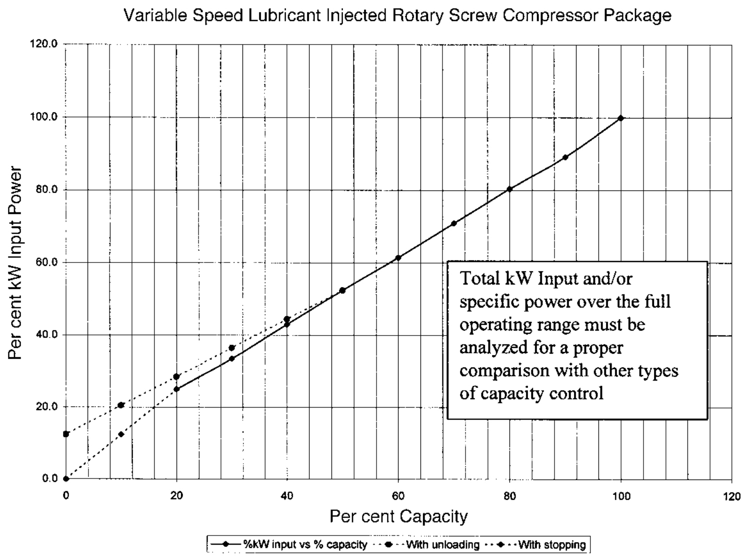

Figure 4.1D Shows Average Power *v* Percent Capacity with this type of control.

The control mode chosen should take into account the receiver/system volume relative to compressor capacity, the range of flow rate normally experienced, and the mean flow rate during a 24 hour period.

It should be noted that in systems with multiple compressors and sequencing controls, it is possible to have most of the compressors running fully loaded on base load with only one compressor on "trim" or part load, providing the most efficient mode for the system. It also is not necessary to have the air receiver/system storage capacity based upon the total capacity of all the compressors, provided they are not all on the same load and unload pressure settings. In such cases only the capacity of the "trim" compressor needs to be considered, provided it is the same as, or close to, the capacity of a compressor that may be fully unloaded or stopped while the "trim" compressor continues to operate.

A primary air receiver allows the compressor(s) to operate in a given discharge pressure range (usually 10 psi) from load to unload. Multiple compressors also can be sequenced as needed and with all but one operating in the most efficient, fully loaded mode. The capacity of the one compressor is modulated to match system demand.

Another option to minimize the effects of artificial demand, is the use of a Pressure/Flow Controller. This normally is located downstream of the primary air receiver and is a sophisticated form of pressure regulator. It is designed to allow flow at the required rate of demand, to maintain a stable downstream pressure, often within +/-1 psi. The stable downstream pressure can be set at the lowest practicable level for satisfactory operation of the pneumatic equipment, reducing the rate of any leakage from the system and allowing improved quality control from pneumatic processes, tools and devices. While this may reduce the flow rate experienced, the compressors may still operate at a higher than required discharge pressure. The primary receiver then will provide a certain amount of storage volume but compressor controls also must be addressed to reduce the compressor discharge pressure, if optimum energy use is to be achieved.

Selecting the Air Compressor Type(s) and Number

Air compressors vary in design characteristics and, although there is some overlap, each has its optimum range of capacity and/or pressure. The design and operational characteristics of each type is discussed in Chapter 2.

Generally, air cooled reciprocating compressors are best suited to a capacity requirement of 40 acfm (approx. 10 hp) or less, although sizes up through 150 hp are available. Standard pressure ratings of 100 psig and 175 psig are common. The 175 psig rating is common in automotive repair facilities for tire changers, hoists, etc., but seldom required for typical industrial applications. Most have cylinder lubrication but lubricant free and lubricant less designs also are available. Smaller sizes generally run on a start/stop type of control requiring an air receiver storage tank with a significant pressure difference between the start and stop settings. Larger sizes may have continuous running with load/unload controlled by pressure settings. In some cases, a specific point of use may benefit from one of these compressors dedicated to it rather than drawing from the main distribution system.

Double-acting reciprocating air compressors are efficient in operation but require relatively large installation space and foundations. These once were the work horses of plant air systems but have been largely displaced by less costly packaged rotary air compressors.

