Compressed Air Drying

ENGINEERING YOUR SUCCESS.

Together, we can separate the good from the bad.

Compressed Air and its Purification from Generation to Application

Compressed air is an essential power source that is widely used throughout industry. This safe, powerful and reliable utility can be the most important part of your production process. However, your compressed air will contain water, dirt, wear particles, and even degraded lubricating oil which all mix together to form an unwanted condensate. This condensate, often acidic, rapidly wears tools and pneumatic machinery, blocks valves and orifices causing high maintenance and costly air leaks. It also corrodes piping systems and can bring your production process to an extremely expensive standstill!

The quality of air required throughout a typical compressed air system can vary.

It is highly recommended that the compressed air is treated prior to entry into the distribution system, as well as at each usage point or application.

This approach to system design provides the most cost effective solution to system purification, as it not only removes the contamination already in the distribution system, it ensures that only the most critical areas receive air treated to the highest level.

In many instances the compressed air system will be supplying air to more than one application and, although the purification equipment specified in the compressor room would remain unchanged, the point-of-use protection will vary depending upon the air quality requirements of each application.

In many cases this action alone is not enough, as modern production systems and processes demand an even higher level of air quality. Where required, "point-of-use" filtration and air dryers can provide the correct air quality.

Sources of Contamination Found in a Compressed Air System

Contaminants in a compressed air system can generally be attributed to the following:

The quality of air being drawn into the compressor

Air compressors draw in a large volume of air from the surrounding atmosphere containing large numbers of airborne contaminants.

The type and operation of the air compressor
The air compressor itself can also add contamination, from wear particles to coolants and lubricants.

Compressed air storage devices and distribution systems
The air receiver and system piping are designed to store and distribute the compressed air. As a consequence, they will also store the large amounts of contaminants drawn into the system. Additionally, piping and air receivers will also cool the moist compressed air, forming condensate, which causes damage and corrosion.

Types of Contamination Found in a Compressed Air System

Atmospheric Dirt
Atmospheric air in an industrial environment typically contains 140 million dirt particles for every cubic meter of air. Eighty percent of these particles are less than 2 micron in size and are too small to be captured by the compressor intake filter, therefore passing directly into the compressed air system.

Water Vapor, Condensed Water and Water Aerosols
Atmospheric air contains water vapor (water in a gaseous form). The ability of compressed air to hold water vapor is dependent upon its temperature. The higher the temperature, the more water vapor that can be held by the air. During compression, the air temperature is increased significantly, which allows it to easily retain the incoming moisture. After the compression stage, air is normally cooled to a usable temperature. This reduces the airs ability to retain water vapor, resulting in a proportion of the water vapor being condensed into liquid water, which is removed by a condensate drain fitted to the compressor after-cooler. The air leaving the after-cooler is now 100% saturated with water vapor and any further cooling of the air will result in more water vapor condensing into liquid water. Condensation occurs at various stages throughout the system as the air is cooled further by the air receiver, piping and the expansion of valves, cylinders,

tools and machinery. The condensed water and water aerosols cause corrosion to the storage and distribution system, damage production equipment, and the end product. It also reduces production efficiency and increases maintenance costs. Water in any form must be removed to enable the system to run correctly and efficiently.

Rust and Pipescale

Rust and pipescale can be found in air receivers and the piping of “wet systems” (systems without adequate purification equipment) or systems which were operated “wet” prior to purification being installed. Over time, this contamination breaks away to cause damage or blockage in production which can also contaminate final product and processes.

Micro-organisms

Bacteria and viruses will also be drawn into the compressed air system through the compressor intake. Warm, moist air provides an ideal environment for the growth of micro-organisms. Ambient air can typically contain up to 3,850 micro-organisms per cubic meter. If only a few micro-organisms were to enter a clean environment, a sterile process or production system, enormous damage could be caused that not only diminishes product quality, but may even render a product entirely unfit for use and subject to recall.

Liquid Oil and Oil Aerosols

Most air compressors use oil in the compression stage for sealing, lubrication and cooling. During operation, lubricating oil is carried over into the compressed air system as liquid oil and aerosols. This oil mixes with water vapor in the air and is often very acidic, causing damage to the compressed air storage and distribution system, production equipment, and final product.

Oil Vapor

In addition to dirt and water vapor, atmospheric air also contains oil in the form of unburned hydrocarbons. The unburned hydrocarbons drawn into the compressor intake, as well as vaporized oil from the compression stage of a lubricated compressor, will carry over into a compressed air system where it can cool and condense, causing the same contamination issues as liquid oil. Typical oil vapor concentrations can vary between 0.05 and 0.5mg per cubic meter of air.

Compressed Air Quality Standards – ISO 8573

ISO 8573 is the group of International Standards relating to the quality of compressed air and consists of nine separate parts. Part 1 specifies the quality requirements of the compressed air and parts 2 - 9 specify the methods of testing for a range of contaminants.

ISO 8573.1 : 2001 is the primary document used from the ISO 8573 series and allows the user to specify the air quality or purity required at key points in a compressed air system. Within ISO 8573.1 : 2001, purity levels for the main contaminants are shown in separate tables, however for ease of use, this document combines all three into one easy to understand table.

Purity Class	Solid Particulate					Water		Oil
	Maximum number of particles per m^3			Particle Size	Concentration	Vapor	Liquid	Total oil (aerosol, liquid and vapor)
	0.1 - 0.5 micron	0.5 - 1 micron	1 - 5 micron	micron	mg/m^3	Pressure Dewpoint	g/m^3	ppm (mg/m^3)
0	*	*	*	*	*	*	*	*
1	100	1	0	—	—	-94°F (-70°C)	—	0.008 (0.01)
2	100,000	1,000	10	—	—	-40°F (-40°C)	—	0.08 (0.1)
3	—	10,000	500	—	—	-4°F (-20°C)	—	0.83 (1)
4	—	—	1,000	—	—	37°F (3°C)	—	4.2 (5)
5	—	—	20,000	—	—	45°F (7°C)	—	—
6	—	—	—	5	5	50F (10°C)	—	—
7	—	—	—	40	10	—	0.5	—
8	—	—	—	—	—	—	5	—
9	—	—	—	—	—	—	10	—

* As specified by the equipment user or supplier.

Specifying Air Purity in Accordance with ISO 8573.1 : 2001

When specifying the purity of air required, the standard must always be referenced, followed by the purity class selected for each contaminant (a different purity class can be selected for each contaminant if required). An example of how to write an air quality specification is shown at right :

Example:

ISO 8573.1 : 2001 Class 2.2.2

ISO8573.1 : 2001 refers to the standard document and its revision. The three digits refer to the purity classifications selected for solid particulate, water and total oil. Selecting an air purity class of 2.2.2 would specify the following air quality when operating at the standard's reference conditions:

Class 2, Particulate

In each cubic meter of compressed air, not more than 100,000 particles in the 0.1 to 0.5 micron size range are allowed in each cubic meter of compressed air. No more than 1,000 particles in the 0.5 to 1.0 micron size range are allowed in each cubic meter of compressed air. Ten particles in the 1.0 to 5.0 micron size range are allowed.

