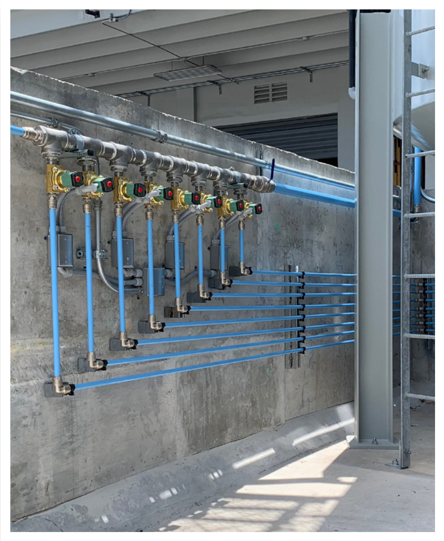
Compressed Air System Design

Chapter 4



CAGI
Compressed Air & Gas Institute

Chapter 4




Compressed Air System Design

Efficient Compressed Air Systems

When a compressed air system is properly designed, installed, operated and maintained, it is a major source of efficient industrial power, possessing many inherent advantages. Compressed air is safe, economical, adaptable, easily transmitted, and provides labor saving power. Compressed air is often referred to as being the fourth utility within industry, behind electricity, water, and natural gas. The cost of a complete compressed air system and pneumatic tools is relatively small in comparison with the utility provided by their use.

Goal of an Efficient Compressed Air System

The primary goal of a compressed air system is to deliver a reliable supply of clean, dry, compressed air at a stable pressure to every end user within the compressed air system, at the lowest cost possible. Many factors must be considered when designing a compressed air system to ensure its efficiency, reliability, and safety.

This chapter will focus on the roles that the following system parameters play in achieving the ultimate goal of a well-designed compressed air system:

- Air flow
- Air pressure
- Air quality
- Type and number of compressors
- Air distribution and system layout
- System efficiency

Air Flow Requirements

Determining the proper compressor capacity (supply) to install in order to satisfy the system compressed air usage (demand) is a vital and basic fundamental that is often misunderstood. It is very important to understand that the flow requirements of all pneumatic equipment are stated in free-air volumes: the amount of atmospheric air, compressed to a desired pressure, required to operate the tool as designed. Air compressor capacity rating is also stated in free-air volume. Determining the total demand of a compressed air system can be a complicated and oftentimes confusing task, especially in large systems with many end users. There are typically two methods for determining the total demand of a compressed air system; calculating or measuring. Both methods will be discussed below.

Calculating Air Flow Requirements

Understanding your constituents of demand is the first step to properly sizing your system. Whether you are increasing the demand of a current facility or beginning with a new system, calculating the demand of your compressed air system can be difficult due to the fluctuating demand of each air-consuming application. Nonetheless, understanding demand begins with summing the average air consumption of each user.

A study of pneumatic tools and devices in a typical manufacturing plant will show that some of these tools and devices operate almost constantly and others operate infrequently, yet 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 tools and devices will vary considerably in different applications. Accordingly, the demand requirement of a compressed air system should not be calculated as the total of the individual maximum consumptions of all pneumatic devices. This would greatly overstate the system demand and result in grossly oversizing the compressor supply. Instead, the demand requirement of a compressed air system should be calculated as the sum of the average air consumption of each device.

Determining the average air consumption of any pneumatic tool or device requires the use of the concept known within the industry as load factor. The cfm consumption for most pneumatic tools and devices is based on air pressure set at 90 psig while the tool/device is running and the tool is operating at 100% of its rated speed; revolutions per minute (rpm) or beats per minute (bpm) for percussive tools. Pneumatic power tools generally are operated only intermittently and often are operated at less than full rated power (speed). The ratio of actual air consumption over a given period of time to the maximum, continuous full-rated-power air consumption, each measured in cubic feet per minute of free air, is known as the load factor of the device. Load factor x full-power-rated cfm = average cfm for the device.

Two parameters affect the load factor. The first is the time factor, which is the percentage of time that the device is actually operated over the course of a work shift. The second is the work factor. Maximum work is done when the tool is operated under a load at its maximum rated speed; full load condition. Depending upon how much work the device is performing, air consumption could be anywhere between the following two conditions; 100% at full load or 0% at stalled load. Work factor is the actual percentage of full load consumption that is required to perform the actual required work. 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 more force that is applied to the work piece, the greater the work, the slower the speed of the tool, and the lower the cfm consumption. 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, and operating at its full-rated power/speed.

The load factor is the product of the time factor and the work factor. Multiplying the cfm rating of the tool or device by its load factor results in the average cfm demand of the tool or device. In one plant, the air actually consumed by 434 portable pneumatic tools on production work was only 15% of the summed, maximum rated capacities of all tools. To assure maximum accuracy when calculating system demand, it is essential that the most accurate determination or estimate of load factor be used in calculating the average consumption of each air-using device in the facility. Aver-

age air consumption calculations should be based upon the maximum continuous full-rated-power air consumption as stated by the manufacturer. Average air calculations should not be based upon the maximum free-speed consumption of the device, as this condition greatly increases the cfm consumption of the device.

When designing an entirely new compressed air 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. Care should be taken when calculating load factor to assure accuracy. Guesses and rules-of-thumb should be avoided. These methods can cause gross error in determining the proper amount of supply to install. For instance, one rule-of-thumb states that the compressor capacity should be about one third of the 100% load factor requirement of all the pneumatic tools. This rule-of-thumb leaves too much to chance.

Table 4.1, shows the air requirements at 100% load factor of various tools and this information can be used for preliminary system demand estimates. These figures are approximate and individual tools from different manufacturers may vary by more than 10% from the figures given. Contact the manufacturer of the air device for the most accurate consumption information. Table 4.2 shows the cubic feet per minute of air required for various sizes of sandblast nozzles.

Many pieces of production equipment are actuated by pneumatic cylinders. These include automatic feed devices, chucks, vises, clamps, presses, intermittent reciprocating and rotary motion devices, door openers and many other devices. Such devices usually have low air consumption and are themselves inexpensive. Accordingly, their usage in automated production processes is significant and increasing. Air consumption for such cylinders is shown in Table 4.3. This table shows the theoretical volume swept out by the piston during one full stroke. For demand calculation purposes, this volume at pressure 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 actual use, the air pressure to the cylinder might be regulated to a pressure considerably lower than the system line pressure. In such a case, the regulated article pressure should be used rather than full line pressure in converting the tabled figures to free air conditions. Similarly, 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.

Table 4.4 demonstrates how to calculate the average demand of a compressed air system by applying the appropriate load factors to the various air consuming devices in an industrial facility. Once you have determined the average demand of the system by calculating and summing the average demands of all air-using applications, you have a baseline of average demand with which to work. While it is usually safe to use an accurately calculated average system consumption for determining the required amount of compressors to be installed, it is important to realize that because many applications are intermittent in nature, the possibility exists for all users to consume at the same time, which would be a momentary demand significantly greater than the average consumption. This maximum demand can be addressed with compressor horsepower or, more efficiently, with the proper amount of storage within the system. The greater the total system storage, the closer to the average calculated demand you can size the supply equipment.

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 ¼-4 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

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	0.00139	0.00278	0.00416	0.00555	0.00694	0.00832	0.00972	0.0111	0.0125	0.0139	0.0153	0.01665
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

* These volumes are for single-acting cylinders. For double-acting cylinders, multiply by 2 and subtract the volume of the piston rod. These volumes are the theoretical volume displaced by the piston during one full stroke. This volume must be converted to a flow rate of free air based upon the operating pressure of the cylinder.

Table 4.4 How to Calculate Average System Demand

				cfm required*		
		Number of Tools	Load Factor* (% of time tools actually operated)	Per Tool When Operating	Total cfm if All Tools Operated Simultaneously	Total cfm Actually Used (A x B x C ÷100)
Type of Tool	**Location**	**(A)**	**(B)**	**(C)**	**(D)**	**(E)**
Blowguns, chucks, 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	20	30	30	6
Screwdriver	Shipping	1	20	35	35	7
Hoist	Shipping	1	20	40	40	8
Total		**51**			**1375**	**528**

*cfm is cubic feet of free air per minute.
*Load factor assumes 100% work factor.
Note: Total of column (E) is the average demand and determines required compressor sizes.

Measurement for Determining Flow Requirements

Another method for determining the demand of a compressed air system is to conduct a supply side assessment of the existing system. By logging compressor power (amps or kW) and discharge pressure or by using a calibrated flow meter device, a trained system auditor can accurately determine the actual demand of the existing system over the course of a week of operation. This data will show average demand, maximum and minimum demand, as well as average, maximum, and minimum pressures. Once this baseline information has been established for the current system, it is possible to determine the additional required supply horsepower that would be needed to satisfy the added demand load of a new expansion or the addition of new, air-consuming equipment. The additional air demand can be calculated by using the load factor method outlined above or by consulting with the manufacturer of the new equipment.

In the case of determining the demand of a new facility, it is a wise practice to perform a system assessment on a similar operation with similar equipment and applications to determine the expected demand of the new facility. For example, a major bakery was building a new bakery in North Carolina. It was to be a duplicate of an existing bakery in Texas. By performing a system assessment on the operating Texas bakery, the corporation knew the exact demand profile that was needed at the new bakery. This greatly simplified the compressed air equipment selection process and allowed the bakery to install an energy efficient and reliable compressed air system.

Measuring system demand increases accuracy by removing the judgmental component that is present when estimating the work factor for each pneumatic application and measuring has the added benefit of including all sources of demand; both appropriate and inappropriate. For example, leakage is included within the measured demand, but a leakage component must be added as a separate demand constituent when employing the calculation method to determine average system demand. A healthy air system should not leak more than 5 to 10% of its air supply, but it is not uncommon to see leaks in air systems exceeding 30%. Another sizeable demand constituent that must be accounted for when calculating average system demand is the wasted purge air from regenerative dryers. 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 calculated estimate of air required at points of use. Purge demand is automatically included in the total demand when the auditing-assessment method is employed to determine system demand.

Variability of Flow Requirements

Understanding the demand dynamics, or demand profile, of an air system is critical when it comes to choosing the right compressor for the job. The demand profile represents the flow requirements of a plant over a period of time. For example, the compressor may be running at 90% capacity during a first shift, at 50% capacity during the second shift, and at 20% capacity during the third shift. This represents an average demand profile of 53% of the rated capacity of the compressor. In satisfying this demand, the compressor is said to have a 53% load factor, which is the ratio of average compressor supply, over a given period of time, to the maximum rated




compressor capacity. The average flow required by end uses may be dramatically different than the maximum and minimum flows experienced in actual operation. There may be times when several demand events can occur simultaneously, resulting in a demand well above the average. There may be no demand during breaks or lunch.

It is important to identify those uses of compressed air that are relatively steady and consistent with regard to scfm and psig. It is more important to identify those uses that are intermittent as these uses determine the dynamics of the system. The average flow required over an hour can be substantially less than the peak flow over a few seconds or minutes. Sufficient storage of compressed air is essential to meet short-term high volume demands without having to rely on compressor horsepower. Secondary storage can be positioned close to the point of high intermittent usage to satisfy the peak demand event without exceeding the average flow in the main distribution piping. Without this dedicated storage, distribution piping and valves must be oversized to handle the peak flow requirement without excessive pressure drop. Air receivers at the point of use provide stored energy for intermittent users to minimize impact on the system pressure.

With the right type of air compressor and the application of air receivers in strategic locations, the compressed air system should be able to handle the varying demands for compressed air both efficiently and reliably. One type of compressor might best satisfy a plant that uses the maximum volume of air 100% of the time and a different type of compressor or compressors may better satisfy a dynamic demand that fluctuates significantly throughout the day.

The goal of an efficient compressed air system is to have the supply match the demand while maintaining a stable system pressure. Two factors are critical in achieving this goal: total system storage and compressor capacity control. Storage dampens the effect on pressure that sudden demand fluctuations create. The capacity control method of the compressor determines how quickly and how efficiently the compressor responds to the dynamic flow changes within the compressed air demand. The various capacity control methods used by compressors are covered in Chapter 2. The capacity control method chosen for the compressor depends on the total system storage volume (piping and receivers), the dynamic range of flow experienced, and the average flow rate during a 24-hour period. Because each compressed air system is unique, no one capacity control method is best for all compressed air systems. A system with steady demand and significant storage requires a different capacity control method than a system with a highly varying demand and minimal storage. When designing a new compressed air system, it is critical to calculate accurate flow requirements and dynamics as described in the previous paragraphs. This information will provide the ability to specify the proper amount of storage to maximize the efficiency of the compressor according to its capacity control method.

Air Pressure Requirements

The pressure at which to operate the compressed air system is one of the more critical factors in the design of an efficient compressed air system. The air pressure at any point of use will be the air pressure at the compressor discharge less the pressure drops created as the air flows through piping, filters, dryers, moisture separators, quick-disconnect fittings, and rubber hose to arrive at its final point of use. One major problem in system design is that the variety of points of use often require a variety of different operating pressures. Air tools generally are rated to operate at 90 psig. They can operate at lower or higher pressures, but at the expense of efficiency and productivity. Torque wrenches will vary in torque output depending upon the air pressure at the tool. This variance in torque can adversely affect the quality of the work being done. Similarly, paint spray might be too sparse or too dense if the air pressure at the paint gun fluctuates significantly.

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, a 2-psig change in system pressure requires a 1% change in the compressor power required to affect the change. For positive displacement compressors this is a direct relationship; i.e., a system pressure increase/decrease of 2 psig requires a 1% increase/decrease in operating compressor horsepower. Operating the complete system at twenty percent higher pressure to accommodate one point of use, would result in the air compressor(s) using 10% more energy and an increase in the artificial demand of all non-regulated users. This scenario, obviously, is to be avoided. The concept of artificial demand will be discussed in detail later in this chapter.

Pressure Drop

As air moves from the compressor discharge to its final point of use, it experiences a drop in pressure. Pressure drop is inherent with every compressed air system and it forces the compressor to generate pressure that is higher than the pressure required at the point of use. System pressure drop plays a major role in determining the proper selection of all compressed air system components. Understanding the causes of pressure drop, identifying the sources of pressure drop within a compressed air system, and identifying the negative effects of pressure drop on system performance are essential in designing an efficient compressed air system.

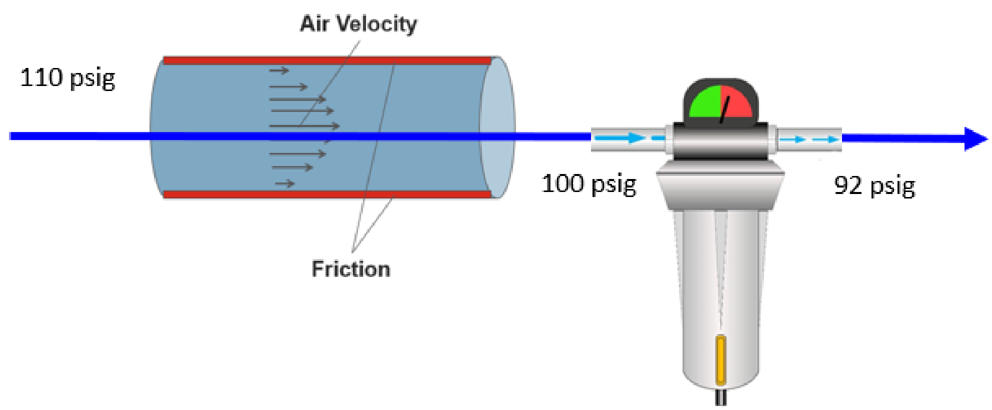


Figure 4.1 Pressure drop as a function of friction due to resistance to flow.

Pressure drop occurs due to friction between the compressed air and the internal surfaces of the components through which the compressed air must travel to get to its final user destination. The greater the friction, the greater the resistance to flow, the greater the pressure drop. Pressure drop is all about physics.