Rotary air compressors are available up through 3,000 acfm with pressures up to 200 psig although most operate around 100-125 psig. These are available both lubricant injected and lubricant free and have a variety of control types available as described in the chapter on rotary air compressors.

Centrifugal air compressors are best suited to relatively high volume, base load conditions and are considered more economical above 1,500 acfm (approx. 300 hp). Although pressures up to 10,000 psig are possible, most industrial centrifugal air compressors operate in the 100-125 psig range. Capacity control may be by inlet throttling, inlet guide vanes and/or discharge bypass. These are described in the chapter on dynamic air compressors.

Depending on the total system requirements, more than one type of compressor may be the best choice. For example, a large volume automotive plant may benefit from centrifugal compressor(s) capable of handling the base load demand and rotary or reciprocating air compressor(s) to function as trim compressor(s) for fluctuating loads. Standby air compressors also would be required to allow for maintenance and any unscheduled down time. A plant shutdown can be much more costly than an additional air compressor. Consideration also should be given to potential plant expansion.

Some plants operate only one shift per day so air demand does not fluctuate as much as in a plant with three shifts of operation and only one shift at full production rate. In plants having three shifts with widely differing requirements, the base load compressor should be capable of handling the demands of the least loaded shift and an additional compressor or compressors running only for the other shift(s). Maintenance needs also must be taken into consideration and a number of identical air compressors can minimize replacement parts considerations.

A centralized air compressor room can facilitate installation and maintenance considerations and minimize the number of standby compressors required and, if required, the number of operators. On the other hand, the distance from the compressor room to the furthest point of use must be considered as extensive lengths of piping cause increased pressure drop, potential leaks and, when run outside, potential line freezing problems. The centralized room also must allow room for maintenance and for future plant expansion and additional air requirements. Depending on compressor type(s), a centralized compressor room keeps the need for noise attenuation at one location, away from the work place.

Comparison of standard air compressor performance generally is made at a discharge pressure of 100 psig. The nameplate hp rating may not correspond with the total package kW required by the compressor so it is important to have the actual kW at the specified operating conditions of air flow rate and pressure when comparing air compressor types and manufacturers. A useful comparison at 100 psig is the total package input kW per 100 acfm of free air delivered. These are at compressor full load operation. Consideration also must be given to efficiency at part load and no load operation. Power costs vary widely throughout the United States and compressor efficiency considerations will be more serious in some areas. For samples of the data sheets, visit the CAGI website: www.cagi.org.

Lubricant free air may be required for all or for some specific point of use applications. If only one point of use or a few points require lubricant free air, there are two ways of dealing with this. One is to draw air from the plant air distribution system and treat it immediately prior to the point of use with the appropriate filtration (see the chapter on Air Treatment). The other is to have a separate lubricant free compressor for the point(s) of use. In general, lubricant free compressors have a higher initial cost and higher maintenance costs than their lubricated or lubricant injected counterparts but have the advantage of condensate uncontaminated by lubricant.

A significantly higher (or lower) pressure may be required for a specific point or points of use. A higher pressure requirement generally would be better served by a dedicated higher pressure air compressor or by a booster compressor drawing from the main air distribution system and boosting it to the pressure required at the specific point(s) of use. This prevents the total air distribution from operating at a higher pressure and absorbing more energy. A lower pressure at a specific point of use with significant air demand may justify a separate air compressor. The alternative is drawing from the main air distribution system through a pressure regulator, which will maintain the required lower pressure at the point of use.

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The location of the compressors chosen must take into account the type of cooling required. The vast majority of air compressors under 100 hp and about 50% of air compressors 200 hp and larger are air cooled. This eliminates the need for cooling water and its drainage considerations, or cooling water systems with higher than ambient water temperatures and the possible need for water treatment and associated costs. However, large radiator type coolers located outdoors in Northern climates also can present problems of lubricant temperature and viscosity at start-up when the compressor is idle overnight. Heated air from radiator type coolers located indoors can be used for space heating in plants in winter months and vented outside when heated air is not required. Adequate ventilation in the compressor room also must be considered.

Another consideration, often overlooked, is the source of inlet air to the compressor. Drawing air from the compressor room may be using air which has been air conditioned and either cooled or heated at cost and a higher than outside ambient temperature also results in a reduced mass flow of air through the compressor. On the other hand, air drawn from outside should be from a location where contaminants such as industrial gases will not be a problem. It also should be remembered that the air intake filter on a standard air compressor package is designed to protect the air compressor and not necessarily the equipment downstream of the compressor.