Class 2, Water

A pressure dewpoint of -40°F (-40°C) or better is required and no liquid water is allowed.

Class 2, Oil

In each cubic meter of compressed air, not more than 0.1mg of oil is allowed. This is a combined level for both aerosol and oil vapor.

Cost Effective System Design

To achieve the stringent air quality levels required for today's modern production facilities, a careful approach to system design, commissioning and operation must be employed. Treatment at one point alone is not enough. It is highly recommended that the compressed air is treated prior to entry into the distribution system, to a quality level suitable for protecting air receivers and distribution piping.

Point-of-use purification should also be employed, with specific attention being focused on the application and the level of air quality required. This approach to system design ensures that air is not “over treated” and provides the most cost effective solution to high quality compressed air.

How Water Gets Into the Air System

Compressed air has become an indispensable source of energy in modern industrial processes.

All atmospheric air contains a certain quantity of water vapor which is mixed with other gases, such as nitrogen, oxygen and carbon monoxide. This water vapor is drawn into the air compressor with the incoming air during the compression cycle.

Water is present in the air which is drawn into the compressor. The water is gaseous – invisible and completely mixed with the air. The exact amount of water is called the "humidity" of the air.

a) **Relative Humidity** – The amount of water vapor that can be held in air is dictated by the temperature of the air. Hot air can hold more water (as vapor) than cold air. Typically, atmospheric air contains approximately 50% of its water vapor holding capacity for a given temperature. This proportion of the maximum vapor holding capacity is referred to as **Relative Humidity.**

b) **Dewpoint and Condensation** – When air with a given relativity humidity is cooled, it reaches a temperature at which it is saturated. At saturation, the relative humidity is 100%, i.e., the air contains as much water vapor as it can hold. The temperature at which the air is at 100% relative humidity is known as the **dewpoint** of the air. Cooling air beyond the dewpoint results in **condensation** of the water vapor.

c) **Cooling and Condensation in Compressed Air** – The following table details the changes in 8 cubic feet of air as it is compressed to 100PSIG and subsequently cooled in an aftercooler. Worthy of note is the effect of the air temperature rise as the air is compressed. The increased temperature of the compressed air increases its vapor holding capacity which, in turn, reduces the relative humidity of the air because the actual water vapor content (74g) has remained constant. We must also note, however, that compressing the air has also increased the dewpoint of the air. This means that subsequent cooling of the air (by an aftercooler or as a result of a cooler ambient temperature) could cause condensation. Using an aftercooler, as shown, can remove a significant proportion of the water vapor (75%, as shown in the table) from the air through the principle of condensation. When leaving the aftercooler, the compressed air is saturated - any further cooling of the air will result in condensation. It is this cooling beyond the dewpoint of the compressed air which causes the water that end users see in their compressed air supplies.

Table 1, Compressing Air

	Intake	Outlet	Aftercooler
Volume	8 cu. ft.	1 cu. ft.	1 cu.ft.
Pressure (Gauge)	0 PSIG	100 PSIG	100 PSIG
Temperature (Example)	68°F (20°C)	158°F (70°C)	68°F (20°C)
Water Content (Vapor)	2.1g	2.1g	0.6g
Relative Humidity	50%	30%	100%
Dew Point (At pressure shown)	50°F (10°C)	97°F (36°C)	68°F (20°C)

d) Sources of Cooling – There are many ways to cool saturated compressed air:

- Ambient Conditions – Expose compressed air lines to cooler outdoor temperatures
  – Expose compressed air lines to unheated rooms
- Pressure Reduction – Pressure regulators, vortex tubes, expansion vessels, and receiving tanks
- Process Equipment – Aftercoolers, Dryers, etc.

The water vapor becomes a major hazard in compressed air systems, given that it is distributed together with the compressed air itself. As the compressed air is cooled while passing through the plants air piping, this water vapor will condense.

How and why does this occur? Compressed air, at normal ambient temperatures, cannot hold as much water vapor as air at atmospheric pressure. However, the heat generated during the compression cycle increases its ability to hold water vapor. When the compressed air is cooled between the compressor and the point of use, this water vapor will condense and become liquid water, depositing itself in the system piping, air receiver, tools, etc. The quantity of water vapor condensed will be that amount which is in excess of the saturated temperature of the compressed air. This condensed water will corrode system components resulting in increased maintenance and reduced system efficiency.

How Much Water Can Be Found In A Typical Compressed Air System?

The amount of water in a compressed air system is staggering. A small 100 cfm (2.8m^3/min) compressor and refrigeration dryer combination, operating for 4,000 hours in typical climatic conditions can produce approximately 10,000 liters or 2,200 gallons of liquid condensate per year.

If the compressor is oil lubricated with a typical 2ppm (2 mg/m^3) oil carryover, oil would in fact account for less than 0.1% of the overall volume. Although, the resulting condensate can falsely resemble oil.

The example above assumes using a small compressor to highlight the large volume of condensate produced. If a compressed air system was operated in warmer, more humid climates, or with larger compressors installed (running for longer periods) the volume of condensate would increase significantly.

How is Water Removed from the Air System?

Getting the Water Out

Usually, compressed air contains water in both the liquid and vapor phases. "Drying" can range from trapping the condensed water, to preventing additional condensation of water vapor, to removing virtually all the water present. The more water removed, the higher the cost of drying.

However, if too much water is permitted to remain in the compressed air supply, the price is paid in maintenance costs, corrosion, and/or product losses. These costs support the importance of specifying the proper drying technology for a given application.

Drying Methods Available

The following list is a summary of the drying technologies available:

- Aftercooler – Reduces the temperature and water content of the compressed air.
- Bulk Liquid Separators – Remove bulk liquid condensed in the distribution system.
- Particulate Filters – Remove solid particle contaminants down to 5 micron and the separation of bulk liquids.
- Coalescing Filters – Remove liquid aerosol and submicron particles (not vapor) down to .01 micron in size.
- Refrigeration Dryers – Drying to dewpoints of approximately 37°F to 50°F (3°C to 10°C)
- Desiccant Dryers – Drying to dewpoints of approximately -40°F to -100°F (-40°C to -73°C).
- Membrane Dryers – Variable drying capabilities as low as -40°F to 35°F (-40°C to 2°C) dewpoint depending on flow.

