PDHonline Course M377 (2 PDH)

Air Compressors – Basic Concepts and Application

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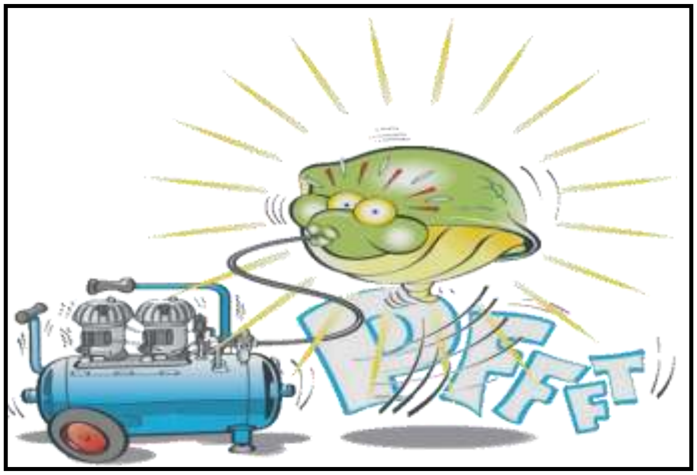
AIR COMPRESSOR BASIC CONCEPTS AND APPLICATION

CONTENTS:

I. INTRODUCTION:

Air compressor is a device that that increases the pressure of a gas by reducing its volume and converts power (using an electric motor, diesel or gasoline engine, etc.) into potential energy stored in a tank or air receiver (i.e., compressed air). This special course brings, didactically the main guidelines of how to calculate and install compressors and tanks or air receivers.

Compressors are similar to pumps; both increase the pressure on a fluid and both are designed and arranged to transport fluid through a pipe. As gases are compressible, the compressor also reduces the volume of a gas. Liquids are relatively incompressible; while some can be compressed, the main action of a pump is to pressurize and transport liquids.



Air Standards: There are three standards available:

1. **API Standard**: 14.7 psia, 60°F, 0% relative humidity
2. **ASME Standard**: 14.7 psia, 68°F, 36% relative humidity
3. **CAGI Standard**: 14.7 psia, 60°F, 36% relative humidity

Pressure (P) is the force per unit area applied in a direction perpendicular to the surface of an object. Mathematically it is P = F/A, where F is Force and A is Area. Celsius (C): (also known as centigrade) at 0°C is defined as the freezing point of water, the temperature at 100 °C is defined as the boiling point of water and 1.033 Kg/cm² is defined as the standard atmospheric pressure.

Some **pressure units**: 101,325 Pa = 1.013 bar = 1.033 kg/cm² abs = 760 mmHg = 14.7 psia

$$C° = \frac{F - 32}{1.8}$$

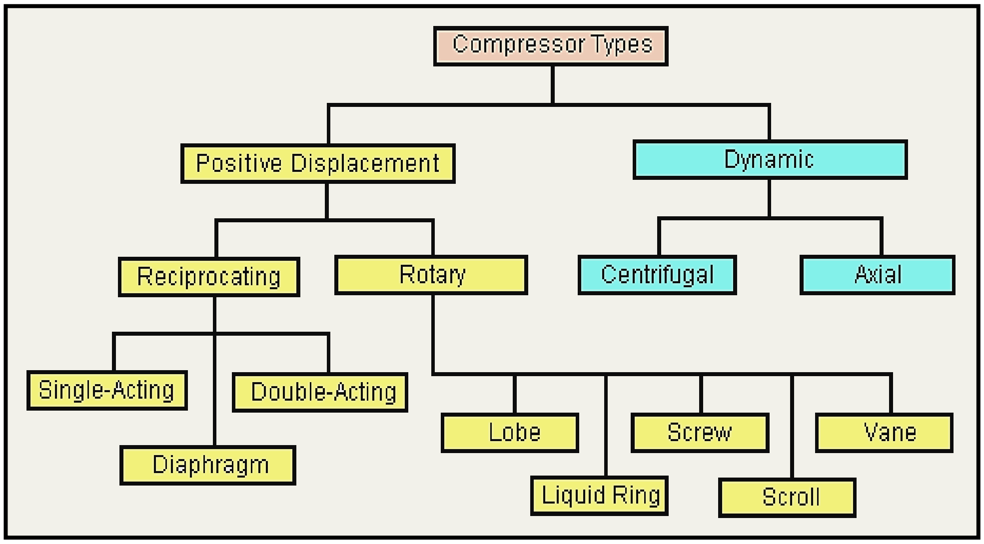
The temperature Fahrenheit at 32°F is defined as the freezing point of water, the temperature at 212°F is defined as the boiling point of water and 14.7 psi is defined as the standard atmospheric pressure.