Available electrical power to the compressor room also must be considered, including voltage and kW capability.

Air Quality

The applications at the points of use will determine the quality of the air required at each point. Considerations include the content of particulate matter, condensate and lubricant. In the chapter on Compressed Air Treatment, reference is made to Air Quality Classes for these contaminants as published in International Standard ISO 8573-1. The chapter also describes the equipment available to meet these classes, including various air dryer types and various filter types. The manufacturer of process machinery and other pneumatic devices should be consulted to determine the air quality required.

When the air intake filter for the compressor(s) is mounted remotely from the compressor(s), the inlet air piping from the air intake filter to the compressor inlet must be clean and, being at atmospheric pressure, may be of plastic material. It should be remembered that the air intake filter is for the protection of the air compressor and does not necessarily protect the compressed air distribution system or equipment installed downstream. Downstream filtration is recommended as discussed under Compressed Air Treatment.

The compressed air distribution system itself may contribute to the contaminant problem, particularly if standard steel piping and air receivers are used. Stainless steel or copper piping is essential to some processes but may be considered too expensive in most industrial plants. Galvanized piping is one alternative and internal epoxy coating of air receivers another. The introduction of plastics for compressed air distribution systems has potential risks. These pipes generally are pressure rated at around 80°F and their pressure capability falls rapidly as temperature is increased. Such piping located near the roof of a building may see relatively high temperatures from the surrounding atmosphere as well as from the compressed air it contains. The plastic material and the joint compounds also may not be compatible with the type of air compressor lubricant used, particularly some of the popular synthetic lubricants. Plastic piping also is much more vulnerable to damage from fork lift trucks, etc., when located at or near floor level. For these reasons, plastic piping is not recommended for compressed air systems.

Fick's Law

This issue relates in general to very low pressure dew points, typically minus 100°F and below. Any leaks in supply and distribution piping downstream of the dryer(s) will allow back diffusion of atmospheric water vapor to enter the compressed air line even if it is at full line pressure.

Dalton's Law states that the total pressure of a mixture of ideal gases is equal to the sum of the partial pressures of the constituent gases. The partial pressure is defined as the pressure that each gas would exert if it alone occupied the volume of the mixture at the mixture temperature.

Fick's law of diffusion applies here. It states that "The rate of diffusion in a given direction is proportional to the negative of the concentration gradient." The concentration gradient relates to the partial pressure of water vapor and is very high at low pressure dew points.

The gradient difference of thousands of ppm of water vapor is the driving force; rather like an osmosis effect across a membrane. There are certain materials recommended to prevent this effect at low dew points and any dew point below minus 100°F should be sampled with nickel, PTFE, or preferably stainless steel tubing. Materials such as PVC or rubber should be totally avoided in these cases.

Also, experience has shown that to purge the piping downstream from the dryer takes an extraordinary amount of dry air to purge out the residual moisture. In tests to obtain minus 94°F dew point, if you have, for example, 1ft.^{3} of piping downstream from the dryer, then you will need about 1,000,000 ft.^{3} of dry air to purge it down to that dew point. This could have considerable impact in an industrial compressed air system, if a very low pressure dew point is required. It is essential that the proper piping material be used and any leaks in piping or joints be rectified to prevent this problem and to save the waste of compressed air.

Air System Efficiency

Efficient operation of the compressed air system requires that pressure fluctuations be minimized. Air compressors have an on load and an off load pressure setting. The differential between these two settings should not be allowed to exceed

10% of the maximum pressure setting. Normally this will require adequate storage volume in the form of an air receiver.

Generally, air receivers are sized in gallons up through 200 gallons, whereas larger sizes are sized in cubic feet. There are different old rules of thumb for the minimum size of an air receiver. One, based upon gallons, is: 1 gal per cfm of compressor capacity.

A similar rule, based upon cubic feet, is: 1 ft^3 per 10 cfm of compressor capacity. (100 psig gives a compression ratio at sea level conditions of 114.7/14.7=7.8 which is close to the conversion factor of 7.481 gallons per cubic foot).