At the Compressor

The standard compressor installation consists of a compressor, an aftercooler (water cooled or air cooled), and a receiver. In a system with an efficient aftercooler, the distance from the receiver to the filter is not important. Since the filter is usually maintained by the personnel responsible for the compressor, it is often convenient to install the filter immediately after the receiver.

Some compressor installations do not have an aftercooler, This is **not** a recommended situation. Air saturated with water vapor leaves the compressor at temperatures between 230°F and 392°F (110°C and 200°C) and cools to approach room temperature in the distribution lines. Although water will condense throughout the air distribution system, about two-thirds of the total water content of the air will be condensed when the air has cooled to 104°F 40°C).

Therefore, to remove most of the water load from the system, a mainline filter must be installed just prior to the first distribution line manifold. However, since the air will continue to cool in the distribution system, additional filters located at end-use points will be required to remove water condensed downstream from the main line filter.

Aftercoolers

An efficient aftercooler is essential to all compressed air systems and will condense up to 75% of the water vapor. For example, if air enters a 3500 SCFH compressor at 68°F (20°C) and exits at 100 PSIG and 248°F (120°C), it will release about 13 gallons (67 liters) of condensed water per day into the air distribution system while cooling down to 68°F (20°C). In the absence of an aftercooler, installing coalescing filters at various points in the system will remove much of the condensate, but if the air temperature at any filter is higher than room temperature, water will condense downstream from the filter as soon as the air cools a few more degrees. The only way to prevent condensation of the water throughout the system is to install an efficient aftercooler immediately after the compressor, and an efficient coalescing filtration system (with automatic drains) downstream from the aftercooler. Water may still condense downstream from the filter if the aftercooler has not reduced the air temperature to room temperature. This relatively small quantity of condensate can be eliminated by the simple technique of pressure reduction.

Bulk Liquid Separators (High Efficiency Water Separators)

These are used to remove bulk condensed liquids after the aftercooler, receiver, or anywhere within the distribution system. Bulk liquid separators also help protect filters in systems where excessive cooling takes place. They remove in excess of 98% of bulk liquid contamination through centrifugal separation techniques.

Particulate Filters

Particulate filters are used for the removal of solid particle contaminants down to 5 micron and the separation of bulk liquids. Note that water vapor, in vapor form, passes through general purpose particulate filters.

This type of filter is generally used in industrial applications, and should be used as a prefilter for the coalescing (oil removal) filter.

Coalescing Filters

Coalescing filters are essential to remove compressor lubricant, water droplets, and particles from the compressed air supply. Coalescing filters are designed to remove only liquids and particulate (not vapors) from a compressed gas stream down to 0.01 micron in size.

A moderately efficient coalescing filter (1.0 micron in size) is used for most air coalescing applications where the removal of liquid aerosols and submicronic particles for general air quality is required. This grade of filter element should be used as a prefilter for the high efficiency coalescing filter.

A high efficiency coalescing filter (down to 0.01 micron in size) is used where the removal of extremely fine particles and virtually "oil free", or high quality air, is necessary.

Installed in pairs, this dual filter installation ensures a continuous supply of high quality compressed air. At the point where the air is used, it ensures that any liquid condensed in the distribution system will be removed, as long as no further cooling occurs. The compressed air delivered after coalescing filtration will be free of liquids, but could be relatively high in water vapor content.

Refrigeration Dryers

As the name implies, refrigerated dryers work by cooling the air to low temperatures; thus condensing much of the water vapor. It is not possible to achieve dewpoints below freezing with this type of dryer. Ideal for general purpose applications, they typically provide pressure dewpoints of 38°F (3°C), 45°F (7°C) , or 50°F (10°C) pdp.

Refrigeration dryers remove the heat from the inlet air and use it to reheat the air at the outlet. Dried air is returned to the air line at reasonable temperatures. The advantages of heating the outlet air are clear: this process eliminates condensation, "sweating", from occurring when exposed to cold pipes, or in humid conditions. Self-contained refrigerant dryers use fans to cool the refrigerant condenser and automatic cooling systems to provide the exact heat exchange required by the air being used. These systems keep the delivered air at a constant humidity or dewpoint. Coalescing filters upstream are required to prevent oil/liquid water from entering the dryer. Oil coating the cooling surfaces causes loss of efficiency and liquid water absorbs some of the system capacity.

Refrigeration dryers are not suitable for installations where piping is installed in ambient temperatures below the dryer dewpoint, i.e. systems with external piping.

Adsorption (Desiccant) Dryers

Adsorption dryers are used in those applications where very dry air is required; they are generally either installed downstream of the aftercooler and/or the refrigeration dryer.

Inline Adsorption Dryers - feature a desiccant material contained within a vessel; the compressed air passes through the vessel and across the desiccant bed, and the water vapor is absorbed by the desiccant material. The air exits the adsorption dryer in a very dry state; the dewpoint achieved varies according to the specific application, but typically the level is -40°F (-40°C) or -100°F (-70°C).

Heated Regenerative Desiccant Dryers - use heat to remove water vapor from the desiccant material in the dryer bed not in use at that point in the cycle. In heated desiccant dryers, heat is applied for typically 75% of the cycle, and the bed is allowed to cool for the remaining 25% of the cycle. A great deal of steam or electricity is required to operate heated desiccant dryers.

Heatless Regenerative Desiccant Dryers - use the dry air generated by the desiccant dryer to remove water vapor from the desiccant material. The dry air is passed over the desiccant bed (not in use) and water vapor evaporates from the desiccant into the dry air system. This moisture laden air is subsequently vented to the atmosphere. The major advantage to using heatless desiccant dryers is the reduced dependence on expensive utilities - namely steam, electricity, or other heat sources. Minimal electricity is required to run a heatless desiccant dryer.

Regenerative desiccant dryers can be conveniently located near the point-of-use to deliver dry compressed air at dewpoints to -100°F (-70°C). These dryers are wall mountable and ideal for delivering instrument-quality air for critical applications.

Note that the actual air temperature after an adsorption dryer is not the same as its dewpoint. Beneficially, a pressure dewpoint of -15°F (-26°C) or better will not only prevent corrosion, but will also inhibit the growth of microorganisms within the compressed air system.

Overall, heatless desiccant dryers have an advantage over heated desiccant dryers in that they do not require excessive outside services, i.e., steam, electricity, or gas for heat, to generate dry air and regenerate the desiccant. In addition to reducing dependency on outside services, costs for operating these dryers are also reduced.

Desiccant dryers should be protected from liquid water by a coalescing filter installed upstream from the dryer. Oil or water entering the dryer will adversely affect the performance of the dryer and/or destroy the desiccant material. It is also good practice to install a filter downstream from the dryer to prevent any carryover of the desiccant to downstream equipment or processes.