Figure 4.1 depicts air flowing through a red pipe and through a filter. In the figure a section of the red pipe preceding the filter has been enlarged for illustration. As the air moves through a pipe, the velocity of the air at the surface of the pipe approaches zero because there is friction between the air molecules and the inner surface of the pipe. This friction, or resistance to flow, results in a pressure drop. Pressure drop can be defined as the difference in total pressure between two points of a fluid-carrying network. In this case, there is a 10 psig pressure drop between the entrance of the red pipe and the discharge of the red pipe as a result of the friction within the pipe that resists the flow of compressed air through the pipe. Once the 100 psig air enters the filter, it encounters further resistance as the air is forced to go through a dense filter media, which creates a further resistance to airflow. Accordingly, there is an 8 psig pressure drop through the filter. It is important to note that pressure drop only exists when there is flow. Compressed air stored in a pipe and not flowing will have zero pressure drop, no matter how rough the internal surface of the pipe is.

Formula: $Psid_2 = Psid_1 \times [cfm_2/cfm_1]^2$

Figure 4.2 Pressure drop as a function of flow.

It might be tempting to attempt to overcome a pressure drop within a system, simply by adding another compressor to increase the flow of air through the system. This sounds logical, and it is often done in the real world, but adding supply to overcome a pressure drop is totally **wrong!** As shown in Figure 4.2, **pressure drop increases as the square of the flow change.** This formula states that when the flow increase is 2 times, the pressure drop increase is 2^2 times or 4 times. For example, a filter exhibits a 4 psig pressure drop at a flow of 350 cfm. Double the flow to 700 cfm and the same filter will exhibit an unacceptable 16 psig pressure drop. This relationship is determined by the velocity, or speed, at which air moves through a resistance, be it an air dryer or a length of distribution header piping.

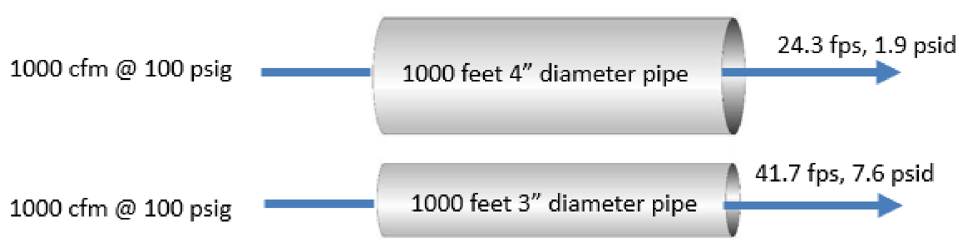


Figure 4.3 Pressure drop as a function of velocity.

While velocity and pressure vary inversely, velocity and **pressure drop** vary directly. For a given volume of compressed air, as its velocity increases the pressure that it exerts on the walls of the pipe reduces. So, the faster that air moves through a pipe, the greater its pressure drop across the length of pipe. This follows Bernoulli's principle that states; "as the velocity of a moving fluid (gas or liquid) increases, the pressure within the fluid decreases." Pipe diameter affects the velocity of air moving through the pipe. The smaller the diameter, the faster the velocity, for a given volume of air at a given initial pressure. This increase in velocity causes additional pressure drop.

As shown in Figure 4.3, 1000 cfm @ 100 psig moves through 4" diameter schedule 40 steel pipe at a rate of 24.3 feet per second. The same 1000 cfm @ 100 psig flows through 3" diameter schedule 40 steel pipe at a velocity of 41.7 feet per second. Following Bernoulli's principal, the faster moving air exhibits a greater pressure drop than does the more slowly moving air. The pressure drop of the 3" pipe is almost 4 times that of the 4" pipe. This is why proper pipe size selection is so important in keeping designed-in system pressure drops to a minimum.

Similarly, when pressure increases, and cfm and pipe diameter remain constant, pressure drop decreases. This relationship is also a function of velocity and Bernoulli's principle. As the pressure increases the density of the air increases and its velocity reduces. The reduced velocity results in a reduced pressure drop. This is one of the confounding relationships that make leaks in a compressed air system very costly. The more the leakage, the less is the pressure and the greater the velocity of the air; resulting in increased pressure drop. Figure 4.4 lists the most common sources of pressure drop within the compressed air system.

SOURCES OF PRESSURE DROP WITHIN A COMPRESSED AIR SYSTEM

- **Plumbing**
  - Piping
  - Hose
  - Fittings
  - Valves
- **Equipment**
  - Filters
  - Dryers
  - Aftercoolers
  - Moisture separators
  - Regulators
- **Leaks**

Figure 4.4 Common sources of pressure drop

A sound design practice is to select pipe size and system pressure so that 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 drip legs in main distribution lines, 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, standard operational pressures is tabulated in Tables 4.5, 4.6, 4.7, 4.8. Use Table 4.9 to determine the pressure loss due to friction at any pressure condition. These tables are based upon non-pulsating flow in a clean, smooth pipe.

Effects of pressure drop.

Pressure drop within a compressed air system is unavoidable, but it must be managed for the system to achieve its maximum operational efficiency. Pressure drop wastes energy, it confounds compressor controls, and excessive pressure drop can adversely affect equipment operation and create quality issues with the work being performed by compressed air. 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 in an attempt to overcome these parasitic pressure drops. This increased compressor discharge pressure

requires additional energy. For example, if the compressor discharge pressure is 100 psig and the pressure at the point of use is 70 psig, the system has a 30 psig pressure drop, which is considered excessive. If the user requires 80 psig for proper operation, then the obvious resolution is to increase the compressor discharge pressure to 110 psig to yield the required 80 psig at the point of use. Remember that for positive displacement compressors, every 2 psig increase in compressor discharge pressure results in a 1% increase in consumed compressor horsepower. In this example, a 10 psig increase in pressure requires a 5% increase in horsepower.

In designing an air distribution system where a given diameter piping may be sufficient for the current flow and pressure, 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.

As covered under the compressor Capacity Control discussion within Chapter 2, compressor controls operate within a deadband between a load pressure and an unload pressure. Any pressure drop occurring in the supply region of the system, between compressor discharge and the point where the compressed air enters the main distribution header piping, reduces the deadband by an equal amount. For example; a load/no load compressor has a load pressure of 100 psig and an unload pressure of 110 psig. This is a 10 psig deadband. If the supply has an 8 psig pressure drop, then the useful deadband is reduced to 2 psig. With such a narrow deadband, any variation in the demand of the system will cause the compressor to rapid cycle. The obvious fix to this condition is to increase the unload pressure setting of the compressor. This fix causes the compressor to compress air to a higher-than-needed pressure, which has proven to be extremely wasteful. Pressure drops need to be reduced to and maintained at their minimum to ensure optimum system efficiency.

Table 4.5 Loss of Air Pressure Due to Friction @ 60 psig

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	0.94	0.12					
100	19.60	....	....	42.8	9.95	4.40	1.18	0.15					
125	19.4	....	....	46.2	12.4	6.90	1.83	0.23					
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, e.g., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.




Table 4.6 Loss of Air Pressure Due to Friction @ 80 psig

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	3.46	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	.042				
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, e.g., 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 @ 100 psig

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, e.g., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.




Table 4.8 Loss of Air Pressure Due to Friction @ 125 psig

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, e.g., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

Table 4.9 Factors for Calculating Loss of Air Pressure Due to Pipe Friction Applicable for any Initial Pressure*

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			
1,100	....	....	....	....	....	....	723	88.4	21.0	2.4			

Chapter 4




Table 4.9 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
1,150	....	....	....	....	....	....	790	96.6	22.9	2.6			
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.

Design with the End in Mind

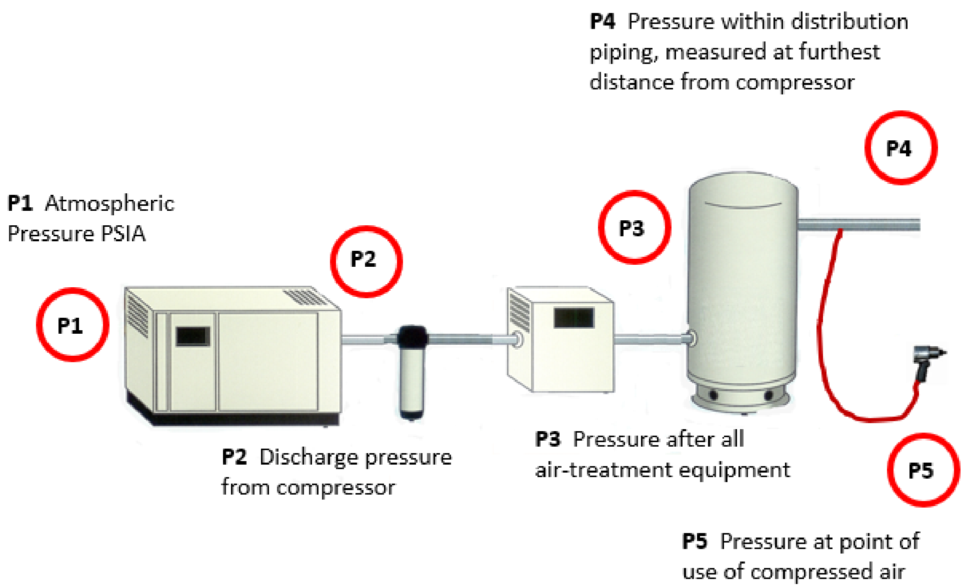


Figure 4.5 Industry-accepted pressure naming convention used for compressed air systems

Figure 4.5 illustrates the industry-accepted pressure naming convention used for compressed air systems. These strategic locations, P1 through P5, mark key locations within a compressed air system and they should be used when reporting pressures within the system. Reducing pressure drop within each location is critical for the efficient operation of any compressed air system.

When designing a compressed air system, it is a best practice to begin with the end users. For each end user, determine the absolute minimum pressure that is required for the application to function as designed. The most energy efficient way to operate your compressed air system is to operate at the minimum allowable pressure required by the highest volume user in the facility. The application with the highest minimum allowable pressure with the most flow requirement establishes your critical pressure. Users with pressure requirements lower than the critical pressure can be satisfied by installing properly sized pressure regulators just prior to the point of use. Users with pressure requirements higher than the critical pressure can be satisfied by a number of options that will be covered later in the System Efficiency section of this chapter.

As shown in Figure 4.6, the correct way to determine the required compressor discharge pressure, P2, is to begin with the critical pressure application, P5. In this example, the critical pressure is 80 psig. From the critical pressure, design backwards toward the compressor adding up the pressure drops introduced by each system component, including piping. This pressure will be the minimum allowable P2 pressure at the compressor discharge; 120 psig in the example. By selecting air treatment components that deliver the least pressure drop for the maximum operating conditions of flow and temperature, you can get the minimum allowable P2 to its minimum. Work with the manufacturers of compressed air components to develop strategies that can reduce the minimum allowable pressure in the system to as low as possible and still deliver expected results.

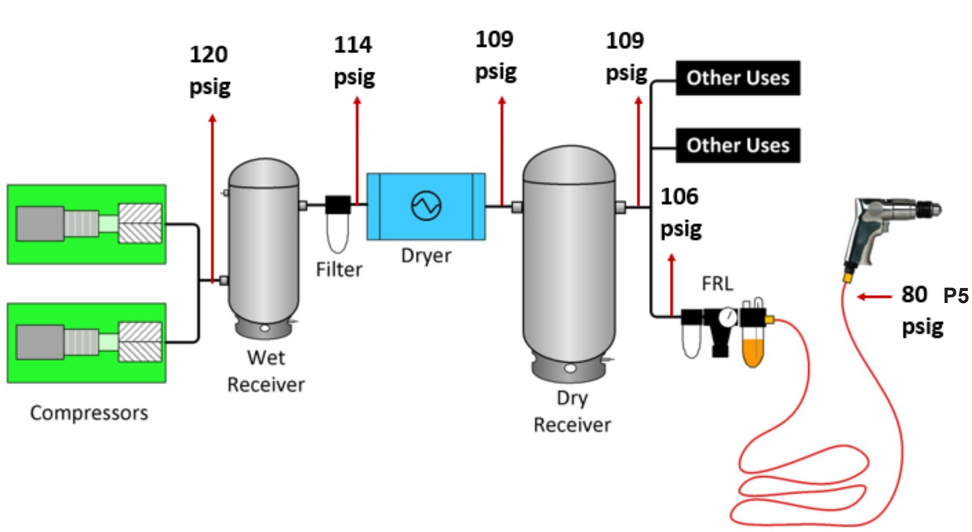





Figure 4.6 Start P2 compressor selection with an analysis of P5 critical pressure

Table 4.10 Discharge of Air Through an Orifice

Gage Pressure before Orifice, psi	Diameter of Orifice, 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

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 all end users that operate at line pressure consume more air and leakage rates also increase. Any unregulated user will consume more air as the pressure increases. This can be seen from Table 4.10 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. This increase in total system demand that is a function of the increase in pressure is referred to as Artificial Demand. Since a 2-psi increase in compressor discharge pressure results in a 1% increase in required horsepower, the combined effect on power consumption of increasing system pressure from 100 psig to 110 psig is approximately 14%; 5% in power for the pressure increase, approximately 14%. The 10 psig pressure increase accounts for 5% of the power increase. The artificial demand created by the increase in pressure results in an increase of 9 cfm through a 1/4" orifice. This is an 8.6% increase in cfm that requires an equal increase in power required to compress the additional air. A compressed air system should be designed and operated so that it generates and maintains the lowest practical pressure.

As illustrated in Figure 4.6, one method of minimizing artificial demand is to install pressure regulators in front of every user so that user pressure, P5, can be adjusted to its minimum. Another method is to install a Pressure-Flow Controller (PFC). The PFC is normally located downstream of the control storage 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 psig. The stable downstream pressure can be set to 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 the PFC can reduce total demand by eliminating artificial demand, the compressors may still operate at a higher than required discharge pressure in order to maximize the useful volume of stored air in the control receiver. To gain maximum compressor efficiency, compressor discharge pressure should be reduced to its minimum allowable for proper system functioning.

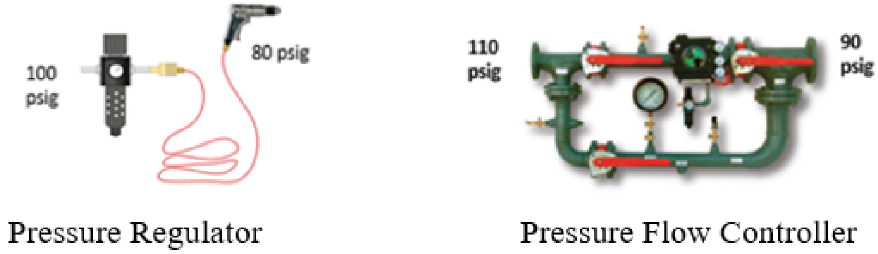


Figure 4.7 Pressure regulation devices for managing artificial demand

Air quality

Different applications for compressed air require different levels of air purity for the application to function reliably, efficiently, and productively. Compressed air quality can vary from hot, dirty, wet, and oily air that is used to power a construction jackhammer to cool, dry, clean, and oil-free air that is used in medical applications. Improving air quality costs money and it is important to know just how much treatment

is required to achieve the desired air quality. Too much treatment is too expensive, insufficient treatment leads to equipment unreliability and high maintenance expenses. As discussed in Chapter 3, Compressed Air Treatment and illustrated in Table 4.11, the ISO 8573-1 Compressed Air Contaminants and Purity Classes specification was developed to provide standard air quality classifications that can be followed when designing compressed air systems. By providing a metric for air quality, the ISO 8573-1 specification allows the system designer to know the effect on air quality of every component within a compressed air system. The final system air quality will be the sum of the effects of all system components.