Smaller air compressors (usually mounted on a receiver/tank) and operating with start/stop capacity control, generally have a receiver size, in cubic feet, approximating the compressor free air capacity in cfm divided by three.

The time taken for the pressure in an air receiver to drop from one pressure to another is:

$$T = V \frac{P_1 - P_2}{CP_a}$$

where:

T	=	time, minutes
P_1	=	Initial receiver pressure, psig
P_2	=	final receiver pressure, psig
P_a	=	atmospheric pressure, psi abs.
C	=	air requirement, cubic feet per minute
V	=	Receiver volume, cubic feet

The equation assumes the receiver to be at a constant atmospheric temperature and that no air is supplied to the receiver during the time interval. If air is being supplied constantly to the receiver at S cubic feet per minute, then C should be replaced by (C - S).

It should be noted that a receiver sized on the latter basis, with the compressor running at full load, will allow a drop in pressure of 2 psi in 4.08 seconds if there is a demand 20% above the compressor capacity. The pressure will fall by 10 psi in 20.4 seconds under the same conditions.

It is essential that such transient loads be taken into account when sizing an air receiver. A receiver located near a point of intermittent high flow rate can help isolate the air compressor(s) and the remainder of the system from severe pressure fluctuations.

The various types of capacity control for each compressor type, and their benefits, are described in the chapter dealing with air compressors. For very small compressor systems, it is common for the air compressor to be mounted on the air receiver. This means that compressed air drying and filtration equipment must be downstream of the air receiver. The receiver may provide some cooling of the air and a means of some of the condensate being collected but a sudden demand for air exceeding the dryer capacity rating could result in air to the point of use at a higher dew point.

When the air receiver is located downstream of the compressed air dryer, it stores air already dried and, since the capacity rating of the dryer normally matches the capacity rating of the air compressor, the dryer is shielded by the air receiver from a sudden high demand for air, so that the point(s) of use always receive(s) properly dried air.

Additional air receiver capacity close to the point of use can be very beneficial where the point of use may require a large volume of air for a relatively short period of time. The rest of the system then is shielded from a potential sharp drop in pressure due to the sudden high demand.

The minimum receiver capacity may be calculated but experience and judgment also are important. It has been claimed that unlike reciprocating compressors, which only load/unload or step type capacity control, rotary compressors with modulation of capacity by inlet throttling or similar means, do not need a receiver. This generally is not true and the installation of an air receiver is a sound investment, minimizing fluctuations in compressor operating conditions.

The following example illustrates the calculation of receiver capacity for a certain application.

Example:

$$V = \frac{TCP_a}{P_1 - P_2}$$

$$= \frac{(3 \times 15)(90)(14.7)}{(60)(100-80)}$$

$$= 49.6 \text{ ft}^3$$

The next larger standard air receiver from Table 4.10 should be used. Air receivers should meet the Code for Unfired Pressure Vessels published by The American Society of Mechanical Engineers (ASME). This normally is a requirement of insurance companies. All federal, state and local codes and/or laws also must be satisfied. ASME coded vessels also have approved safety valves, pressure gauge, drain valve and hand holes for manhole covers. Vertical receivers also have a skirt base. The location of the air receiver is important and regular draining is essential. The potential of freezing must be considered and provision made for heated drains where necessary. Provision for connecting a portable compressor for emergency conditions is also recommended.

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Table 4.10 Data for Selection of Receiver

Diameter, in.	Length, ft	Actual Compressor Capacity*	Volume, ft^3
14	4	40	4 $^1/_2$
18	6	110	11
24	6	190	19
30	7	340	34
36	8	570	57
42	10	960	96
48	12	2115	151
54	14	3120	223
60	16	4400	314
66	18	6000	428

*Cubic feet of free air per minute at 40 to 125 psig for constant-speed regulation. For automatic start-and-stop service, the receivers are suitable only for capacities one-half of the actual compressor capacities listed here-to avoid starting too frequently.

Energy management systems, with adequate and strategically placed air receivers, intermediate controls and pressure regulation, have been developed to suit specific plant systems, maintaining a very narrow band of system pressures, minimizing run time of compressors and improving quality of work performed by pneumatic tools and equipment.

Machines can be equipped to detect an idle threshold and stop air flow to an idle machine. A simple solenoid valve, arranged to close when the machine is idle, can reduce air consumption significantly.