Membrane Air Dryers

Membrane materials selectively permeable to water vapor are an excellent medium for producing dry air from standard compressed air. The water vapor in the compressed air is removed by the principle of selective permeation through a membrane. The membrane module consists of bundles of hollow membrane fibers, each permeable to water vapor. As the compressed air passes through the center of these fibres, water vapor permeates through the walls of the fiber. A small portion of the dry air (purge flow) is redirected along the outside of each hollow fiber, carrying away the moisture-laden air which is then exhausted to room atmosphere. The remainder of the dry air is piped to the application.

Membrane dryers can be conveniently located near the point-of-use and can supply clean dry compressed air with dewpoints as low as -40°F to 35°F (-40°C to 2°C) depending on flow.

Coalescing filters should be installed upstream from a membrane dryer to protect the membrane from being saturated by water or coated by oil. If saturation or coating occurs, the membrane drying function could be seriously inhibited.

Important Note Regarding Compressed Air Dryers

As refrigeration, adsorption and membrane dryers are designed to remove only water vapor and not water in a liquid form, they require the use of coalescing filters and possibly a bulk liquid separator to work efficiently.

Specifying the Right Dryer

Several factors should be considered when specifying a dryer. Dew point requirements, inlet air temperature, ambient air temperature, operating pressure, air flow requirements, and available utilities.

In specifying the right dryer for a compressed air installation, also keep the following information in mind as a guideline.

1. Dryness is relative - For general plant air the compressed air's dew point should be about 20°F (-7°C) lower than the lowest ambient air temperature to avoid freezing.
2. Inlet temperature is a key factor - A 20°F (-7°C) reduction in temperature condensates 50% of the humidity out of the air.

3 Do not overspecify - Drying the entire compressed air supply in a factory to dewpoints less than -40°F (-40°C) is wasteful. It is more sensible to subdivide the compressed air supply by application, treating each end use point as needed to provide appropriately dry air for the downstream application served.

4 Do not underspecify - Damage caused by wet air costs money in maintenance time and supplies, downtime, and lost product. Design a drying system to meet specific needs.

5 Avoid potential problems - A drying system which only contains an aftercooler and a coalescing filter could create problems with condensation downstream from the aftercooler. The air is still saturated with water vapor which is likely to condense if the ambient temperature is lower than the compressed air temperature.

6 Refrigeration dryers - Ideal for general purpose applications. This type of dryer does not achieve dew points below freezing.

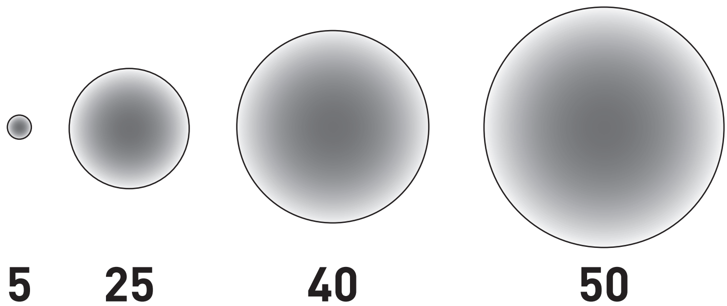
7 Desiccant dryers - Ideal for applications where very dry air -40°F (-40°C) or below is required. Two main types: heated and heatless. General rule of thumb is applications below 1000 SCFM (28.3m^3 / min.) work best with heatless and applications above 1000 SCFM (28.3m^3 / min.) work best with heated.

8 Membrane dryers - Ideal for applications where dry air down to -40°F (-40°C) is required. No electricity required and, with proper pre-filtration and post-filtration, is maintenance free.

Summary

Protect your equipment, save money and optimize your energy consumption by improving air purity with the installation air drying products. Work with a supplier who understands the processing requirements. Select properly sized equipment with appropriate dew point and flow in order to preserve assets, eliminate headaches, avoid downtime and increase value. Assure that your system provides clean dry air.

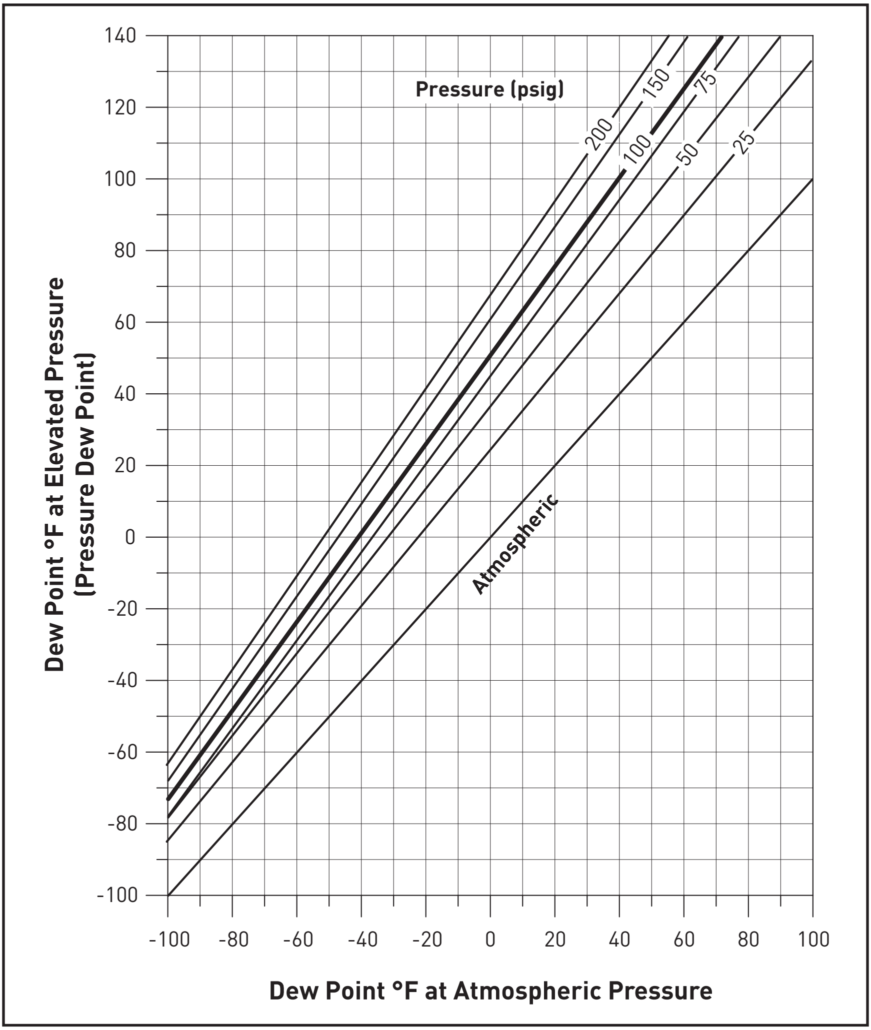
Relative Particle Size
(Micron)