Table 4.11 ISO 8573-1 Compressed Air Contaminants and Purity Classes

Class	Particles				Water		Oil
	By Particle Size (maximum number of particles per m^3)			By Mass	Vapor Pressure Dewpoint		Liquid, Aerosol, & Vapor
	0.10 – 0.5 microns	0.5 – 1.0 microns	1.0 – 5.0 microns	mg/m^3	ºC	ºF	mg/m^3
0							
1	20,000	400	10	-	≤ -70	≤ -94	< 0.01
2	400,000	6,000	100	-	≤ -40	≤ -40	< 0.1
3	-	90,000	1,000	-	≤ -20	≤ -4	< 1
4	-	-	10,000	-	≤ +3	≤ +37	< 5
5	-	-	100,000	-	≤ +7	≤ +45	-
6	-	-	-	0 – 5	≤ +10	≤ +50	-

The application of the compressed air at the points of use determines the quality of the air that is required. Air quality considerations include the content of particulate matter, water, and oil. The ISO 8573-1 Compressed Air Contaminants and Purity Classes specification describes the standard purity levels of compressed air and these specifications should be utilized when designing a compressed air system. It is wise to contact the manufacturers of process machinery and other pneumatic devices to determine the ISO 8573-1 class of compressed air that is required to operate their equipment reliably and efficiently. Once the required ISO 8573-1 air quality class is determined, equipment selection can begin with this air quality benchmark as the goal.

The quality of compressed air that is delivered to the various points of use in a compressed air system is determined by a variety of factors within the system. As we learned in Chapter 2, Compressed Air Production, compressors can be either oil-free or oil injected in their compression process. If the required ISO 8573-1 system air specification is very strict on limiting contaminants, then it might be wise to begin with oil-free compressors. Doing so will eliminate the intensive filtration that is required to treat the air from oil injected compressors to arrive at the higher air quality specification.

Chapter 3, Compressed Air Treatment, thoroughly describes the equipment available to meet the specific ISO 8573-1 classes, including various types of dryers, filters, moisture separators, and drains. Once the compressed air is generated and treated, it must travel through oftentimes extensive lengths of piping to reach its intended point of use. This compressed air distribution system itself can contribute to the contamination problem, particularly if standard steel or iron piping and air receivers are used. Later in this chapter, there will be a thorough discussion regarding the piping material options that are available to system designers that limit corrosion contamination and wasteful pressure drop.

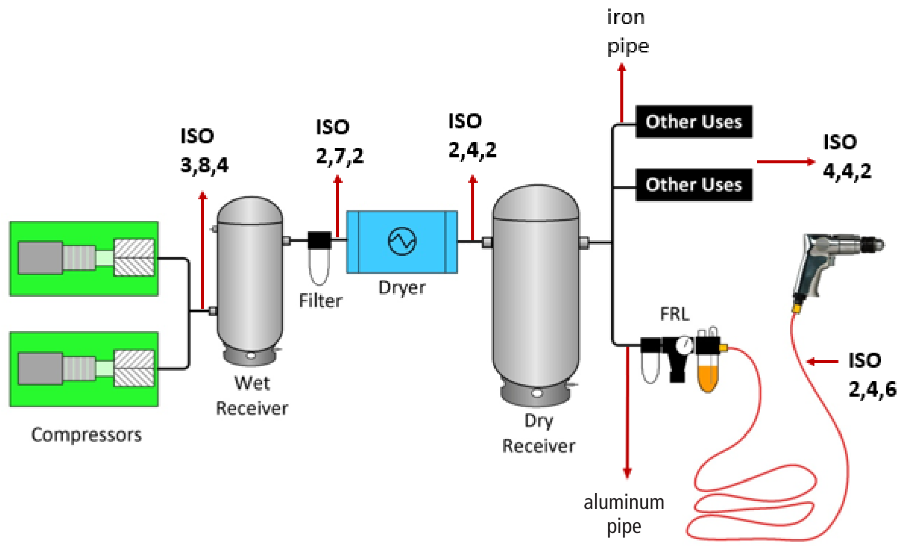


Figure 4.8 Air quality requirements as a systems approach

It is important to view air quality as a function of the total compressed air system as illustrated in Figure 4.8. Every component of a compressed air system affects the quality of the air at the points of use. The final air quality is the sum of all of the effects that the system has on air quality. For example, in the above system the supply and air treatment equipment deliver ISO 8573-1 class 2,4,2 quality air to the main distribution header. If the header piping material imparts corrosion particulate into the air, the final air quality can degrade to an unacceptable level. In the example above, the use of iron piping on the upper branch of the distribution system allows the air to be contaminated with rust and corrosion particles from the interior walls of the pipe. This results in a high particle contamination of the air and increases its ISO particulate class from 2 to 4, which may be too high for the reliable functioning of the application receiving such contaminated air. Similarly, the FRL will yield an air quality of 2,4,6 due to its filtration of particulate and addition of oil from the lubricator.

Type and Number of Air Compressors

There are many different types of compressors in various sizes. Earlier in this chapter, we learned how to calculate the required scfm flow of a compressed air system. The task now becomes one of determining what type, size, and quantity of compressor to purchase and install. These decisions have a fundamental impact upon the reliability, efficiency, and productivity of the system.

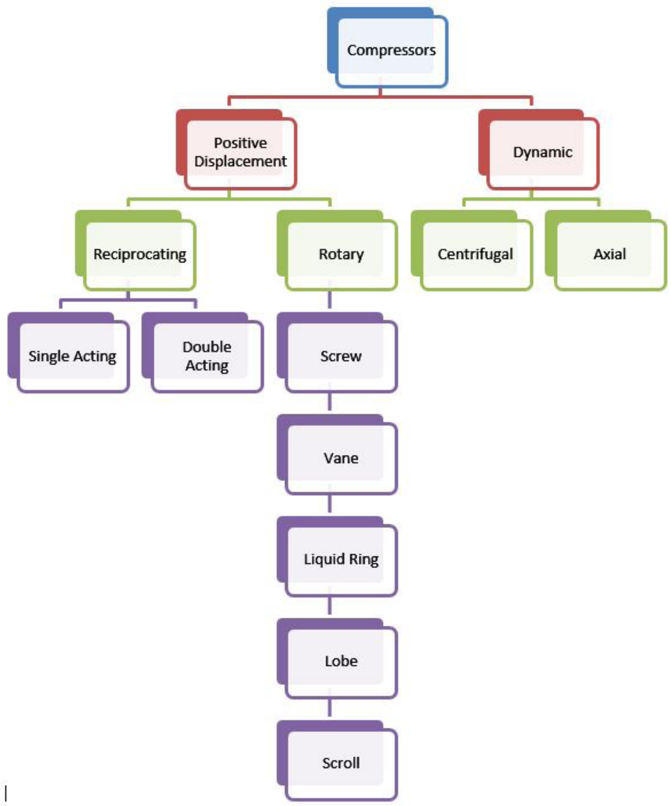


Figure 4.9 Types of Compressors

Air compressors vary in design and performance characteristics. The two basic types are Dynamic compressors and Positive Displacement compressors as described thoroughly in Chapter 2, Compressed Air Production and illustrated in Figure 4.8. Although there is some overlap amongst compressor performance, each compressor has its optimum range of capacity and pressure. This variety can make selecting the right compressor confusing, but by following some basic guidelines, you can narrow the choices to a manageable few.

Certain system characteristics favor one compressor type over another. Centrifugal compressors are best suited for relatively high volume, base load applications where they operate consistently in a fully-loaded condition. Although pressures up to 10,000 psig are possible, most industrial centrifugal air compressors operate in the 100-125 psig range. Capacities start at about 1500 cfm, 300 horsepower, extending up to and exceeding 100,000 cfm. Typical centrifugal compressor applications include automotive assembly operations, large industrial weaving operations utilizing air-jet looms, large blow-molding operations manufacturing plastic drinking cups, and numerous applications within process industries, such as petro-chemical.

Air cooled reciprocating compressors are best suited for applications having a capacity requirement of 100 cfm or less, 30 horsepower, with an intermittent demand profile. Standard industrial pressure ranges of 100 psig to 175 psig are common. Smaller sizes generally run on a start/stop type of capacity control requiring an air receiver storage tank with a significant pressure difference between the start and stop settings. Larger sizes may run continuously with loading and unloading being controlled by pressure settings. In some cases, a specific point of use application requiring a pressure that is higher than the normal P4 distribution pressure might best be satisfied by dedicating one of these compressors to it rather than increasing the P4 to accommodate the unique application. Typical applications for air cooled reciprocating compressors are automotive repair and body shops, small dry-cleaning laundry operations, oil change facilities, and small manufacturing applications.

Double-acting, water cooled reciprocating compressors, once the workhorse of industry, are now best suited for specialty applications, such as high pressure or non-air gas compression. These compressors are efficient in operation, but require a relatively large installation space, extensive foundations, and frequent maintenance. Typical double-acting reciprocating compressor service is high pressure PET bottle blowing or nitrogen compression which is used extensively in the oil production industry for well stimulation.

Rotary compressors have become the work horses of plant air systems and rotary compressors of various types are available up through 3,000 acfm with pressures up to 200 psig, although most operate around 100-125 psig. These are available both oil injected and oil-free and have a variety of capacity control types available to allow them to efficiently match their output to the varying demands of the facility. Being easy to install, operate, and maintain, rotary compressors serve the majority of modern industries that require a reliable supply of clean compressed air at a stable pressure.

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. Some plants operate only one shift per day with air demand being very constant throughout the shift. Depending upon the demand fluctuation, one compressor might be able to handle the full range of daily demand efficiently and reliably. Other plants operate three shifts each with a separate demand profile, depending upon what is happening during each shift. In plants having three shifts with widely differing demands, the base load compressor should be capable of handling the demands of the least loaded shift with an additional compressor or compressors running only for the other shift(s) with larger demands. When considering systems with multiple types of compressors, it is important to consider long-term maintenance needs and procedures. Operating a number of identical air compressors can minimize replacement parts considerations and make maintenance procedures more uniform and easier.

Oil-free air might be required for all applications or only for some specific applications that require higher air quality. If only one or a few applications require oil-free air, there are two options for approaching this situation other than supplying oil-free air to the entire facility. One option is to draw air from the plant air distribution system and filter it immediately prior to the point of use that requires the higher air quality specification. The other option is to operate a separate oil-free compressor to supply the applications with the higher air quality requirements. In general, oil-free compressors have a higher initial cost and higher maintenance costs than their lubricant injected counterparts, but they have the advantage of simple condensate disposal due to the fact that the condensate is uncontaminated with lubricant.

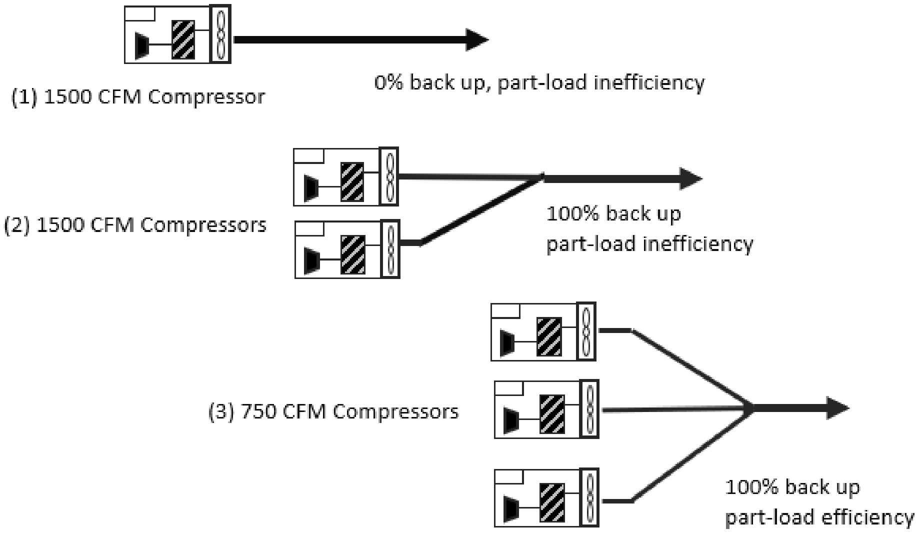


Figure 4.10 Compressor redundancy options

One of the most important aspects when designing or upgrading any compressed air system is to consider the cost of lost production should the supply of compressed air fail. In engineering terms, redundancy is the duplication of critical components or functions of a system with the intention of increasing reliability of the system, usually in the form of a backup or fail-safe. In many installations, the cost of installing redundancy to the system, including both compressors and air treatment equipment, is far outweighed by the cost of lost production due to the failure of an air compressor. Consider the example as shown in Figure 4.10. Plant air demand varies from 500 cfm to 1500 cfm throughout the day during three, 8-hour shifts of operation. In a single, 1500 cfm compressor installation, should it fail, there would be no compressed air and no production. Also, with this one-compressor installation, the 500 cfm demand would deeply part-load the compressor and depending upon its capacity control method, this could be very inefficient operation. With two, 1500 cfm compressors there is 100% redundancy, but the part-load inefficiency still exists. With three, like-sized compressors, each sized at 50% of the maximum 1500 cfm demand, there is 100% redundancy in the event of any one compressor failing. This arrangement also provides part-load efficiency since running the 750 cfm compressor to handle the 500 cfm demand is more efficient than supplying the 500 cfm demand from a 1500 cfm compressor.

With any of the above options, it is always a best practice to have a rental compressor port installed into the supply side of the system to facilitate a quick installation of a rental compressor should the need arise, which it will. Maintenance needs also must be taken into consideration and a number of identical air compressors can minimize replacement parts inventory considerations. This same logic can be applied to the air treatment equipment; filtration and air dryers.

Site Conditions and Their Impact on Compressor Selection and Sizing

Once the system demand profile, required pressure, air quality, and number and types of compressors are determined, it is necessary to size the equipment, especially the compressors, so that they deliver the required scfm capacity at the actual site conditions where the compressors will be operating. The geographic location of the compressed air system plays an important role when determining the compressor capacity and the overall compressed air system design.

Elevation affects the amount of air that the compressor delivers into the system. When customers purchase an air compressor or compressors, they have a specific need. Their application requires a specific amount of air (scfm) compressed to a specific pressure (psig). The geographic location of the compressed air system plays an important role when determining the volume of air that is required and what size of compressor to select. As previously discussed, atmospheric pressure varies with elevation. The closer to sea level the application is, the greater the atmospheric pressure and the denser the air.

A cubic foot of Denver air is less dense than a cubic foot of air at Miami Beach, and a cubic foot of air at the bottom of Death Valley is denser yet. The atmospheric pressure affects the density of the air that the compressor intakes, which in turn affects the amount of compressed air the compressor delivers into the system. The denser the air, the more scfm delivered, all other factors being equal.

Table 4.12 Density of Air at Different Temperatures

Density of Air (lb/ft^3) at Different Temperatures

Air Temp. (°F)	Gauge Pressure (psi)				
	0	5	10	20	30
30	0.081	0.109	0.136	0.192	0.247
40	0.080	0.107	0.134	0.188	0.242
50	0.078	0.105	0.131	0.185	0.238
60	0.076	0.102	0.128	0.180	0.232
70	0.075	0.101	0.126	0.177	0.228
80	0.074	0.099	0.124	0.174	0.224
90	0.072	0.097	0.121	0.171	0.220
100	0.071	0.095	0.119	0.168	0.216
120	0.069	0.092	0.115	0.162	0.208
140	0.066	0.089	0.111	0.156	0.201
150	0.065	0.087	0.109	0.154	0.198
200	0.060	0.081	0.101	0.142	0.183
250	0.056	0.075	0.094	0.132	0.170
300	0.052	0.070	0.088	0.123	0.159
400	0.046	0.062	0.078	0.109	0.141
500	0.041	0.056	0.070	0.098	0.126
600	0.038	0.050	0.063	0.089	0.114

Temperature affects the amount of air the compressor delivers into the system. According to Charles' Law, the volume of a fixed amount of air at a constant pressure varies directly with temperature; a temperature rise causes air to expand and a temperature decrease causes air to contract. As illustrated in Table 4.12, a cubic foot of air at a given pressure weighs less at a higher temperature; the air becomes less dense. This is how a hot air balloon operates; hot air inside the balloon is less dense than the surrounding cooler air and the balloon floats upward. Similarly, the density of inlet air affects the amount (weight) of air that the compressor delivers to the system. With all other conditions being equal, the same compressor will deliver less weight of air on a 90°F day than on a 68°F day.