When an air distribution system is operated at a higher than required pressure, this requires approximately one percent additional power for each 2 psi of operating pressure. In addition, leakage increases as pressure increases. This can be seen from Table 4.11 which shows that a 1/4 inch diameter orifice with 100 psig on one side and atmospheric pressure on the other will have a flow rate of 104 acfm of free air. At 110 psig the flow rate increases to 113 acfm. The combined effect of an increase of 5% in power for the 10 psi pressure increase, combined with the 8.6% increase in leakage flow rate, results in an overall power increase of approximately 14%.

Table 4.11 Discharge of Air through an Orifice

Gage Pressure before Orifice, psi	Nominal Diameter, In.										
	1/64	1/32	1/16	1/8	1/4	3/8	1/2	5/8	3/4	7/8	1
	Discharge, Cu. ft. Free Air Per Min.										
1	.028	0.112	0.450	1.80	7.18	16.2	28.7	45.0	64.7	88.1	115
2	.040	0.158	0.633	2.53	10.1	22.8	40.5	63.3	91.2	124	162
3	.048	0.194	0.775	3.10	12.4	27.8	49.5	77.5	111	152	198
4	.056	0.223	0.892	3.56	14.3	32.1	57.0	89.2	128	175	228
5	.062	0.248	0.993	3.97	15.9	35.7	63.5	99.3	143	195	254
6	.068	0.272	1.09	4.34	17.4	39.1	69.5	109	156	213	278
7	.073	0.293	1.17	4.68	18.7	42.2	75.0	117	168	230	300
9	.083	0.331	1.32	5.30	21.1	47.7	84.7	132	191	260	339
12	.095	0.379	1.52	6.07	24.3	54.6	97.0	152	218	297	388
15	.105	0.420	1.68	6.72	26.9	60.5	108	168	242	329	430
20	.123	0.491	1.96	7.86	31.4	70.7	126	196	283	385	503
25	.140	0.562	2.25	8.98	35.9	80.9	144	225	323	440	575
30	.158	0.633	2.53	10.1	40.5	91.1	162	253	365	496	648
35	.176	0.703	2.81	11.3	45.0	101	180	281	405	551	720
40	.194	0.774	3.10	12.4	49.6	112	198	310	446	607	793
45	.211	0.845	3.38	13.5	54.1	122	216	338	487	662	865
50	.229	0.916	3.66	14.7	58.6	132	235	366	528	718	938
60	.264	1.06	4.23	16.9	67.6	152	271	423	609	828	1,082
70	.300	1.20	4.79	19.2	76.7	173	307	479	690	939	1,227
80	.335	1.34	5.36	21.4	85.7	193	343	536	771	1,050	1,371
90	.370	1.48	5.92	23.7	94.8	213	379	592	853	1,161	1,516
100	.406	1.62	6.49	26.0	104	234	415	649	934	1,272	1,661
110	.441	1.76	7.05	28.2	113	254	452	705	1,016	1,383	1,806
120	.476	1.91	7.62	30.5	122	274	488	762	1,097	1,494	1,951
125	.494	1.98	7.90	31.6	126	284	506	790	1,138	1,549	2,023

Based on 100% coefficient of flow. For well-rounded entrance multiply values by 0.97. For sharp-edged orifices a multiplier of 0.65 may be used.

This table will give approximate results only. For accurate measurements see ASME Power Test Code, Velocity Volume Flow Measurement.

Values for pressures from 1 to 15 psig calculated by standard adiabatic formula.

Values for pressures above 15 psig calculated by approximate formula proposed by S. A. Moss: $w = 0.5303\, aCP_1 / \sqrt{T_1}$ where w = discharge in lb per sec, a = area of orifice in sq. in., C = coefficient of flow, P_1 = upstream total pressure in psia, and T_1 = upstream temperature in deg F abs.

Values used in calculating above table were C = 1.0, P_1 = gage pressure + 14.7 psi, T_1 = 530 F abs.

Weights (w) were converted to volumes using density factor of 0.07494 lb. per cu. ft. This is correct for dry air at 14.7 psia and 70°F.

Formula cannot be used where P_1 is less than two times the barometric pressure.