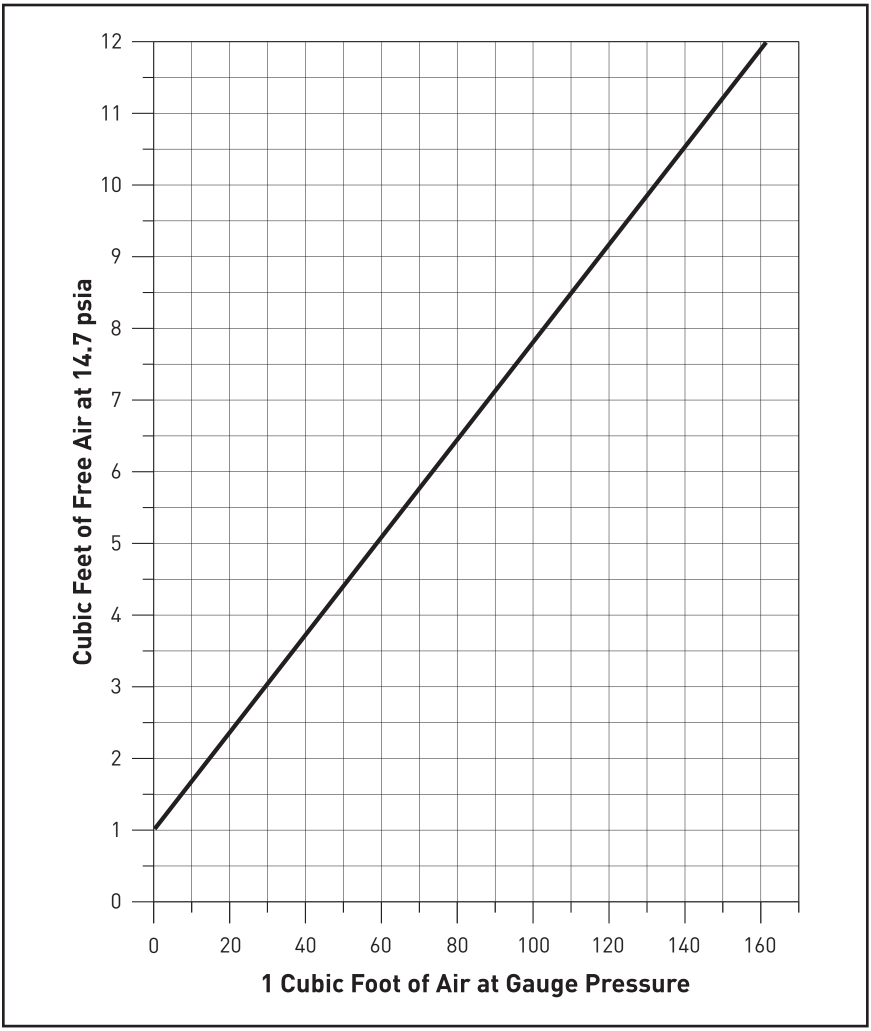
Sizes of Familiar Objects

Grain of Table Salt	100	.0039
Human Hair	70	.0027
Lower Limit of Visibility	40	.00158
White Blood Cells	25	.00039
Talcum Powder	10	.00039
Red Blood Cells	8	.0003
Bacteria (Average)	1	.000039

Dew Point Conversion Chart



Ratio of Compression



The above is based on the following equation:

$$\frac{\text{Absolute Pressure (PSIA)}}{\text{Atmospheric Pressure}}$$

Note: Absolute pressure is the sum of gauge pressure and atmospheric pressure.

Discharge of Air (SCFM) Through an Orifice

Size of Orifice (In.)	Pressure (psig) Across Orifice																		
	2	**5**	**10**	**15**	**20**	**25**	**30**	**35**	**40**	**45**	**50**	**60**	**70**	**80**	**90**	**100**	**125**	**150**	**200**
1/64"	0.024	0.04	0.061	0.079	0.096	0.113	0.129	0.145	0.161	0.177	0.193	0.225	0.256	0.287	0.319	0.35	0.428	0.506	0.661
1/32"	0.098	0.161	0.243	0.319	0.384	0.451	0.516	0.581	0.645	0.709	0.772	0.898	1.02	1.15	1.27	1.4	1.71	2.02	2.64
3/64"	0.22	0.363	0.547	0.71	0.86	1.01	1.16	1.31	1.45	1.59	1.74	2.02	2.3	2.59	2.87	3.15	3.85	4.55	5.95
1/16"	0.391	0.645	0.972	1.26	1.54	1.8	2.07	2.32	2.58	2.84	3.09	3.59	4.1	4.6	5.1	5.6	6.85	8.09	10.6
3/32"	0.881	1.45	2.19	2.84	3.46	4.06	4.65	5.23	5.81	6.38	6.95	8.09	9.22	10.3	11.5	12.6	15.4	18.2	23.8
1/8"	1.57	2.58	3.89	5.05	6.15	7.21	10.1	11.3	12.4	13.5	14.7	16.9	19.2	21.4	23.7	26	31.6	37.5	49
3/16"	3.52	5.8	8.75	11.4	13.8	16.2	18.6	20.9	23.2	25.5	27.8	32.3	36.9	41.4	45.9	50.4	61.6	72.8	95.2
1/4"	6.26	10.3	15.6	20.2	24.6	28.9	40.5	45	49.6	54.1	58.6	67.6	76.7	85.7	94.8	104	110	150	196
3/8"	14.1	23.2	35	45.4	55.3	64.9	74.4	83.7	92.9	102	111	129	147	166	184	202	246	291	381
1/2"	25.1	41.3	62.2	80.8	98.4	115	132	149	165	181	198	230	262	294	326	358	438	518	677
5/8"	39.1	64.5	97.2	126	154	180	207	232	258	284	309	359	410	460	510	560	685	809	1058
3/4"	56.4	92.9	140	182	221	260	297	335	372	408	445	518	590	662	734	806	986	1165	1523
7/8"	76.7	126	191	247	301	354	405	455	506	556	605	704	803	901	999	1097	1342	1586	2073
1"	100	165	249	323	393	462	529	595	661	726	791	920	1049	1177	1305	1433	1752	2071	
1-1/8"	127	209	315	409	498	584	669	753	836	919	1001	1164	1327	1490	1652	1814	2218		
1-1/4"	157	258	389	505	615	721	826	930	1032	1134	1236	1438	1639	1839	2039	2239			
1-3/8"	189	312	471	611	744	893	1000	1125	1249	1372	1495	1739	1983	2226	2468				
1-1/2"	225	371	560	727	885	1039	1190	1339	1486	1633	1779	2070	2360						
1-3/4"	307	506	762	990	1205	1414	1619	1822	2023	2223	2422								
2"	401	660	996	1293	1574	1847	2115	2380											