F° = 1.8 C° + 32

Note: CAGI is a nonprofit organization of 45 companies that manufacture air and gas compressors, pneumatic machinery and air and gas drying equipment; products of applications worldwide in construction, manufacturing, mining, and the process and natural gas industries.

II. AIR COMPRESSORS TYPES:

The universal types of compressors are:



These types are further specified by:

a) The number of compression stages;
b) Cooling method (air, water, oil);
c) Drive method (motor, engine, steam, other);
d) Lubrication (oil free means no lubricating oil contacts the compressed air).

1) Reciprocating Air Compressors:

Reciprocating Air Compressors are positive displacement machines, meaning that they increase the pressure of the air by reducing its volume. This means they are taking in successive volumes of air which is confined within a closed space and elevating this air to a higher pressure. The Reciprocating Air Compressor accomplishes this by a piston within a cylinder as the compressing and displacing element specified as single or double-stage and single or double-acting.

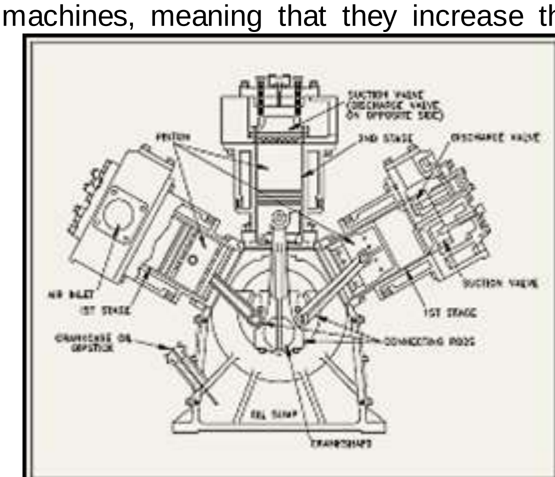


Figure 1 Reciprocating Air Compressor

Single-stage: When the entire compression is accomplished with a single cylinder or a group of cylinders in parallel generally used for pressures in the range of *70 psig to 100 psig.*

Double-stage: Is when two or more steps of compression are grouped in series generally used for pressures in the range of *100 psig to 250 psig.*

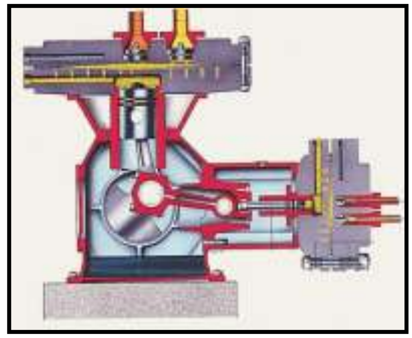
Single acting: Is when the compressing is accomplished using only one side of the piston.

Double acting: Are those using both sides of the piston.

Load reduction is achieved by unloading individual cylinders by throttling the suction pressure to the cylinder or bypassing air either within or outside the compressor. Capacity control is achieved by varying speed in engine-driven units through fuel flow control. Reciprocating Air Compressors are available either as air-cooled or water-cooled in lubricated and non-lubricated configurations and provide a wide range of pressure and capacity selections.