Compressed Air & Gas Institute • 1300 Sumner Avenue • Cleveland, OH 44115
Phone: 216/241-7333 • **Fax:** 216/241-0105 • **E-mail:** cagi@cagi.org

Air System Safety

Safety in the workplace is a primary design consideration. In a compressed air distribution system there are several factors involved. We will consider here only the distribution system and not the air compressors that have their own built-in safety systems.

The pressure rating of all piping must meet or exceed the maximum pressure to which the system may be subjected. The pressure rating should take into account the maximum temperature to which the piping will be exposed. Any exposed piping at an elevated temperature and which may be contacted by a person, should be shielded from contact or a suitable warning displayed. Piping must be adequately supported and allow for thermal expansion. Shut-off valves should be installed to allow maintenance of the various pieces of equipment but a pressure relief valve must be installed between the compressor and a shut-off valve to prevent over pressurization when the valve is closed.

Any air receiver should meet ASME Code for Unfired Pressure Vessels and be complete with a pressure relief valve and a pressure gauge. Applicable federal, state and local codes also must be met.

Pressure containing piping should be located away from passageways where fork lift trucks and other vehicles could come into contact with it, but should be accessible for maintenance.

Air Distribution System Layout

The foregoing gives recommendations for sizing the required air compressors, air receivers, headers and piping systems for minimum pressure drops and optimum efficiencies. Valves and fittings offering the least resistance to flow should be selected, including long radius elbows.

Where possible, a header loop system around the work place is recommended. This gives a two-way distribution to the point where demand is greatest. The header size should allow the desired minimum pressure drop regardless of the direction of flow around the loop. The header should have a slight slope to allow drainage of condensate and drop legs from the bottom side of the header should be provided to allow collection and drainage of the condensate. The direction of the slope should be away from the compressor.

Piping from the header to each point of use should be kept as short as possible. As previously stated, air piping from the header to the point of use should be taken from the top of the header to prevent the inclusion of condensate. These pipes normally run vertically downward from an overhead header to the point of use and pressure drop from the header to the point of use should not exceed 1 psi during the duty cycle.

For a lubricant free compressor, it is recommended that the air distribution header and piping be corrosion resistant.

A primary air receiver should be located close to the compressor(s) to shield the compressor(s) from fluctuations in air demand, maintaining a more stable compressor discharge pressure. Consideration should be given to secondary receiver(s) close to point(s) of use where relatively large surges in demand may occur. This also helps to stabilize the air distribution system.

Isolating valves to facilitate maintenance of system components, and tapped connections for system measurement and analysis, should be provided throughout the system.

Cost of Compressed Air

Compressor types, accessories, systems and operating pressures vary widely, which makes cost of operation difficult to establish. However, there are some good guidelines that can be used. The most common condition for comparison is a pressure at the discharge of the air compressor of 100 psig. This also is the standard inlet pressure used for rating of compressed air dryers and will be used in the following cost estimates:

Air Compressors

The most common air compressor in industrial use today is the lubricant injected type rotary screw compressor. These are available in single or two-stage versions, the two-stage version being more efficient than the single-stage. Generally a single-acting air cooled reciprocating compressor is the least efficient while a multi-stage, double acting, water cooled reciprocating compressor is the most efficient. Dynamic compressors efficiencies are closer to the multi-stage water cooled reciprocating compressors.

A common means of comparison is expressed in bhp/100 cfm with a compressor discharge pressure of 100 psig. The compressor capacity in cfm, or acfm, is the amount of air delivered from the compressor but measured at prevailing ambient inlet conditions. While the bhp/100 cfm can range from about 18 for an efficient two-stage water cooled reciprocating compressor to 30 for a small air cooled reciprocating compressor, a typical single-stage lubricant injected rotary screw compressor has a power consumption of approximately 22.2 bhp/100 cfm.

If we then allow a drive motor full load efficiency of 92% and 1 bhp = 0.746 kW, the resultant power consumption in electrical terms is 18 kW/100 cfm. This is what will be used in these discussions. CAGI Performance Data Sheets require total package input.

It should be noted that this is based upon 100 psig at the discharge of the air compressor and not at the point of use. Pressure drops will occur through piping and components downstream of the air compressor, resulting in a lower pressure at the point of use. If it is necessary to raise the discharge pressure of the compressor to achieve a required pressure at point of use, it is necessary to make sure that the air compressor is capable of the higher discharge pressure. It also should be noted that for each 2 psi increase in compressor discharge pressure, the power consumption will increase by 1%.