Vacuum Flow (SCFM) Through an Orifice

Orifice Diameter (In)"	Vacuum in Inches of Mercury (Hg.) Across Orifice								
	2"	**4"**	**6"**	**8"**	**10"**	**12"**	**14"**	**18"**	**24"**
1/64"	0.018	0.026	0.032	0.037	0.041	0.045	0.048	0.055	0.063
1/32"	0.074	0.1	0.128	0.148	0.165	0.18	0.195	0.22	0.25
1/16"	0.3	0.42	0.517	0.595	0.66	0.725	0.78	0.88	1
1/8"	1.2	1.68	2.06	2.37	2.64	2.89	3.12	3.53	4.04
1/4"	4.78	6.74	8.25	9.52	10.6	11.6	12.4	14	16.2
3/8"	10.8	15.2	18.5	21.4	23.8	26	28	31.8	36.4
1/2"	19.1	27	33	38.5	42.3	46.3	50	56.5	64.6
5/8"	30	42.2	51.7	59.5	66.2	72.6	78	88	101
3/4"	43	60.6	74	85.3	95.2	104	112	127	145
7/8"	58.8	82.6	101	116	130	142	153	173	198
1"	76.5	108	131	152	169	185	200	225	258

Atmospheric Pressure and Barometer Readings at Different Altitudes

Altitude Above Sea Level, Feet	Atmospheric Pressure Lbs. per Square Inch	Barometer Reading Inches of Mercury
0	14.69	29.92
500	14.42	29.38
1,000	14.16	28.86
1,500	13.91	28.33
2,000	13.66	27.82
2,500	13.41	27.31
3,000	13.16	26.81
3,500	12.92	26.32
4,000	12.68	25.84
4,500	12.45	25.36
5,000	12.22	24.89
5.500	11.99	24.43
6,000	11.77	23.98
6,500	11.55	23.53
7,000	11.33	23.09
7,500	11.12	22.65
8,000	10.91	22.22
8,500	10.70	21.80
9,000	10.50	21.38
9,500	10.30	20.98
10,000	10.10	20.58

Weight of Water in a Cubic Foot of Air at Various Temperatures and Percentages of Saturation

Weights shown in grains. 7,000 grains = 1 lb.											
Temp	Relative Humidity										Temp
°C	10%	20%	30%	40%	50%	60%	70%	80%	90%	100%	°F
-23	.028	.057	.086	.114	.142	.171	.200	.228	.256	.285	-10
-18	.048	.096	.144	.192	.240	.289	.337	.385	.433	.481	0
-12	.078	.155	.233	.310	.388	.466	.543	.621	.698	.776	10
-6.7	.124	.247	.370	.494	.618	.741	.864	.988	1.11	1.24	20
0	.211	.422	.634	.845	1.06	1.27	1.48	1.69	1.90	2.11	32
1.6	.237	.473	.710	.946	1.18	1.42	1.66	1.89	2.13	2.37	35
4.4	.285	.570	.855	1.14	1.42	1.71	1.99	2.28	2.56	2.85	40
7.2	.341	.683	1.02	1.37	1.71	2.05	2.39	2.73	3.07	3.41	45
10	.408	.815	1.22	1.63	2.04	2.45	2.85	3.26	3.67	4.08	50
12.7	.485	.970	1.46	1.94	2.42	2.91	3.39	3.88	4.36	4.85	55
15.6	.574	1.15	1.72	2.30	2.87	3.45	4.02	4.60	5.17	5.75	60
18.4	.678	1.36	2.03	2.71	3.39	4.07	4.75	5.42	6.10	6.78	65
21	.798	1.60	2.39	3.19	3.99	4.79	5.59	6.38	7.18	7.98	70
24.9	.936	1.87	2.81	3.74	4.68	5.62	6.55	7.49	8.42	9.36	75
26.7	1.09	2.19	3.28	4.37	5.47	6.56	7.65	8.75	9.84	10.93	80
29.5	1.27	2.54	3.81	5.08	6.35	7.62	8.89	10.16	11.43	12.73	85
32.2	1.48	2.96	4.44	5.92	7.40	8.87	10.35	11.83	13.31	14.78	90
35	1.72	3.44	5.16	6.88	8.60	10.32	12.04	13.76	15.48	17.15	95
37.8	1.98	3.95	5.93	7.91	9.88	11.86	13.84	15.81	17.79	19.77	100
43.3	2.63	5.26	7.89	10.52	13.15	15.78	18.41	21.04	23.67	26.33	110
48.9	3.45	6.90	10.35	13.80	17.25	20.70	24.15	27.60	31.05	34.48	120
54.4	4.44	8.88	13.32	17.76	22.20	26.64	31.08	35.52	39.96	44.42	130

SATURATION COLUMN

HP to Compress Air

These tables show the approximate HP required to compress 1 SCFM (standard cubic foot per minute) of air from atmospheric pressure of 0 PSIG to the pressure shown in the tables. Inlet air is assumed to be about room temperature.

The tables were prepared from information published in Machinery's Handbook. Please refer to your copy of the Handbook for additional information on the compression of air.

Use the chart below for your kind of compressor. Calculate the HP to compress 1 SCFM of air to the regulator inlet pressure. Then calculate the HP to compress 1 SCFM to the regulator outlet or reduced pressure. Subtract the two. This will show the HP wasted for every 1 SCFM passing through the regulator. Multiply times the SCFM air flow through the circuit.

Horsepower for Compressing Air
Efficiency of All Compressors is Assumed to be 85%

1-Stage Compressor		***2-Stage Compressor***		***3-Stage Compressor***	
PSIG	**HP***	**PSIG**	**HP***	**PSIG**	**HP***
5	.021	50	.116	100	.159
10	.040	60	.128	150	.190
15	.056	70	.138	200	.212
20	.067	80	.148	250	.230
25	.079	90	.156	300	.245
30	.095	100	.164	350	.258
35	.099	110	.171	400	.269
40	.107	120	.178	450	.279
45	.116	130	.185	500	.289
50	.123	140	.190	550	.297
55	.130	150	.196	600	.305
60	.136	160	.201	650	.311
65	.143	170	.206	700	.317
70	.148	180	.211	750	.323
75	.155	190	.216	800	.329
80	.160	200	.220	850	.335
85	.166	210	.224	900	.340
90	.170	220	.228	950	.345
95	.175	230	.232	1000	.350
100	.179	240	.236	1050	.354
110	.188	250	.239	1100	.358
120	.196	260	.243	1150	.362
130	.204	270	.246	1200	.366
140	.211	280	.250	1250	.370
150	.218	290	.253	1300	.374
160	.225	300	.255	1350	.378
170	.232	350	.269	1400	.380
180	.239	400	.282	1450	.383
190	.244	450	.293	1500	.386
200	.250	500	.303	1550	.390

*HP to compress 1 SCFM from 0 PSIG to the values shown.