Chapter 4

Table 4.13 Partial Pressure of Moisture at Various Temperatures

Temp. ºF	Ambient psi	Temp. ºF	Ambient psi	Temp. ºF	Ambient psi	Temp. ºF	Ambient psi	Temp. ºF	Ambient psi	Temp. ºF	Ambient psi
32	0.008854	49	0.1716	67	0.3276	85	0.5959	103	1.0382	121	1.7400
33	0.0922	50	0.1781	68	0.3390	86	0.6152	104	1.0695	122	1.7888
34	0.0960	51	0.1849	69	0.3509	87	0.6351	105	1.1016	123	1.8387
35	0.1000	52	0.1918	70	0.3631	88	0.6556	106	1.1345	124	1.8897
36	0.1040	53	0.1990	71	0.3756	89	0.6766	107	1.1683	125	1.9420
37	0.1082	54	0.2064	72	0.3886	90	0.6982	108	1.2029	126	1.9955
38	0.1126	55	0.2141	73	0.4019	91	0.7204	109	1.2384	127	2.0503
39	0.1171	56	0.2220	74	0.4156	92	0.7432	110	1.2748	128	2.1064
40	0.1217	57	0.2302	75	0.4298	93	0.7666	111	1.3121	129	2.1638
41	0.1265	58	0.2386	76	0.4443	94	0.7906	112	1.3504	130	2.2225
42	0.1315	59	0.2473	77	0.4593	95	0.8153	113	1.3896	131	2.2826
43	0.1367	60	0.2563	78	0.4747	96	0.8407	114	1.4298	132	2.3440
44	0.1420	61	0.2655	79	0.4906	97	0.8668	115	1.4709	133	2.4069
45	0.1475	62	0.2751	80	0.5096	98	0.8935	116	1.5130	134	2.4712
46	0.1532	63	0.2850	81	0.5237	99	0.9210	117	1.5563	135	2.5370
47	0.1591	64	0.2951	82	0.5410	100	0.9492	118	1.6006	136	2.6042
48	0.1653	65	0.3056	83	0.5588	101	0.9781	119	1.6459	137	2.6729
		66	0.3160	84	0.5771	102	1.0078	120	1.6924		

The water vapor content of air affects the amount of air that the compressor delivers into the system. Ambient air always contains water vapor, more in hot weather than in cold weather. As illustrated in Table 4.13, as air temperature rises, more water vapor is contained in the air and this water vapor increase is measured by the partial pressure that it exerts. The more water vapor, the greater the partial pressure. In a cubic foot of ambient air, this water vapor occupies space that air could occupy if there were no water vapor present, a condition of 0% relative humidity. During the compression process, most of this water vapor gets condensed and removed from the final discharge air as liquid condensate. So, to determine the amount of dry air that the compressor delivers into the system, the intake volume needs to be reduced by the amount of volume that the water vapor occupies, which is removed. The scfm flow specification can now be converted to an acfm compressor capacity.

$$acfm = scfm \times \frac{P_s}{[P_a - (PP_{wv} \times rh)]} \times \frac{(T_a + 460)}{(T_s + 460)}$$

Where:

P_s = Standard pressure, psia
P_a = Atmospheric pressure, psia
PP_{wv} = Partial pressure of water vapor at ambient temperature
rh = Relative humidity
T_a = Ambient temperature, ºF
T_s = Standard temperature, ºF

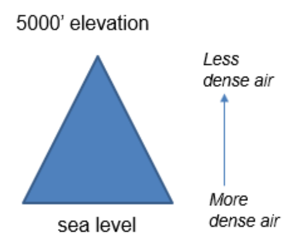
Figure 4.11 Conversion formula; scfm to acfm

Most compressed air requirements are specified in scfm, which translates into a specific weight of air delivered every minute of compressor operation. All compressors are rated in acfm capacity, the volume of air that the compressor delivers into the system. scfm does not equal acfm except on the one-time condition when the compressor is operated at standard conditions. In that case, scfm equals acfm. As previously learned, inlet pressure, relative humidity, and temperature affect the scfm performance of a compressor, i.e., how much weight-flow the compressor delivers into the system. The formula shown in Figure 4.11 is used to convert the required scfm weight-flow to the acfm volume-flow that will be required at a specific geographic location. Once the acfm performance is determined, this capacity can be used to select the size of compressor required based upon manufacturer's ratings, which are reported as acfm.

Since scfm compressor performance is significantly affected by elevation, temperature, and relative humidity, it is wise to size the compressor for the worst-case conditions of temperature and relative humidity. By doing so, this assures that the compressor will always be large enough to supply the required scfm. Some geographic locations have wide changes in ambient temperature from day to night and from season to season. As temperature and relative humidity fall, the compressor will deliver more than its required scfm. Understanding this relationship allows selection of the right compressor with the proper capacity control that will efficiently manage compressor output to match the varying demand.

Site Conditions

Example: Determine acfm



Requirement:

- 1000 scfm
- Altitude 5000 ft above sea level
- Maximum ambient temperature 100°F
- Maximum Relative Humidity 50%

- Atmospheric pressure at 5000 ft. = 12.2 psia
- Partial pressure of moisture at 100°F from vapor pressure chart = 0.95 psia
- Partial pressure at 50% RH = 0.95 x 0.50

Use CAGI standard conditions: 14.5 psia, 0% RH, 68°F

$$1000scfm\ x\ \underbrace{\frac{14.5}{[12.2-(0.95\ x\ 0.50)]}}_{\text{pressure adjustment}}\ x\ \underbrace{\frac{(460+100)}{(460+68)}}_{\text{temperature adjustment}} = 1000\ x\ \frac{14.5}{11.725}\ x\ \frac{560}{528} = 1311\ acfm$$

Figure 4.12 Example conversion formula; scfm to acfm

In this example, the scfm-acfm conversion formula is used to determine the acfm rating of the compressor that is required to deliver 1000 scfm when the compressor is operated at a location that is at 5000 feet above sea level, has a maximum ambient temperature of 100 degrees Fahrenheit, and has a maximum relative humidity of 50%. The standard conditions used are the CAGI standard conditions of: 14.5 psia, 0% RH, 68°F.

In the first part of the formula, the scfm was adjusted by the pressure differences caused by elevation and the partial pressure of water vapor. These are pressure differences between site conditions and standard conditions. The numerator is the standard pressure, 14.5 psia. The denominator is the actual inlet pressure (12.2 psia) minus the product of the vapor pressure at 100°F (.95 psi) times the relative humidity (50%).

In the second part of the formula, the scfm was adjusted by the temperature differences. These are temperature differences between site conditions and standard conditions. Temperatures are in Rankine which is the actual Fahrenheit temperature plus 460°F. So, the numerator is the ambient temperature (100°F) plus 460°F. The denominator is the standard temperature (68°F) plus 460°F.

The completed arithmetic demonstrates that to deliver the 1000 scfm at the installation location, the compressor must have a rating of at least 1311 acfm. If a 1000 acfm compressor was supplied, not understanding the difference between scfm and acfm, the compressor would be undersized by 311 cfm, about 24%. This could be a very costly mistake, one that is often made in sizing compressors.

A properly designed compressed air system will not only handle current conditions, but it should be designed with the future in mind. It makes good business sense to plan for any potential changes that the system may experience. Such potential change events could be as follows:

- Business is good and the operation needs to be expanded by adding additional production lines.
- The 5-year plan includes introducing a new product in year 5 that will require oil-free air.
- A new innovative manufacturing process is implemented, that requires a significantly higher pressure than the current compressor can deliver.
- Is it better to have one compressor to deliver the full supply or would it make sense to divide the supply amongst 3 like-sized compressors, each capable of supplying 50% of the current demand? In so doing there is allowance for one compressor to be a back-up unit to allow for full supply even in the event of one compressor failing or being down for service.

All of these considerations should be studied when designing a compressed air system.

Air Distribution and System Layout

Proper design of the distribution system is essential to avoid energy waste and to ensure proper use of all pneumatic devices. Piping and the plumbing components used to connect piping into a closed air-distribution system are a major source of manageable pressure drops. Sizing the pipe properly for the expected flow is important in keeping wasteful pressure drops to their minimum. Compressor placement as well as selecting the type of distribution network, either looped or trunk and branch, are critical factors that determine the efficiency of the final compressed air system.

There are many, different piping materials available for compressed air systems, each with their own characteristics regarding internal friction, safety, corrosion resistance, ease of installation, and cost. Some of the common materials are aluminum, copper, steel (black and galvanized), stainless steel, and rubber hose.

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. Federal, state and local codes should be consulted before deciding on the type of piping to be used. Most often ANSI B31.1 is applied. Special applications, such as compressed air systems for healthcare facilities, are guided by specific, industry-accepted specifications and regulations such as the National Fire Protection Association standard, NFPA 99.

It should be noted that the roughness of the internal bore will be a factor in the amount of pressure drop experienced; both the roughness when new and the roughness after years of service. The smoother the surface of the internal diameter of the pipe, the less friction exists between the compressed air and the pipe wall. The less friction, the lower the pressure drop when air flows through the pipe. The extrusion process imparts a very smooth, low-friction internal wall in aluminum and copper piping and tubing. Both materials are corrosion resistant, so they retain their low friction qualities throughout their service lives. Welded and seamless steel and stainless-steel pipes are both manufactured using a drawing process that provides a relatively smooth internal surface. Whereas stainless-steel resists corrosion and will retain its smooth internal bore over its lifetime, steel pipe will rust in the presence of humidity and liquid water. This rusting will increase the friction factor of the piping surface, increasing pressure drop as the pipe ages. For this reason, steel pipe is often galvanized to reduce the susceptibility to rust and corrosion. Rubber hose presents the highest friction factor of all of the above materials and its use as compressed air piping should be kept to as short a length as is possible to reduce the pressure drop through the hose.

In compressed air piping systems using oil-free compressors, corrosion-resistant piping should be used as there is no lubricant carryover to coat and protect the inside surface of the pipe. An oil-free system might experience internal pipe corrosion from the moisture in the warm air as the relative humidity of the air approaches 60%.

Also, when systems operate on extremely low dew point air, such as -100°F, there is a tendency for the dry air to attract moisture from the ambient environment. This phenomenon is a function of Fick's Law of Diffusion which states, "The rate of diffusion in a given direction is proportional to the negative of the concentration gradient." In other words, the ambient air has a much higher concentration of water vapor in it than does the -100°F dew point system air and the ambient water vapor tends to try to equalize with the extremely dry air within the system. Another phenomenon that can introduce unwanted moisture into an extremely low dew point air system via leaks is known as the Joule-Thomson effect. Accordingly, as compressed air exits through any leak, rapid expansion takes place, which absorbs heat from the surroundings. This endothermic process chills the surface of the pipe near the leak. If liquid water condenses on the chilled pipe surface, this liquid water can migrate into the opening, diffuse along the metallic surface, and be drawn inside into the dry air, where it evaporates. Water vapor will diffuse into the system through any available entry point, such as a leak or porous piping. Such diffusion will make it impossible to maintain the required low dew point. To prevent this diffusion, system piping in such low dew point systems should be welded stainless steel or sweated copper to eliminate all possible diffusion points.

PVC, thermoplastic piping is not recommended for compressed air service. Per OSHA, only one type of plastic pipe has been approved for use with compressed air. That pipe, Acrylonitrile-Butadiene-Styrene (ABS), is marked on the pipe as approved for compressed air supply. By law, employers must protect their workers by avoiding the use of unapproved PVC pipe in such systems. The pressure rating of PVC piping is normally stated at 80°F and compressed air discharge temperatures oftentimes approach 120°F which can significantly degrade the pressure rating of the PVC pipe. Some synthetic compressor lubricants can degrade both the PVC material and the adhesives used in pipe assembly. Plastic piping at floor level is also more vulnerable to accidental impact damage than is a more durable, metal piping. OSHA has issued a safety hazard information bulletin regarding the use of PVC plastic pipe for transporting compressed gases including air in above ground applications. If plastic piping is used in a compressed air system, the manufacturer's specifications and limitations must be observed and the design must be based upon a thorough evaluation of the system.

Once the type of piping material has been selected, the next variable that affects the pressure drop experienced within the pipe is its size, specifically its internal diameter. As the velocity of air increases, its flow becomes turbulent and this turbulence creates resistance to flow; i.e., pressure drop. For a given cfm flow rate, the smaller the pipe diameter the greater the velocity of the air and the greater the pressure drop. As discussed earlier, a sound design practice followed to keep pressure drop to a minimum is to select pipe size and system pressure so that air velocity through the distribution piping should not exceed 30 ft. per second. To avoid liquid water from being entrained in the airflow and carried beyond drainage drip legs in main distribution lines, the velocity should not exceed 20 ft. per second. Branch lines having an air velocity over 33 ft. per second 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.

The main header and distribution piping should be sized to take into account anticipated future expansions and flow requirements. If the initial piping is sized only for present flow requirements, any increase in supply in response to additional demand requirements could result in significantly increased pressure drops in the entire system. Upsizing the pipe by one size larger than is required to handle the existing flow provides for future expansion. Upsizing will not add significantly to material and installation costs. A rough rule of thumb is that by doubling the internal diameter of a pipe, its cross-sectional area increases by four times, the velocity of a given cfm flow rate is reduced to one-fourth, and the pressure drop is reduced by a substantial amount. Over the lifetime of the compressed air system, the upsizing of the distribution system will most likely pay for itself in the form of energy savings by reducing parasitic pressure drops.

Table 4.14 Loss of Air Pressure Due to Friction @ 100 psig

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	1.28	6.50	0.99	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

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, e.g., for 500 ft, one-half of the above; for 4,000 ft, four times the above, etc.

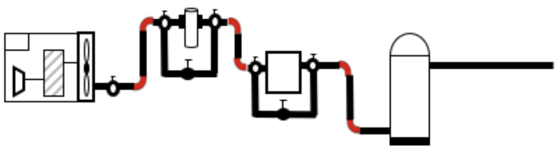
Table 4.14 shows the expected pressure drop of air travelling through clean and smooth steel pipe of various nominal diameter sizes. These pressure drop tables are useful in calculating the total expected pressure drop within a compressed air system under various conditions of pressure and pipe size. This table shows the pressure drop in 1000 feet of pipe with an initial pressure of 100 psig. For example; if a 100 hp rotary screw compressor discharges its full capacity of 500 cfm @ 100 psig into a 1000-foot length of 2" diameter steel piping, the pressure drop over the full 1000 feet will be 19.2 psig. Pressure drop is directly proportional to the length of pipe. So, using the above example; the pressure drop across 500 feet of 2" diameter steel piping will be one half of that experienced with 1000 feet of pipe. 500/1000 x 19.2 = 9.6 psig.

When installing compressors, contractors often incorrectly use piping of the same size that matches the discharge port of the compressor. This pipe size might be too small and the system will suffer an excessive and extremely wasteful pressure drop. As an example, most 100 HP rotary screw compressors have 2" discharge ports. If this general practice of matching pipe size to compressor discharge port size is followed for a 2-100 HP compressor system of 1000 cfm, the system begins with an excessive pressure drop handicap. Excessive pressure drop can quickly rob a system of its reliability and efficiency, and pressure drop never improves, it normally worsens with the ageing of the system.

Table 4.15 Loss of Air Pressure Through Screw Pipe Fittings, Given as Equivalent Length in Feet of Schedule 40 Pipe.

Nominal Pipe Size Inches	Actual Inside Diameter Inches	Long Radius Elbow	Standard Elbow	Tee Through Side Outlet	Globe Valve	Gate Valve
1/2	0.622	0.62	1.55	3.10	17.30	0.036
3/4	0.824	0.82	2.06	4.12	22.90	0.048
1	1.049	1.05	2.62	5.24	29.10	0.061
1 1/2	1.610	1.61	4.02	8.04	44.70	0.094
2	2.067	2.07	5.17	10.30	57.40	1.21
3	3.068	3.07	6.16	15.30	85.20	1.79
4	4.026	4.03	7.67	20.20	112.00	2.35
6	6.065	6.07	15.20	30.40	168.00	3.54

As is shown in Table 4.15, pressure drop tables exist that show the expected pressure drop of air travelling through the various fittings and valves used to plumb a compressed air distribution system. The amount of pressure drop is given as an equivalent length of similar-sized pipe, which is added to the total length of piping to arrive at a total pressure drop for the system. As the table indicates, the more turbulence that the fitting or valve creates, the greater the resistance to flow and the greater the pressure drop.