In addition, air compressors normally do not operate continuously at full capacity. While the average amount of air used may be less than the capacity of the air compressor, the reduction in power requirement is not directly proportional to the actual capacity. Rotary air compressors utilizing inlet valve throttling do not provide a significant reduction in kW as the capacity is reduced. Although the mass flow of air is reduced in direct proportion to the absolute inlet pressure at the inlet to the air end, the reduction in inlet absolute pressure results in a corresponding increase in the ratios of compression, adversely affecting kW/100 cfm. Other types of capacity control are available but all result in a higher kW/100 cfm at reduced capacity conditions.

Compressed Air Dryers

Like air compressors, dryer ratings are based upon a quantity of air measured at ambient conditions but with a dryer inlet condition of 100 psig and 100°F and with an ambient temperature of 100°F. Different dryer types have different requirements but each will have a pressure drop as the compressed air passes through it. This should be taken into account with the required increase in discharge pressure from the air compressor and its additional power requirement as indicated above.

Refrigerant Type Dryers
Typical rated Dew Point +35 to +40°F

Based upon Manufacturers' Data Sheets giving total power input in amps at a specified voltage, a reasonable rule of thumb is a power requirement of 0.5-0.8 kW/100 cfm of the rated capacity of the dryer. The dryer rating, rather than the compressor rating should be used in determining the power requirement of the refrigerant type dryer. Smaller capacity refrigerant type dryers may require more power/100 cfm and large dryers less but the above approximation is good from 500 through 1,000 cfm, not including energy to cover pressure drop through the dryer, which normally is approximately 3-5 psi, an additional 0.27- 0.45 kW/100 cfm.

Regenerative Desiccant Type Dryers
Typical rated Dew Point -40°F

The amount of purge air required to regenerate the desiccant bed may vary, based upon the type of controls used. Standard heatless dryers of this type require approximately 15% of the rated dryer capacity for purge air. Therefore, the electrical power requirement of this dryer will be 15% of the air compressor electrical power requirement as the amount of air delivered from the dryer to the system will be reduced by 15%. Based upon the above, this represents 2.7 kW/100 cfm. Special controls are available for most dryers of this type to economize the amount of purge air required that can be reduced to 10% or less, which represents 1.8 kW/100 cfm. Again, an energy allowance for pressure drop of 3-5 psi must be made, an additional 0.27- 0.45 kW/100 cfm.

Internal or external heating can be used to reduce the amount of purge air required. Purge air can be supplied by a blower, rather than compressed air. Vacuum purge also can be accomplished. These may provide overall energy savings.

Heat of Compression Type Dryers
Typical rated Dew Point is variable, depending on ambient and/or cooling water temperatures.

These may be either twin tower regenerative type or single rotating drum type. Technically, there is no reduction of air capacity with this type of dryer since hot unsaturated air from the compressor discharge is used for regeneration, then cooled and some of the moisture removed as condensate before passing through the drying section to be dried. However, for this to occur, an inefficient entrainment type nozzle has to be used and an electric motor also is used to rotate the dryer drum. Considering pressure drop and compressor operating cost, it is estimated that the total power requirement is approximately 0.5 - 0.8 kW/100 cfm.

Deliquescent Type Dryers
Typical Dew Point Suppression 15°F

Since the drying medium is consumed and not regenerated, there is no requirement for purge air, therefore, pressure drop through the dryer (and any associated filtration) and loss of air volume during the drain cycle are the operating costs which, excluding the replacement cost of the drying medium, are estimated at 0.2 kW/100 cfm.

Membrane Type Dryers
Typical Dew Point +40°F

This type of dryer requires an amount of purge air, or sweep air, to displace the moisture that passes through the permeable membranes. For comparable dew point depression with a refrigerant type dryer, this will be in the range of 15 to 20% of the air capacity of the dryer. Therefore, 15 to 20% should be added to the air compressor power requirement. Based upon the above, this amounts to 2.7 - 3.6 kW/100 cfm. Energy for a pressure drop of 3-5 psi will add 0.27 - 0.45 kW/100 cfm.

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