Note: The power required from other types of compressors of the same number of stages will be related to these values as the efficiency of the other compressor is to the assumed 85% efficiency used for these tables.

Maximum Recommended Air Flow (SCFM) Through ANSI Standard Weight Schedule 40 Metal Pipe

The flow values in the table below are based on a pressure drop of 10% of the applied pressure per 100 feet of pipe for 1/8", 1/4", 3/8", and 1/2" pipe sizes; and a pressure drop of 5% of the applied pressure per 100 feet of pipe for 3/4", 1", 1-1/4", 2", 2-1/2", and 3" pipe sizes. The table gives recommended flows for pipe sizes at listed pressures and should be used to determine appropriate piping for air systems.

Applied Pressure **PSI**	**Nominal Standard Pipe Sizes**										
	1/8"	***1/4"***	***3/8"***	***1/2"***	***3/4"***	***1"***	***1-1/4"***	***1-1/2"***	***2"***	***2-1/2"***	***3"***
5	0.5	1.2	2.7	4.9	6.6	13	27	40	80	135	240
10	0.8	1.7	3.9	7.7	11.0	21	44	64	125	200	370
20	1.3	3.0	6.6	13.0	18.5	35	75	110	215	350	600
40	2.5	5.5	12.0	23.0	34.0	62	135	200	385	640	1100
60	3.5	8.0	18.0	34.0	50.0	93	195	290	560	900	1600
80	4.7	10.5	23.0	44.0	65.0	120	255	380	720	1200	2100
100	5.8	13.0	29.0	54.0	80.0	150	315	470	900	1450	2600
150	8.6	20.0	41.0	80.0	115.0	220	460	680	1350	2200	3900
200	11.5	26.0	58.0	108.0	155.0	290	620	910	1750	2800	5000
250	14.5	33.0	73.0	135.0	200.0	370	770	1150	2200	3500	6100

Gas Laws

$$PV = \frac{MRT}{144} \quad \text{where}$$

P = absolute pressure psi
V = volume, ft^3
M = weight of air in pounds
T = absolute temperature ºR (Rankine)
R = universal gas constant (air = 53.3 psf)

Density Of Water Vapor

$$dv = \frac{(Pv)(144)(1)}{(85.78)(T)}$$

dv = pounds of water vapor/ft
Pv = water vapor pressure psi @ dew point temperature
85.78 = water vapor constant

$$dv = \frac{(Pv)(144)(1.004)}{(85.78)(T)}$$

1.004 = correction factor for deviation from Ideal Gas Law

$$dv = \frac{(Pv)(1.6854)}{(T)}$$

Problem: How many pounds of water vapor in one cubic foot of air at dew point temperature 50ºF and 14.7 psia.

$$dv = \frac{(0.17798)(1.6854)}{(50 + 460)} = .00059 \text{ pound/ft}^3$$

0.17798 = saturated water vapor pressure @ 50ºF (From "Psychometric Tables and Charts" by O. T. Zimmerman and Irvine Levine)

Problem: Same as the previous problem, but at 114.7 psia.

$$dv = \frac{(0.17798)(1.6854)}{(50 + 460)} = .00059 \text{ pound/ft}^3$$

Note the answers are the same for both pressures because the water vapor behaves independently of the air. This is true so long as the air and water vapor follow the Ideal Gas Law. At the pressures and temperatures encountered in the typical industrial system the deviations are insignificant.

Boyles Law;

The pressure of a given mass of gas at a constant temperature is inversely proportional to its volume.

Example:

At constant temperature T

If V increases, P decreases

If V decreases, P increases

$$\frac{P_1}{P_2} = \frac{V_2}{V_1}$$

or

$$P1V1 = P2V2 = Constant$$

Charles Law and Gay Lussac's Law;

If the pressure remains constant, a given mass of gas will increase in its volume proportional to an increase in temperature. For every 1°C rise in temperature, a volume initially at 0°C will increase by 1/273, keeping the pressure constant.

Example:

At constant temperature P

If T increases, V increases

If T decreases, V decreases

Perfect (Ideal) Gas Law;

By combining the relationships found in Boyles and Charles Laws, the Perfect Gas Law is developed:

$$PV = nRT$$

Where;

P = Pressure (Absolute)
V = Volume
R = Gas Constant (Air = 639.6)
n = Molecular Weight of Gas (Moles)
T = Temperature of Gas (Absolute)

Pascals Law;

The ability of a gas to transmit equal pressure in all directions, at right angles to the wall of its container, regardless of the containers shape. Force is equal to Pressure (PSIG) times Area (cubic inches).

$$P = F \div A$$

or

$$F = P \times A$$

Common Formulae

$$\text{SCFM} = \frac{\text{Volume (Cubic Inches)}}{\text{Time (Seconds)}} \times \frac{\text{Compression Factor}}{28.8}$$

SCFM (Standard Cubic Feet Per Minute)

(28.8 is the conversion factor for cubic inches per second to cubic feet per minute.)

$$\text{Compression Factor} = \frac{\text{Atmospheric Pressure + Gauge Pressure}}{\text{Atmospheric Pressure}}$$

$$\text{SCFM} = \text{CFM} \times \frac{\text{Atmospheric Pressure + Gauge Pressure}}{14.7} \times \frac{528}{\text{Air Temperature (°F)} + 528}$$

Absolute Temperature (°R) = Gauge Temperature (°F) +528

Absolute Pressure (PSIA) = Gauge Pressure (PSIG) + Atmospheric Pressure

Flow Coefficient

$$C_V = \frac{Q}{22.48}\sqrt{\frac{GT}{(P_1-P_2)P_2}}$$

Where:

- C_V = Dimensionless Number
- Q = Flow (SCFM)
- G = Ratio of molecular weight of gas of that of air. For air =1
- T = Absolute Temperature (°R)
- P_1 = Absolute inlet pressure (PSI)
- P_2 = Absolute Secondary Pressure (PSI)
  P_2 must be greater than .53P1

Handy Rules Of Thumb

Air Compressors:

1. Air compressors are normally rated to deliver 4 to 5 CFM per horsepower at 100 PSIG discharge pressure.
2. A 50 horsepower compressor rejects approximately 126,000 BTU per hour for heat recovery.
3. Motor amperage draw: 1 Phase: 115V - 10 Amps per HP
   230V - 5 Amps per HP
   3 Phase: 230V - 2.50 Amps per HP
   460V - 1.25 Amps per HP

Air Receivers:

1. Size air receiver tanks for about 1 gallon capacity for each CFM of rotary compressor capacity. Standard receiver tank sizes are listed below:

Storage Tank (Receiver) Sizes and Capacities

Diameter in Inches	Height in Inches	Gallons	Cu. Ft.
24	72	120	16
30	84	240	32
36	96	400	53
42	120	660	88
48	144	1060	142
54	168	1550	207
60	192	2200	294
66	214	3000	401
72	228	3800	508
84	232	5000	668
90	241	6000	802

Calculation for Minimum Receiver Capacity

- T = Time interval in minutes, during which a receiver can supply air without excessive drop in pressure
- V = Volume of receiver in cubic feet
- C = Air requirement of cubic feet of free air per minute
- Cap = Compressor capacity in cubic feet of free air per minute
- Pa = Absolute atmospheric pressure, psia.
- P1 = Initial Tank pressure, psig (compressor discharge pressure)
- P2 = Minimum Tank Pressure, psig (pressure required to operate plant)

FORMULA: $$V = \frac{T(C-Cap)(Pa)}{(P1-P2)}$$

If Cap is > C, resulting negative answer indicates that the air compressor will supply required load.

If Compressor is unloaded or shut down, Cap becomes zero, and receiver must supply the load for T minutes.

Rule of thumb: 2 gallons of storage per cfm of compressor capacity.

2. Cubic Feet: Gallons x 0.13368

Water Content:

1. The water vapor content at 100°F of saturated compressed air is about two gallons per hour for each 100 CFM of compressor capacity.
2. Every 20°F temperature drop in saturated compressed air at constant pressure, 50% of the water vapor condenses to liquid.

Water-Cooled Aftercoolers:

1. Most water-cooled aftercoolers will require about 3 GPM per 100 CFM of compressed air at Discharge Air Temperature at 100 psig.

Compressor Discharge Temperature (Before Aftercooling):

1. Approximate discharge temperatures (before aftercooling) at 80°F ambient:

Pressure	100 PSIG	150 PSIG	200 PSIG
Single-Stage	510	615	—
Two-Stage	325	365	395
Rotary (Oil-Cooled)	180 - 200	190 - 205	200 - 215

Horsepower & Power Cost:

1. Every 2 psig change in pressure equals 1% change in horsepower.
2. Most AIR MOTORS require 30 CFM at 90 psig per horsepower.
3. 10¢/ KWH Electric Power Rate = $806/ Year for 1 HP/3 shift Constant Run.
4. KW = HP x 0.7457

Saturated Compressed Air

1. At 100 psig every 20°F increase in saturated air temperature doubles the amount of moisture in the air.

Ventilation Required

The following formula will estimate the required ventilation air in cfm to adequately control ambient heat rejection from an air compressor. BE SURE TO USE FULL ABSORBED HORSEPOWER!

EXAMPLE:

WHEN

TOTAL BHP = 110
Start Temp. (T1) = 80°F
Max. Allow Temp. (T2) = 100°F

$$\text{CFM} = \frac{\text{(Sensible Heat BTU/ Hr.)}}{(1.08)(\text{T1 - T2})\text{Temp. Rise}}$$

$$\text{CFM} = \frac{110 \text{ HP} \times 2546}{(1.08)(20°)\text{Temp. Rise}} = \frac{280{,}060}{21.60}$$

CFM = 12,966 CFM Vent. Air Required

EXAMPLE:

Assume a 100 H.P. compressor system. The flow capacity of this compressor will be approximately 400 scfm. Assume further that on the average the compressor is delivering 80% of its capacity to the system over an 8 hour work shift.

100 H.P. = 400 scfm x .80 = 320 scfm average air delivered.

2.25 grains of water condensing per cubic foot
of ambient air ingested = 17.59 grains
condensed for each 7.8 ft^3 ingested
therefore
$17.59 \div 7.8 = 2.25$.

320 scfm x 60 (hour) x 8 hours x 2.25 grains
= 345,600 grains

345,600 ÷ 7,000 grains/lb. = 49.37 lbs.
of water condensed
49.37 ÷ 8.33 lbs./gallon = 5.93 gallons of water
condensed every 8 hours at the aftercooler / receiver
where compressed air temperature is assumed to be 100ºF.

The compressed air now leaves the receiver tank where it is piped into a cooler environment. Assume an indoor plant where the lowest ambient temperature will be 60ºF. As the air gradually cools additional condensation will occur and collect at points of use, low places in the piping, etc. The total condensation in the piping system can be determined using the above calculation procedure.

Water vapor content per ft^3 at 100ºF
in storage = 19.77 grains (see page 22).

Water vapor content per ft^3 at 60ºF
(the lowest ambient) = 5.75 grains

19.77 – 5.75 = 14.02 grains condensing for each
7.8 ft^3 of free air delivered to the system,
therefore 14.02 ÷ 7.8 = 1.8 grains condensing per ft^3.

320 x 60 x 8 x 1.8 = 276,480 grains/8 hrs.
276,480 ÷ 7,000 = 39.5 lbs.
39.5 ÷ 8.33 = 4.74 gallons
of water condensing in piping system or at
the points of use every 8 hours.

Assume we install a refrigerated dryer at the storage tank and chill the air to 50ºF (this is a 50º pressure dew point). We can now calculate the amount of condensation which will occur at the dryer. If this is more than will condense in the piping system we will have a dry air system. Let's see:

Water vapor content per ft^3 at 100ºF
in storage — 19.77 grains

Water vapor content per ft^3 at 50ºF
(dew point of air from dryer) = 4.08 grains

19.77 – 4.08 = 15.69 ÷ 7.8 = 2.01 grains
condensing per ft^3 delivered to the system,
therefore:

320 x 60 x 8 x 2.01 = 308,736 grains
308,736 ÷ 7,000 ÷ 8.33 = 5.29 gallons
condensing at the dryer every 8 hours.

We have a dry system!

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Parker Hannifin Corporation
Pneumatic Division
8676 E. M89
P.O. Box 901
Richland, MI 49083 USA
Tel: 269 629 5000
Fax: 269 629 5385

Applications Engineering
Phone: 877 321 4PDN Option #2
E-mail: pdnapps@parker.com

Customer Support
Phone: 877 321 4PDN Option #1
E-mail: pdncustsvc@parker.com
Web site: www.parker.com/pneumatics
www.parker.com/pneu/dryers