Pipe length calculation

	Piping and Equivalent Length Fitting Adjustment	
	2" piping and components	3" piping and components
Piping	500'	500'
5 Gate Valves	6.05 equivalent feet 1.21 x 5	8.95 equivalent feet 1.79 x 5
6 Long Radius Elbows	12.42 equivalent feet 2.07 x 6	18.42 equivalent feet 3.07 x 6
Total Piping	**518.47 feet**	**527.37 feet**

Pressure drop calculation

	pressure drop @ 500 cfm with 2" piping and components	pressure drop @ 500 cfm with 3" piping and components
Piping	9.95 psid 518.47/1000 x 19.2 psid	1.23 psid 527.37/1000 x 2.34 psid
Filter	2 psid	2 psid
Dryer	3 psid	3 psid
System Total	**14.95 psid**	**6.23 psid**

System conditions and assumptions

- 500 cfm compressor, fully loaded
- 50' of pipe in compressor room
- 450' of pipe in distribution header
- 5 open gate valves
- 6 long radius elbows
- Pipe is schedule 40 steel
- Filter pressure drop is 2 psid
- Dryer pressure drop is 3 psid

Figure 4.13 Example of Calculating Expected Pressure Drop Through a Simple System

Figure 4.13 shows an example of how to utilize the information in Tables 4.14 and 4.15 to calculate the expected pressure drop of compressed air travelling through a simple compressed air system. The system contains one, 100 HP compressor rated at 500 cfm @ 100 psig, one 500 cfm rated filter, and one 500 cfm rated dryer. There is a total of 500 feet of schedule 40 steel piping used to plumb the system. The air flows through 5 open gate valves and 6 long radius elbows as it flows to the points of use at the end of the header. A comparison is made between 2″ piping and 3″ piping. The "Pipe Length Calculation" table uses the equivalent pipe length method to calculate the total amount of piping in each of the two scenarios.

The "Pressure Drop Calculation" table uses the total piping length from the previous table to calculate the total pressure drop within the system; piping, fittings, valves, filter, and dryer. There is no pressure drop through a receiver. As the results show, the 2″ system has a 2.4 times greater pressure drop than does the 3″ system, operating under the same flow conditions. This example illustrates a frequently-encountered situation where the piping is too small and the system suffers an excessive and extremely wasteful pressure drop.

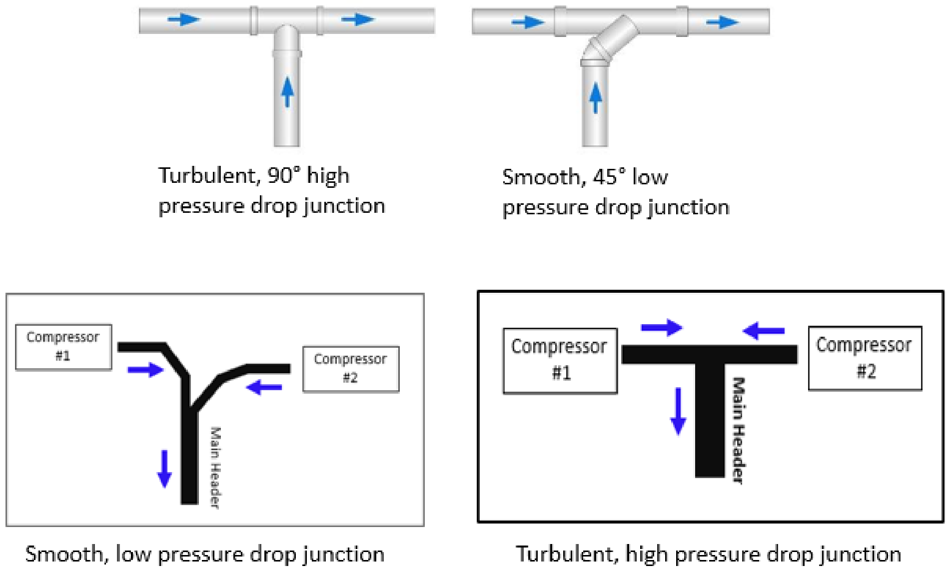


Figure 4.14 Various piping transitions and their associated pressure drops

A significant source of excessive pressure drop occurs at pipe junctions. As described in Figure 4.13, right angle or head-on junctions of compressed air streams create extensive turbulence at the junction which creates resistance to flow and an accompanying pressure drop. Where possible, long radius 45° elbows should be used to introduce flow tangentially from one pipe into another so that both streams are flowing in the same direction. This reduces both turbulence and pressure drop. Flows from several different compressors feeding a main header in the compressor room oftentimes merge head-on which creates tremendous turbulence and significant pressure drop. Tangential entry in the direction of flow is always the preferred method for piping multiple compressors into a main header.

The compressor room header into which the air compressor(s) merge should be sized so that the air velocity within the header does not exceed 20 feet per second. Higher velocities tend to entrain liquid condensate back into the airstream, blowing the condensate across the drip leg and rendering it useless. This airborne liquid can quickly foul the internal surfaces of dryers and filters, degrading their performance.

Compressor Room Location

Several factors affect the decision to satisfy compressed air demand with one, centralized system or with multiple, smaller systems positioned close to the points of use. These factors include space, cost, demand variability, maintenance expense and power availability. Each of these layouts has certain advantages as well as disadvantages and we will explore them all.

Figure 4.15 depicts a centralized system layout where all of the compressors and their associated air treatment equipment are housed in one central location within the facility. Compressed air enters the distribution piping from one point and the distribution piping is responsible for delivering the compressed air to its eventual points of use at the required pressure and volume. The further the end user is from




the compressor room, the longer the distance the air must travel to reach the point of use and the greater the pressure drop will be. Looped distribution systems, like the one shown in Figure 4.15, have the advantage over single trunk distribution systems in that air can be delivered to any user from two directions, effectively reducing the distance that the air must travel through the pipes. Piping layouts will be explained further in this chapter.

The advantages of a centralized system include simplified maintenance and, if required, reduced operators since all compressors are in one location that has been designed for compressors. The main compressor room usually has an overhead crane to make installing, removing, and maintaining compressed air equipment easy. If rental compressors are needed, the main compressor room affords one, simple connection. Locating all compressors within close proximity of each other simplifies controlling the compressors. Having one compressor room simplifies the expensive job of running the required utilities, power and water, and limits the need for noise attenuation to one location, away from the work place. The proper ventilation to keep compressors running reliably and efficiently is more easily provided in one main room than in multiple satellite compressor rooms which might be small, unventilated rooms in the middle of the plant. Similarly, having one compressor room simplifies and promotes the use of heat recovery systems since the associated utilities are centralized and accessible.

The main disadvantage of operating a centralized compressor room is if the distribution and storage systems are not designed correctly from the beginning. Usually as a result of cost-cutting, an improperly designed system will have marginally-sized distribution pipes and significant, wasteful pressure drops will be designed into the system from the start. The distance from the compressor room to the furthest point of use must be considered as extensive lengths of piping will create increased pressure drop, potential leaks, and, when run outside, potential line freezing problems. Also, those applications furthest from the central compressor room supply most likely will experience insufficient flow and insufficient pressure unless properly-sized storage is strategically placed within the system. Frequently in a centralized system, the compressed air equipment is combined with boilers and other heat-generating equipment in a centralized "powerhouse." Unless it is well ventilated, the powerhouse will have a hot atmosphere that will cause frequent compressor issues and increased maintenance expense. Again, planning for future growth and changes is paramount if a centralized system is to be efficient.

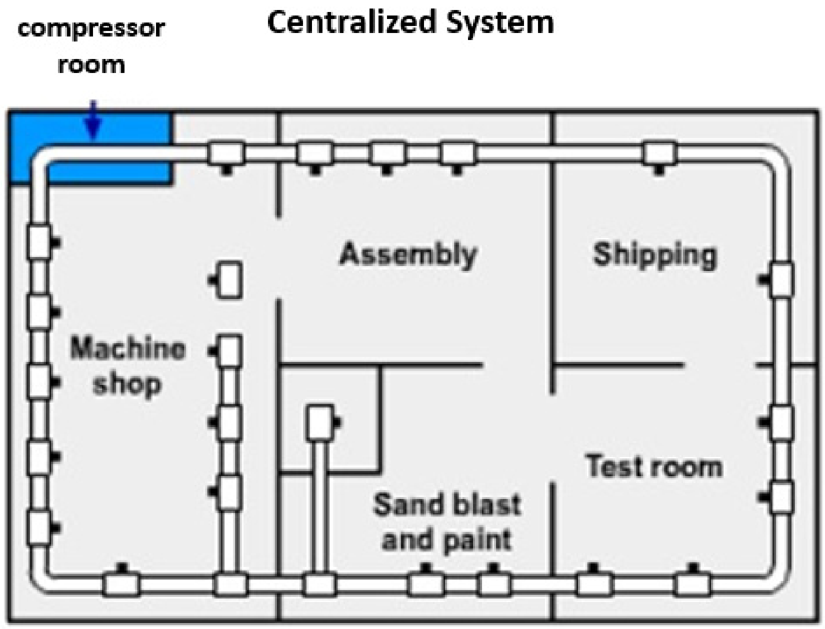


Figure 4.15 Centralized system layout

Figure 4.16 depicts a de-centralized system layout where there are multiple compressor rooms, 5 in this example, each with its own compressors and air treatment equipment. Compressed air enters the distribution piping from 5 separate locations. De-centralized systems can be designed so that applications having similar air requirements, pressure and air quality, can be located in close proximity in a "cell" and served by a localized compressor. With a de-centralized system, pressure drop can be controlled tighter than with centralized systems as the air in a de-centralized system has a shorter distance to travel to reach the end users. With cells of significantly differing pressure requirements, the compressed air needs can be best served by having dedicated compressor rooms rather than operating the entire system at an elevated pressure just to satisfy one cell that requires the higher pressure.

In cells that have high volume applications, having a dedicated compressor room can mitigate the system control problems associated with intermittent, high volume events. Most de-centralized systems have isolation valves in the distribution piping that allow a cell to be isolated totally from the main compressed air piping in the event of maintenance requirements or equipment relocation within the cell. Isolation valves allow the individual compressors to act together, or separately as the system demand determines.

One unfortunate reason for de-centralized systems is due to a lack of planning for future growth with the initial system design. Systems often start out as centralized systems and as plant operations grow and expand, additional compressors are added. If no expansion allowances were planned for in the main compressor room, then the additional compressor equipment must be installed in a separate compressor room, not always in the most desired position to affect system efficiency. This creates a multiple entry-point system with its associated control problems. It is best to consider future expansion possibilities when designing compressed air systems.

De-Centralized System

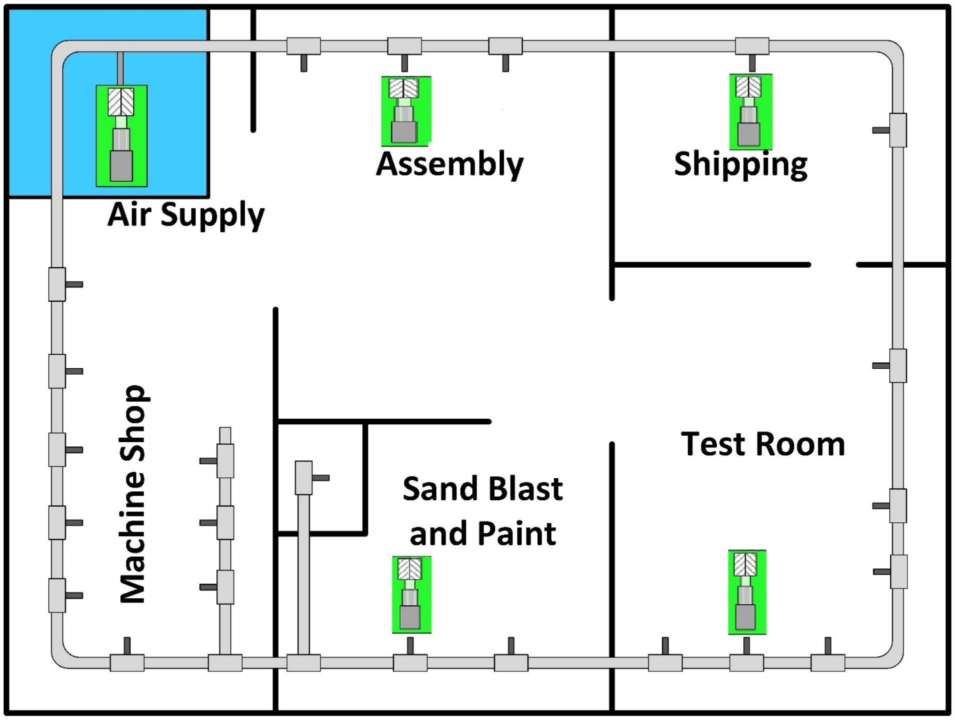


Figure 4.16 De-Centralized layout

One advantage of operating de-centralized systems is that the supply is located close to the demand and requires short piping runs which minimize pressure drops and piping cost. Operating specific compressors for applications in a cell that require the same pressure eliminates running the entire system at a higher-than-needed pressure just to satisfy the high-pressure cell applications. This can be a significant savings in energy over the lifecycle of the system. Similarly, if different cells have differing air quality specifications, the dedicated compressor approach can eliminate the costly and wasteful practice of over-filtering or over-drying the entire plant supply just to satisfy the needs of one cell or application. Also, by having each cell supplied by its own dedicated supply, the events of one cell are isolated from the other applications in the plant.

Maintenance may be neglected if there are multiple compressor rooms within the system. Satellite compressor rooms are often small, not well ventilated, and lacking in the heavy lifting equipment required to perform major compressor repairs and overhauls. Power and possibly water will need to be brought to the multiple compressor rooms and this can be very costly. Controlling multiple compressors with multiple system entry points is impossible without installing expensive master system controls that have been properly selected for the control requirements.

Regardless of whether the installation is a centralized or de-centralized layout, an important, but often overlooked, consideration is the source of inlet air to the compressor. Drawing process air from the compressor room might result in the compressor room having a negative pressure, especially if the room is a tightly-sealed structure. Such negative pressure reduces the efficiency of the compressor as well as the ability of the coolers to perform their task if the compressor is air cooled. Proper ventilation is required to eliminate this condition.

On the other hand, process air drawn from outside the compressor room should be from a location where contaminants such as industrial gases, chemicals, and particulates will not be a problem. When the air intake filter for the compressor(s) is mounted remotely, the inlet air piping from the air intake filter to the compressor inlet must be clean and, being at atmospheric pressure, may be of PVC plastic material. Care must be taken to size the intake piping sufficiently so as not to create a pressure drop through the pipe length. 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 to assure that users receive the quality of air that they require for reliable operation.

Distribution Piping Layouts

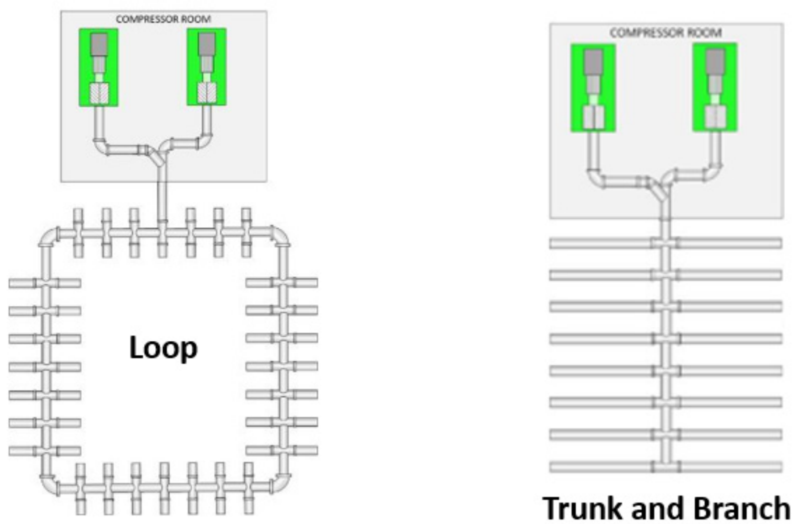


Figure 4.17 Two basic distribution piping layouts

The two types of compressed air piping system layouts are the Loop system and the Trunk and Branch system as illustrated in Figure 4.17. In the Loop system, the main distribution piping, also known as the header, forms a complete loop. Drops are made from the header at points where compressed air is needed. In the Trunk and Branch system, the header dead-ends at its farthest point and branches are piped off of the header to deliver compressed air to various sections of the plant. Both the header and the branches dead-end. Drops are made from the Branches at points where compressed air is needed.

When possible, the distance from the air compressor or compressors to the point of compressed air use should be minimized. This is because the further the air must move through a pipe, the greater the pressure drop. A loop distribution layout allows air to reach any user from two directions. The maximum distance air must travel through the system is one-half the total length of the loop, reducing pressure drop by one-half as compared to the pressure drop in a similar length of straight pipe as would be the case in a trunk system. Also, since air can reach the user from two directions, the volume of air from each feed direction is less than the total required. Less flow means less pressure drop. This also allows for smaller header pipe to be installed; a cost saving. However, care must be taken not to reduce header pipe size too great as the smaller the pipe, the greater the velocity of the air and the greater

the pressure drop. With a properly designed distribution loop, the compressor room can be placed anywhere along the loop.

A long narrow plant might have a distribution system consisting of a straight "trunk" header pipe with branches to each point of use. Although this layout might be less costly than a loop layout, the trunk and branch layout exposes users at the distant branches to the possibility of chronic low pressure as the air must travel the full length of the piping to reach them. Pressure drop as a result of pipe friction will decay the pressure as the air travels the long distances, as illustrated in Figure 4.18. If the distant users are experiencing low pressure, the solution usually is to increase the P2 discharge pressure of the compressor to compensate for the pressure lost during transmission.

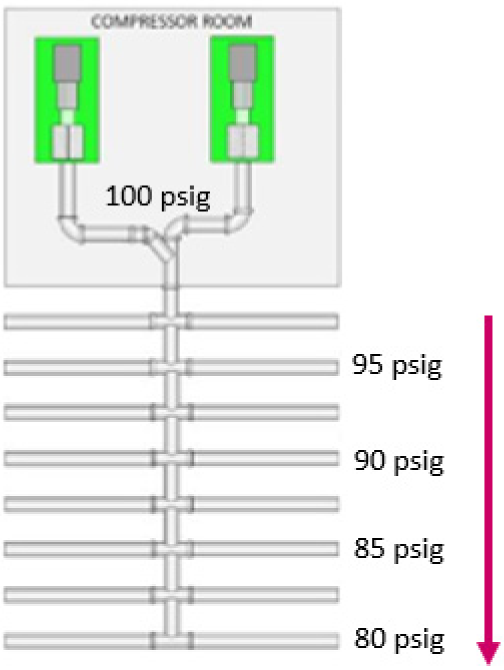


Figure 4.18 Pressure drop in a trunk distribution layout

Compressing to a higher-than-needed pressure is energy inefficient since for every 2 psig increase in pressure, the consumed horsepower is increased by 1%. The fact that air has only one direction of travel in a trunk and branch layout can cause users at the distant ends to suffer significant pressure fluctuations if large volume, intermittent users are at the beginning of the trunk. These events will be first served, leaving little volume to satisfy distant users. If a trunk and branch layout is selected, it must be designed with oversized piping in both the trunk and the branches to minimize the effects of excessive pressure drop. By oversizing the piping, pressure drop decreases and the header becomes a large storage vessel that will delay the pressure fluctuations as air is used throughout the system. Even if a trunk and branch layout is oversized, it does not allow for easy growth and expansion of the system. A new, higher-pressure user cannot be placed at the distant end of the trunk and must be placed closer to the compressor. This limits expansion flexibility and often leads to installing a separate compressor room with all of its associated control and maintenance issues.

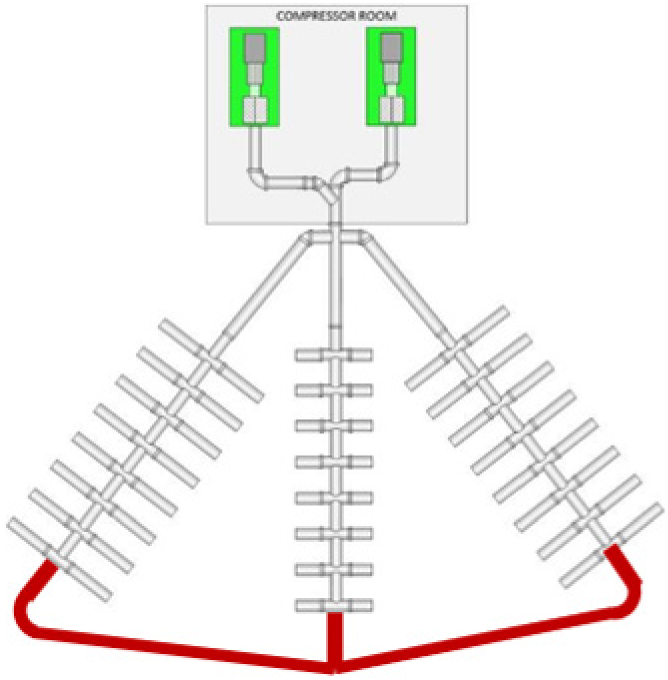


Figure 4.19 Piping optimization

Oftentimes in situations where piping layouts create excessive pressure or flow abnormalities, the layout can be optimized to increase system efficiency and reliability. As shown in Figure 4.19, a 3-Trunk and Branch distribution piping layout was installed to serve three separate locations within a multiple function plant. As pressure drops and varying demands caused severe low-pressure events at the distant ends of the trunks, the trunks were joined to create a looped layout (red pipe joining trunks). Due to the advantage of air being able to flow in multiple directions to any one point of use, the pressure drops within the trunks were equalized and the low-pressure issues were eliminated.

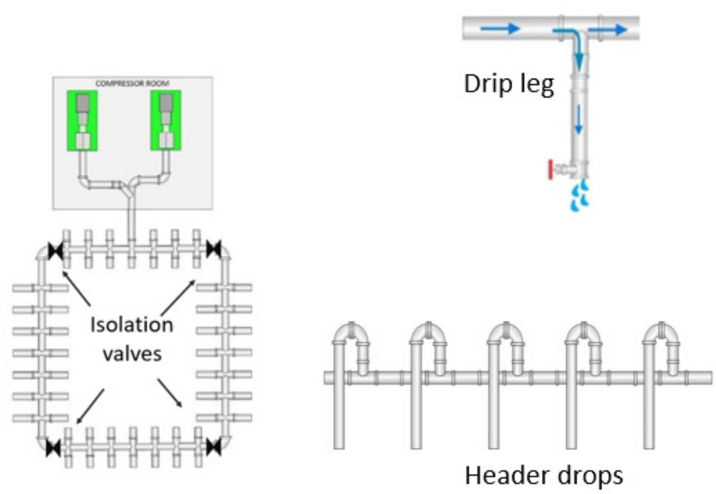


Figure 4.20 Piping best practices

Piping Best Practices

Any piping that is operated at an elevated temperature should be shielded from physical contact or the pipe should display a suitable high-temperature warning sign. Piping must be adequately supported to eliminate sag and weak points and allowance for thermal expansion must be considered on long piping runs. Pressurized piping should be located away from passage ways where fork lift trucks and other vehicles could accidently come into contact with it. As shown in Figure 4.20, good system design places isolation valves in strategic locations within the header piping. This practice allows a section of the system to be isolated for equipment relocation or maintenance purposes without having to bring the entire system offline.

Piping from the header to each point of use should be kept as short as possible. Air piping from the header to the point of use should be taken from the top of the header. If condensation accumulates within the header, it will remain on the bottom of the pipe. Taking air from the top of the header eliminates the chance of liquid condensate contaminating the drop airstream. These pipes normally run vertically downward from an overhead header to the point of use and pipe should be sufficiently sized so that pressure drop from the header to the point of use does not exceed 1 psi during the duty cycle. On the other hand, drip legs, should be plumbed from the bottom of the header so that condensate will flow down by gravity. All piping should be arranged with a slight slope away from the compressor and towards the end users. This will eliminate condensate from dripping back into the compressor during periods when the compressor is idle. Drip legs should be plumbed at strategic, low-points in the header to remove collected condensate. Headers and piping also should have an ample number of tapped connections to allow evaluation of air pressure at points throughout the system.

System Efficiency

Efficient operation of the compressed air system requires that pressure fluctuations be minimized. Pressure fluctuations have three main sources: pressure drop due to resistance of flow through components, fluctuations in the demand for air, and the capacity control method of the compressor. Minimizing and managing pressure drop requires a total system approach in both design and maintenance of the compressed air system. The capacity control method used by the compressor to match its supply to the actual system demand affects the pressure of the system. Capacity control methods for rotary screw compressors are discussed in Chapter 2. Capacity control induced pressure fluctuations as well as pressure fluctuations caused by large demand events can be controlled with the proper application of storage volume.

Pressure Drop

As previously discussed, all components through which compressed air flows generate a pressure drop. The magnitude of the pressure drop increases directly with the flow rate and temperature of the air. Accordingly, all compressed air components should be sized for the highest operating conditions of these two variables, not for the average flow and temperature of the system. Piping, aftercoolers, valves, moisture separators, dryers, and filters can add significantly to drops in pressure if they are undersized for the worst-case conditions. All system components should be selected based upon these conditions and the manufacturer of each component can supply pressure drop information based upon these operating conditions. Remember, the goal of a well-designed compressed air system is to keep the pressure drop between compressor discharge (P2) and the point-of-use to less than 10% of the P2 pressure. Specify pressure regulators, lubricators, hoses, and quick-disconnects that have the best performance characteristics at the lowest pressure differential. The added cost of these correctly-sized components should be recovered quickly from the resulting energy savings.

Users with pressure requirements that are lower than the critical pressure can be satisfied by installing properly sized pressure regulators just prior to the point of use. If there are applications that require compressed air at a significantly higher pressure than the previously-established critical pressure, there are five choices to handling this issue.

1. Operate the whole facility at the higher pressure, which will be costly due to the need to compress air to a higher-than-needed pressure and to the wasteful artificial demand this higher system pressure creates. Operating at an increased pressure also increases unwanted stress on all air-operated components.
2. Replace the application or modify the components to operate at a lower pressure.
3. If the system includes a pressure flow controller (PFC), supply the high-pressure application by running a dedicated pipe to the application from the high-pressure side of the supply in front of the PFC. The majority of the system will continue to run at its lowest critical application pressure.
4. Isolate this application from the rest of the facility and use a smaller compressor set at a higher pressure to feed this one application.
5. Lastly, install a booster pump or pneumatic amplifier to generate a higher pressure at the discharge of the booster that can operate the high-pressure application. These booster pumps use a significant amount of P4-pressure compressed air to pump line-pressure air to a higher pressure.

As discussed earlier in this chapter, undersized distribution piping can introduce wasteful and significant pressure drops into any compressed air system. Filters, too, are a major source of pressure drop because they are designed to plug up and as they accomplish their filtering task, their pressure drop increases due to the fouling effect. A new filter might have an initial pressure drop of 1 psig, but after several months of service that pressure drop might increase to 6 psig. Oftentimes filters are selected to match the diameter of the pipe into which they are plumbed rather than by the peak flow and temperature conditions to which they will be subjected. Too small of a filter will compound the pressure drop through the filter by a square function. It is a wise practice to oversize filters by a factor of 1.5 based upon the maximum flow and temperature conditions. For example; if a system has a maximum peak flow of 400 cfm, select a filter that is rated to handle 600 cfm (400 cfm x 1.5 = 600 cfm).

Once the proper equipment is installed, the recommended maintenance procedures should be followed and documented to maintain the minimum pressure drop through the system. Restriction and friction create pressure drop. Anything that impedes the flow of air through the system will create an unwanted pressure drop. Friction and restriction come from two major areas; corrosion and contamination. Corrosion within the piping creates excessive friction and results from exposing metal surfaces to air that has a relative humidity of 60% or greater. Contamination comes mainly from the compressors as a result of dirty intake air and excessive oil carryover. Both corrosion and contamination can be controlled by performing standard maintenance upon all equipment on a scheduled basis.

Proper compressor maintenance limits the amount of contaminants, dirt and oil, that get discharged into the compressed air system. The narrow, internal passages of refrigerated dryers can quickly become fouled with oil and dirt from a neglected compressor and this reduces the ability of the dryer to remove water vapor from the air. Ferrous piping will corrode quickly under these conditions, and its rough internal surface will introduce significant pressure drop into the distribution system. Neglected filters plug quickly and impart excessive pressure drop into the system. Neglected drains either fail open, which creates a continuous and wasteful leak, or they fail closed, in which case the condensate that they collect re-entrains into the airstream and blows downstream to further corrode piping and components. With constant use, fittings and quick disconnects at the terminal end of the distribution system get loose and leak, adding to pressure drop. These components should be checked regularly and replaced at first sign of leakage.

Managing a compressed air system to minimize wasteful pressure drop is an ongoing responsibility. First and foremost, it is critical to understand what pressure requirement is actually needed by the end use equipment or processes. Manufacturers often inflate minimum pressure requirements as a safety-factor when designing equipment. Care should be taken to receive accurate minimum pressure and flow requirements for each piece of equipment that uses compressed air. With proper controls, it is possible to provide a point-of-use pressure within +/- 1 psig and the manufacturer should be asked to supply the minimum acceptable pressure with such tight tolerance control. Obvious restrictions to flow; i.e., excessive elbows in a piping configuration, undersized filters and piping, valves stuck half open, should be identified and eliminated. Fixing leaks is the easiest and quickest way to make a significant reduction in wasteful pressure drop. Operators should strive to operate their systems at the lowest pressure possible. This can be accomplished by slowly reducing P2 pressure over a period of weeks, finally achieving a pressure at which an end user experiences operational issues as a result of low pressure. This becomes the system critical pressure and the goal is to reduce P2 pressure to the lowest possible pressure that delivers the required critical pressure. This is accomplished by monitoring, identifying, and reducing pressure drops within the system.

Reducing system pressure drop begins with the identification of the individual pressure drops within the system. This requires pressure measurements at different points in the system to identify the component or components causing the high pressure drop. To facilitate this analysis, it is a best practice to have pressure monitoring ports installed before and after every flow-restricting component in a compressed air system. This allows taking pressure readings at strategic points without interrupting production. Excessive or progressively increasing pressure drop across a component is an indication that maintenance or replacement is required. Following are some common places to measure pressure drop.

1. In the compressor package. Fouled aftercoolers can introduce significant pressure drops. The discharge pressure of each compressor should be measured.
2. In the piping and pipe fittings. Within the supply piping and at the ends of headers and branch lines to make sure the pressure drop is less than 10% of the P2.
3. Across all filters, dryers, heat exchangers, and moisture separators.
4. Across all regulators. Excessive pressure drop on the regulated side of the regulator during a high-volume event could signal that the regulator is undersized, requiring a higher-than-needed inlet pressure in order to maintain the regulated pressure. This situation often results in operators bypassing regulators or adjusting regulators to the 100% open position so that the application experiences line pressure.

In many compressed air systems, the highest pressure drops usually are found at the terminal points of use. Rubber hose has one of the highest coefficients of friction with compressed air and a long, coiled length of undersized rubber hose can inflict a serious 30 psig pressure drop between the beginning of the hose and the tool. Leaks at the terminal connection to the user are commonplace. These include leaking push-to-lock fittings used on plastic tubing, inexpensive quick-disconnect couplings that do not seal tightly and leak, and leaking components on Filter-Regulator-Lubricators (FRLs). All too often in an attempt to reduce inventory and save money, companies will select a standard sized FRL for all users within the facility. Instead, each FRL should be sized for the peak flow of the individual application that it serves. Undersized FRLs are a major source of pressure drop as the smaller the pipe size the greater the velocity of the air and the greater the pressure drop through the FRL. The distribution piping system is frequently diagnosed as having a high pressure drop just because a point-of-use pressure regulator cannot maintain the required point-of-use pressure.

In a regulated application, if the regulator is set at 85 psig and it is properly sized for the actual flow, but the undersized and/or dirty filter upstream of the regulator has a pressure drop of 20 psi, the pressure in the distribution piping upstream of the filter and regulator would have to be maintained at a minimum of 105 psig. The 20 psi pressure drop, and the need to operate the distribution piping at an elevated 105 psig pressure, might be incorrectly blamed on the distribution piping rather than on the point-of-use components that are causing the excessive pressure drop.

Demand Fluctuations, Storage, and Capacity Control

Compressed air systems are dynamic, meaning that, conditions of flow rate and pressure 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 changes are gradual. This simplifies the controls necessary for maintaining a constant, stable system pressure. However, in most industrial plants, demand fluctuates as a variety of tools are used and as intermittent demand events occur. Often, a demand event occurs at some considerable distance from the compressor(s) supplying the compressed air. If the system was not properly designed to handle such large flow events, pressure drop throughout the extended distribution system can vary erratically. Such large pressure fluctuations confound the compressor controls resulting in the inefficient operation of the compressor.

Theoretically, if the system volume (air receivers plus distribution piping) were large enough, demand event pressure fluctuations would be minimal and the air compressors would see a constant discharge pressure. This constant pressure would simplify the controls necessary for maintaining a constant, stable system pressure. Obviously a grossly oversized system volume is not practical, but it demonstrates that many pressure problems can be eliminated by designing a system with properly sized distribution piping and applying storage in the form of air receivers at proper locations within the system.

Key Storage Terms

Understanding storage within a compressed air system requires the definition of some basic compressed air storage terms. Air is a fluid and without a pressure differential there is no flow. To be effective in performing work, air must flow. To flow, air must be stored at a higher pressure than the pressure at which it is used to do work.

- **Useful differential** is the difference between stored pressure and use pressure. Although air must be stored at higher pressures to do work, those systems that can run compressors at the lowest possible pressures will be more energy efficient.

- **Useful storage** is the amount of air that is stored above use pressure, in volume. Useful storage is a function of two factors: available storage volume, and useful differential. The larger the physical volume of the receiver the more air in storage. The greater the useful differential the greater the useful storage. There must be a pressure differential in a receiver to obtain useful storage. System volume is static, a constant number, but system pressure can vary. The more pressure that is stored in a given volume, the more cubic feet of air are available to do work. Useful storage is measured in cubic feet and represents the amount of air available to be applied to a system event.

- **Rate of Change** is the time component in compressed air systems. Rate of change is the relationship of pressure change, both positive and negative, versus time as a function of flow. When supply exceeds demand, pressure rises and there is a positive rate of change. When the opposite occurs and demand exceeds supply, pressure falls and there is a negative rate of change. If you are lucky enough to have supply equal demand, which rarely happens, then there is a zero rate of change.

The primary function of storage in a compressed air system is to manage the rate of change in the system. By slowing the rate of change, storage stabilizes plant pressure and a relatively constant pressure benefits many applications. For example, with stable pressure, hand tools operate at a constant speed and torque, improving the quality of the work being done. Production is enhanced and scrap and re-works rates are reduced significantly with a stable system pressure. In painting applications, paint spray is consistent, eliminating over-spray or under-spray defects that are caused by pressure fluctuations. Stable pressure allows the system to be more accurately managed to deliver its maximum efficiency.

Supplying peak airflow demand with on-line compressor capacity alone requires one or more compressors to operate in the part-load or unloaded condition, ready to load at a moment's notice. As the peak demand occurs, the compressor(s) will load for a short time during the demand event and then return to part-load or unloaded operation. The result is poor overall system efficiency. By slowing the rate of change, storage provides time for the peak demand event to stop before the system pressure falls to the point that an additional compressor starts. Substantial energy is wasted when an unneeded compressor starts, pumps a gulp of air into the system, unloads, and runs unloaded, ideally for only as long as the functioning auto/stop timer allows. In real-life, the unloaded compressor usually continues to run and waste energy until it is manually shut down. Proper storage prevents unneeded compressors from starting.

By slowing the rate of change, storage provides time for the individual capacity control method of each compressor to properly respond to the change in demand as reflected by the change in pressure. Without time to respond, mechanical and electronic controls often can fight each other, resulting in an uncontrolled system that is extremely energy inefficient. In such a situation, compressors rapid cycle or stop and start at an accelerated frequency, which is mechanically detrimental to the airend and all cycling components. By eliminating or reducing this cycling, storage improves the reliability and longevity of any compressor.

For a compressed air system to achieve maximum operating efficiency, the compressed air supply should incorporate both compressed air generation and storage. The goal is to supply the average air demand with compressors and to supply peak airflow demands from storage. The pressure of the stored air and the size of the air receiver are critical variables in managing the rate of change and properly sizing the storage.

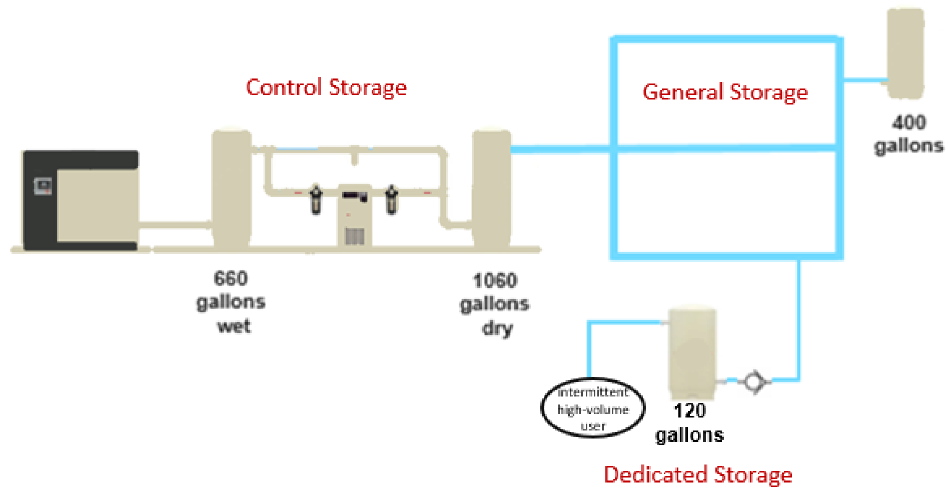


Figure 4.21 Three types of storage in a compressed air system

Types of Storage

As illustrated in Figure 4.21, storage within a compressed air system can be divided into three categories, each with its own specific function; control storage, general storage, and dedicated storage.

- **Control Storage** is the volume of storage on the supply side of the system between the compressors and the point where the supply room header joins the distribution header.
- **General Storage** is the volume of storage in the overhead piping system from the discharge of the compressor room to the end users in addition to any remote mounted receivers.
- **Dedicated storage** is storage that is check-valve-protected and isolated from supplying the entire system in order to serve a specific point-of-use application.

Control Storage

Control Storage is the volume of storage on the supply side of the system between the compressors and either, 1) the point where the supply room header joins the distribution header or (2) in systems with pressure flow controllers, control storage is the volume in front of the pressure flow controller. The primary functions of control storage are to maintain the integrity of system pressure under all conditions and to operate the compressors as efficiently as their controls will allow. Control storage is required to delay an unneeded compressor from starting. Properly sized control storage is also required to keep plant pressure from falling below a minimum level during the time it takes (start permissive) for a compressor to start pumping air into the system after it is started. This start permissive time can be up to 30 seconds in length and during that period, control storage must satisfy the additional plant demand. The start permissive of a backup compressor following the unexpected failure

of a running compressor is oftentimes the largest event that a compressed air system will experience. Properly sized control storage will handle this event without allowing plant pressure to fall below its minimum allowable level.

Control storage is critical in allowing the individual capacity controls of the compressors to function normally and not fight each other. This allows the system to be operated at its lowest possible pressure, which saves energy. Additionally, control storage reduces compressor cycling frequency, which improves the reliability and longevity of the compressor as well as its operational efficiency.

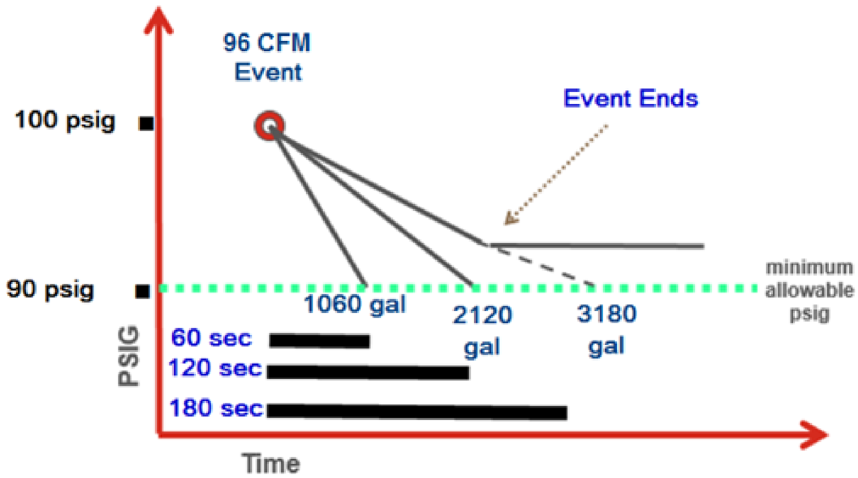


Figure 4.22 Effect of control storage on rate of change

Effect of Storage on Rate of Change

Control storage has a tremendous influence upon rate of change. Larger control storage means slower rate of change. Slower rate of change allows the system to handle more demand events without adding on-line HP. Consider the following example, as depicted in Figure 4.22:

- The minimum allowable pressure, critical pressure, for a system is 90 psig.
- The one operating compressor in the system is fully loaded and maintaining the system at a discharge pressure of 100 psig.
- Someone operates a tool in the plant that consumes an additional 96 cfm.
- Since the compressor is fully loaded and cannot deliver the additional 96 cfm, there is a negative rate of change of 96 cfm and system pressure falls because demand is greater than supply.
- If the control storage is 1060 gallons, it will take 60 seconds, one minute, for the plant pressure to fall from 100 psig to 90 psig.
- If the control storage is doubled to 2120 gallons, it will take double the time, 2 minutes, for the plant pressure to fall from 100 psig to 90 psig.
- If the control storage is tripled to 3180 gallons, it will take triple the time, 3 minutes, for the plant pressure to fall from 100 psig to 90 psig.

As this example shows, the time to reach the critical pressure during any event is directly proportional to the total amount of control storage in the system. By slowing the rate of change, control storage allows time for the system dynamics to change to the point that the demand becomes less than supply and pressure recovers without having to start an additional compressor.

As discussed in the capacity control section in Chapter 2, all compressors require control storage to function efficiently. The method of capacity control utilized by the compressor determines the amount of control storage that is required for the compressor to operate efficiently and reliably.

Compressors operating in either Start/Stop or Load/No Load capacity control require significantly more control storage than do compressors operating in the modulation, variable displacement, and variable speed capacity control methods. In stop/start and load/no load compressors, compressor capacity is not changed. The compressor delivers either 100% of its rated capacity or 0%. Part load capacity is achieved by varying the amount of time that the compressor is loaded and unloaded, the average of these times corresponding to the average cfm of the system demand. An adjustable pressure deadband controls when the compressor is loaded or unloaded.

When a load/no load compressor reaches its unload pressure setpoint, the inlet valve closes, capacity is 0%, the pressure of the airend is what exists in the sump/separator vessel, and consumed power is approximately 70% of full load power. It is not until the pressure in the sump separator vessel has been bled down to its fully unloaded pressure that the fully unloaded power of the compressor is attained; approximately 25% of full load power.

Rapid bleed down of pressure would cause excessive foaming of the oil and increased oil carry-over downstream. To prevent oil foaming, sump bleed-down time is regulated from 30 to 100 seconds. If control storage is inadequate to allow the sump to fully bleed-down, the compressor will re-load before the fully unloaded condition is reached. This is very inefficient operation. Properly sized control storage is essential if a compressor is to operate efficiently in the load/no load control method. As shown in Figure 4.23 the performance efficiency of a load/no load compressor varies directly with the volume of control storage.

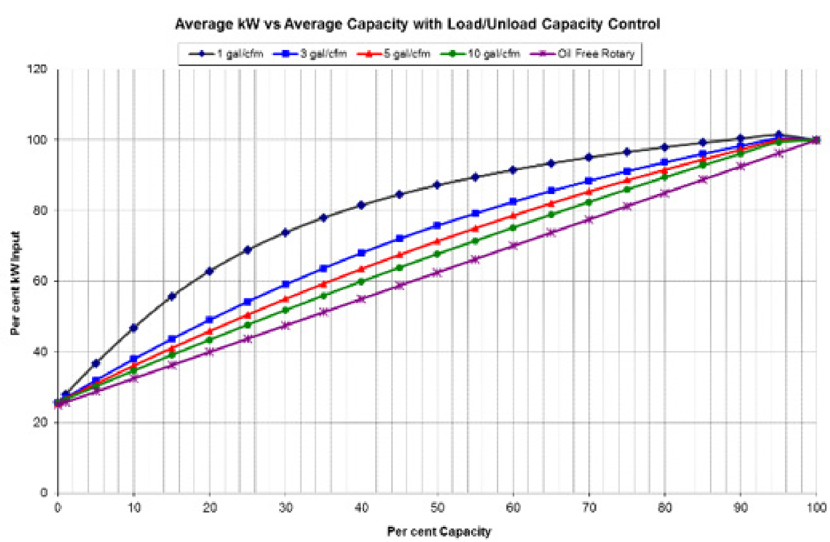


Figure 4.23 Performance curve for a load/no load compressor as a function of control storage volume

With load/no load compressors, part load efficiency depends upon the compressor being able to completely blow down its sump prior to re-loading. Adequate storage to allow this blowdown is required. Compressors that operate on any other capacity control method; modulation, variable displacement, or variable speed, control system pressure by varying the flow of the compressor. The part load efficiency of these compressors is mostly unaffected by the amount of control storage within the system. The primary need of control storage for these compressors is to minimize unnecessary starts and stops and to allow time for other compressors in the system to start and come up to pressure should more online horsepower be required. The amount of control storage required to limit unnecessary starts and stops is significantly less than that required to make a load/no load compressor operate at maximum efficiently.

Control Storage Placement

There are several opinions of where to place the control storage within the supply side of the system, which are shown in Figure 4.24. Storage that is placed prior to the dryer is called wet storage as the receiver stores wet, un-dried compressed air. Wet storage provides some additional measure of radiant cooling of the compressed air as it exits the compressor. Depending upon how long the air resides within the wet receiver, this cooling effect will condense additional water and oil vapor from the airstream. Although wet receivers are not designed to be moisture separators and they will re-entrain considerable condensate into the turbulent airstream within the receiver, they will retain liquid condensate at the bottom of the receiver, which requires the receiver to have a reliable drain valve. If any of the compressors are of the reciprocating type, the wet receiver will act as a pulsation dampener to the pulsations created by the reciprocating compressor. The wet receiver creates no pressure drop and it stores air at the same pressure that exists the compressor. If the compressor unloads, the P2 pressure that the compressor responds to will fall the full amount of the programmed pressure deadband rather than the deadband being reduced by the pressure drop within the supply as is the case when the control storage is placed after the clean-up equipment as a dry receiver. The wet receiver will reduce the cycling of the compressor if it is of a load/no-load type.

One caution with having wet storage is that a large demand event in the system could create a negative rate of change where all compressors are fully loaded and demand exceeds supply and pressure falls. In this case air, will be drawn from the wet receiver, in addition to the full capacities of the running compressors, and this total supply might be too high for the dryer. If this were the case, the dryer would be overloaded and it would not be able to maintain its design dew point. Depending upon the size of the wet receiver and the duration of the event, the dew point of the air leaving the dryer could be significantly elevated and wet air will contaminate the system.

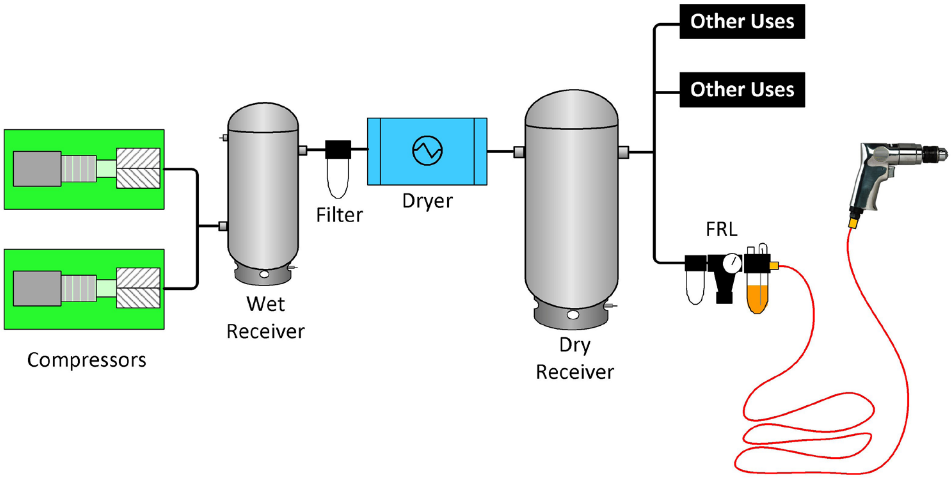


Figure 4.24 Wet and dry receivers as control storage

Storage that is placed after the dryer is called dry storage as the receiver stores dry, compressed air. Dry storage creates a ready supply of dried air to be released immediately into the system to handle events. No further drying is necessary so in a negative rate of change situation, the dryer cannot be accidentally overflown as is the case with wet storage. If a system has only dry storage, there are pressure drops between P2 and the dry receiver due to filters, aftercoolers, and dryers. When the compressor unloads, the useful differential to which the compressor responds is controlled by the dry receiver and it will be reduced by the total supply pressure drop. This reduction in useable differential will cause the compressor to cycle more frequently which will negatively affect compressor efficiency and reliability. This unwanted condition can be resolved by properly sizing the dry receiver to limit the number of compressor cycles to 10 per hour and by maintaining pressure drops to their minimum.

The ideal compressed air system includes both wet and dry storage. This ideal arrangement captures the advantages of both wet and dry storage. The potential of overloading the dryer still exists, but by portioning the control storage as 1/3 wet and 2/3 dry, the risk is mitigated. Remember; storage is the one variable in a compressed air system that can never be oversized. More is always better.

General Storage

As shown in Figure 4.25, general storage is the term given to the volume of storage in the overhead piping and non-checked receivers in the system from the discharge of the compressor room to the point-of-use drops. In this example, the general storage consists of the distribution headers and the 400-gallon remote receiver. It does not include the dedicated 120-gallon checked receiver nor the 1060-gallon dry control storage receiver. Its purpose is to support point-of-use events instantly until control storage or compressor capacity can service the event. Since air has a finite speed or velocity based upon pressure differential in the piping, general storage supports the user during the seconds that it takes for compressor supply or control storage supply to arrive at the event site, supply the required volume, and stop the decay in system pressure. The total amount of general storage, the transmission time from supply, and the size of the event will determine how much the pressure will drop. Inadequate general storage volume usually results in running the system at a higher pressure to increase the storage capacity of the general storage and to increase transmission velocity due to higher differential pressure. This is inefficient as the compressors must compress to a higher-than-needed pressure to offset the inadequate general storage volume. It should be noted that most compressed air system operators grossly overestimate the amount of actual storage that exists in the distribution piping.

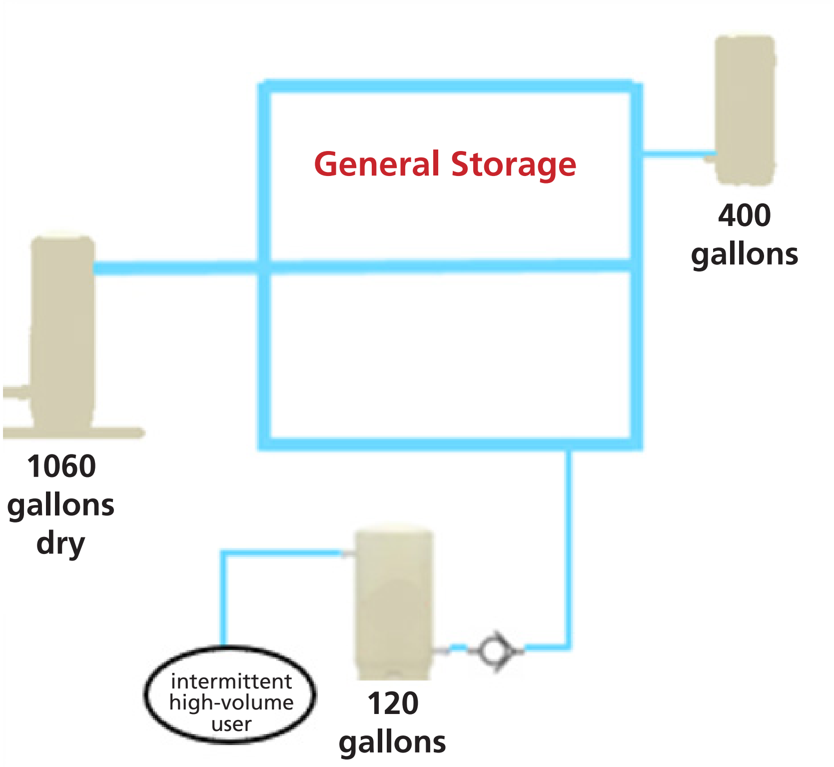


Figure 4.25 General storage constituents

Dedicated Storage

In many industrial plants, there could be one or more applications with an intermittent demand of relatively high volume. This can cause severe dynamic pressure fluctuations in the whole air system, causing end users close to the event to be starved for air which results in quality issues with the starved, end use applications. This disruptive intermittent low-pressure event can be solved by adding the correct amount of dedicated storage to handle the event.

As shown in Figure 4.26, dedicated storage is storage that is isolated from the general storage by some flow-limiting device; a check valve, a metering needle valve, or a restricting orifice. The flow-restricting device prevents the general storage from experiencing the dramatic demand for air, which will be satisfied by the volume stored in the dedicated receiver. Since the time between the events is often longer than the event itself, this flow-restricting device allows the dedicated receiver to be filled slowly, without negatively impacting the pressure in the rest of the general storage. This metered recovery stretches the demand over a long period of time to reduce the surge impact upon the system. The dedicated receiver must be sized properly so that it alone has sufficient volume to handle the event. When properly sized, dedicated storage eliminates the need to operate the general storage at a higher-than-needed pressure, which saves energy.

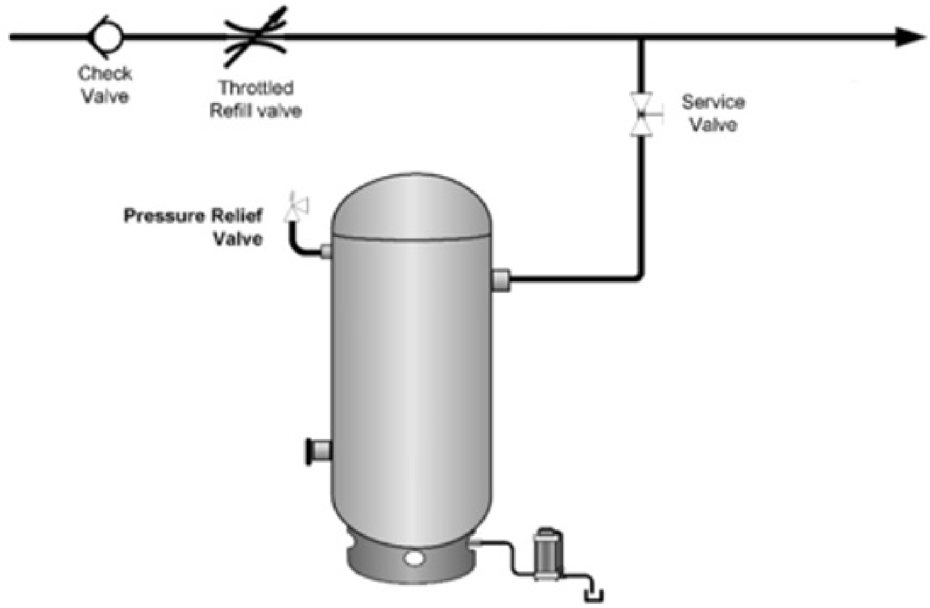


Figure 4.26 Dedicated storage components

Receiver Sizing

There are many different rules-of-thumb for selecting the minimum size of a control storage air receiver. One such rule-of-thumb states that control storage should equal 1 gallon per cfm of the capacity of the largest trim compressor. Another more realistic rule-of-thumb is 10 gallons per cfm capacity of the largest trim compressor. Rules-of-thumb might be convenient, but they are often no better than guessing. Rather, the volume of storage should be calculated based upon system data so that no compressor cycles more than 10 times per hour.

It should be noted that in systems with multiple compressors and sequencing controls, system efficiency is achieved by having most of the compressors running fully loaded on base load with only one compressor on "trim" or part load. It is the trim compressor that controls the pressure of the system and only the capacity of the trim compressor should be considered when sizing the proper control storage to limit the trim compressor to 10 cycles per hour. Additional control storage might be required to protect system pressure in the event of a compressor failure that requires another compressor to start. The additional control storage would cover the start permissive of the added compressor.

The formula illustrated in Figure 4.27 is one of the most important formulas that needs to be understood in order to correctly analyze events within a system and to be able to recommend solutions to compressed air problems involving flow and pressure. This formula can be transposed to solve for any of the four variables within the formula; cfm, pressure, volume, and time. Once three of the four variables are known in this formula, for any given system, the fourth unknown variable can be solved.

In its current form it is solving for the volume of storage required to limit the pressure decay within the receiver to a specific amount when the demand is a specific cfm that lasts for a specific amount of time. It is important to note that the volume is always expressed in cubic feet and time is always in minutes.

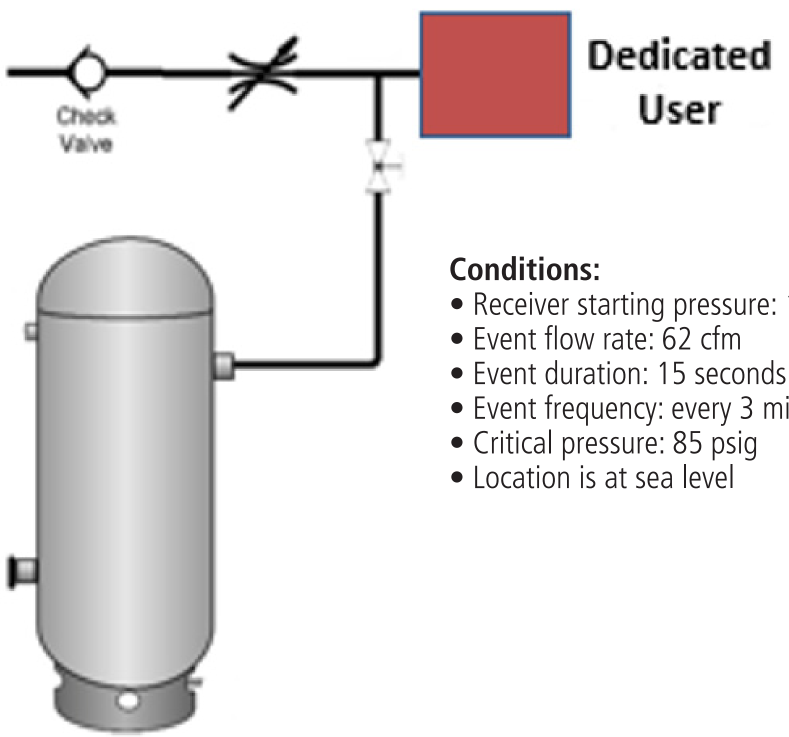
Air Receiver Formula

$$V = \frac{T_m \times C \times P_a}{P_1 - P_2}$$

Where:
V = Receiver volume, ft^3
T_m = Time allowed for allowed pressure drop to occur in **minutes**
C = Air demand, ft^3/min (cfm) free air
P_a = Absolute atmospheric pressure, psia
P_1 = Initial receiver pressure, psig
P_2 = Final receiver pressure, psig

Figure 4.27 Air receiver sizing formula

Figure 4.28 presents a situation for determining the proper size of a dedicated receiver using the air receiver formula from Figure 4.26. The object is to size a dedicated receiver so that when a reoccurring event of 62 cfm that lasts for 15 seconds happens, the pressure in the receiver does not drop below 85 psig. Eighty-five (85) is the minimum pressure at which the application will work reliably (critical pressure). Once the event happens, the receiver drops to 85 psig in 15 seconds, and then the receiver slowly refills back to general storage pressure of 100 psig. Space in the plant is limited, so the customer wants the smallest receiver that will satisfy the conditions.



Conditions:
- Receiver starting pressure: 100 psig
- Event flow rate: 62 cfm
- Event duration: 15 seconds
- Event frequency: every 3 minutes
- Critical pressure: 85 psig
- Location is at sea level

Problem
What is the smallest size receiver in ft^3 that will maintain receiver pressure at or above 85 psi during the event

Figure 4.28 Dedicated receiver sizing example

The air receiver formula will be used to solve for the required receiver volume. The following information is provided:

1. The flow requirement of the event is 62 cfm.
2. The 62-cfm event lasts for 15 seconds, which is one quarter of a minute.
3. The receiver is at 100 psig before the event starts and that at no time during the event can receiver pressure fall below 85 psig, the critical pressure.
4. 100 psig is P1 and 85 psig is P2.
5. Since the plant is about 300 feet above sea level, the atmospheric pressure is 14.5 psia.
6. The event repeats in three minutes, so there is sufficient time to refill the receiver through a properly sized metering valve before the event reoccurs.

Solving the equation is simply left to inserting the values into the right positions and performing math.

$$V? = \frac{.25 \text{ x } 62 \text{ x } 14.5}{15} = 14.98 \, ft^3$$

Figure 4.29 Dedicated receiver sizing example solution

Figure 4.29 shows the equation with all of the values placed in their proper positions. The units have been left out for simplicity, but they will cancel, leaving the final volume value expressed in cubic feet. Since in North America compressed air receivers are sold in gallons, not in cubic feet, the cubic foot requirement needs to be converted into gallons. This is done by using the conversion factor of 1 cubic foot equals 7.48 gallons. This converts the 14.98 cubic foot requirement into ~112 gallons. A 112-gallon receiver is not a standard receiver size and getting one made is possible, but very costly. As shown in Table 4.16, the 120-gallon is the next standard size available so this is the receiver size to use. Since the 120-gallon receiver is larger than the minimum requirement of 112 gallons, this means that the receiver pressure will never reach the critical pressure of 85 psig during the event, a nice cushion for error or change.

Table 4.16 Standard Air Receiver Sizes

Gallons	Cubic Feet	Working Pressure psig
30	4.01	250
60	8.02	250
80	10.7	250
120	16.04	200

Air receivers should meet the Code for Unfired Pressure Vessels published by the American Society of Mechanical Engineers (ASME). ASME coded vessels will also have an approved safety valve, pressure gauge, drain port, and hand holes for manhole covers. This normally is a requirement of insurance companies. In addition, all receivers must meet all federal, state and local codes and/or laws that might apply to the specific application or location of the receiver.

Chapter 4