2) Diaphragm Compressors:

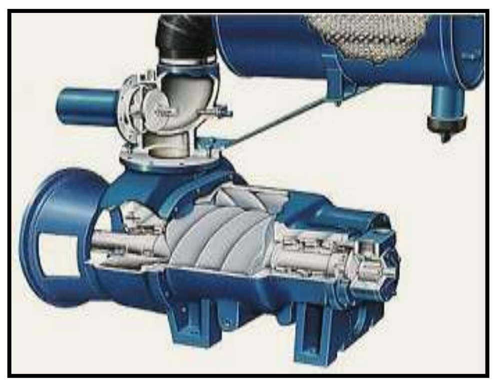
A Diaphragm Compressor is a variant of the classic reciprocating compressor with backup and piston rings and rod seal. The compression of gas occurs by means of a flexible membrane, instead of an intake element. The back and forth moving membrane is driven by a rod and a crankshaft mechanism. Only the membrane and the compressor box come in touch with pumped gas. For this reason this construction is the best suited for pumping toxic and explosive gases.



The membrane has to be reliable enough to take the strain of pumped gas. It must also have adequate chemical properties and sufficient temperature resistance. A Diaphragm Compressor is the same as a Membrane Compressor.

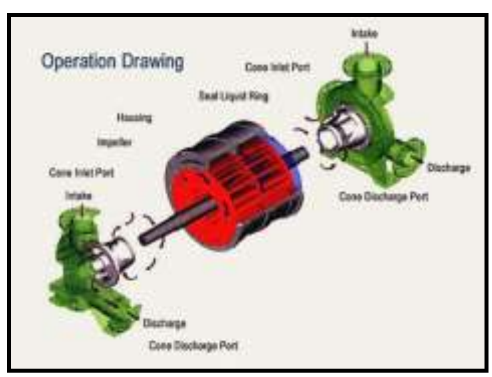
3) Rotary Screw Compressors:

Rotary Screw Compressors are also positive displacement compressors. The most common Rotary Screw Compressor is the single stage helical or spiral lobe oil flooded screw air compressor. These compressors consist of two rotors within a casing where the rotors compress the air internally. There are no valves. These units are basically oil cooled (with air cooled or water cooled oil coolers) where the oil seals the internal clearances. The working parts never experience extreme operating temperatures. The rotary compressor, therefore, is a continuous duty, air cooled or water cooled compressor package.



Rotary Screw Compressors are easy to maintain and operate. Capacity control for these compressors is accomplished by variable speed and variable compressor displacement. For the latter control technique, a slide valve is positioned in the casing. As the compressor capacity is reduced, the slide valve opens, by-passing a portion of the compressed air back to the suction. Advantages of the rotary screw compressor include smooth, pulse-free air output in a compact size with high output volume over a long life.

4) Rotary Vane (Hydrovane) Compressors:



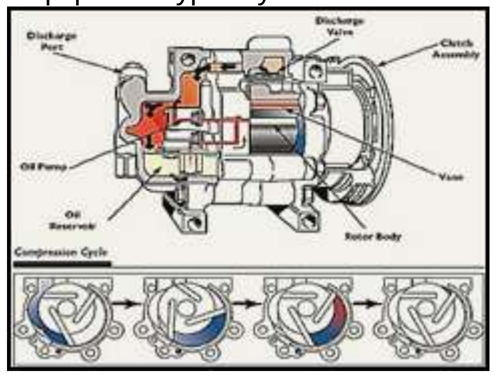
Vane compressors use "air-tool" type technology to compress air. The hydrovane design operates on the same principle as the air motor. A circular wheel (rotor) is fitted with multiple “vanes” that sweep air through as it turns. These vanes are spring loaded and by putting the rotor within an enclosure that is off center, compression is achieved with each turn of the rotor.

Their efficiency is about **4 SCFM per HP** and can work in extremely dirty environments. This means that dirt and contaminants are far less likely to get in a plug up the controls as can easily happen on an air controlled system.

For the smaller user that requires a continuously running a machine, the rotary vane design, or hydrovane, is an excellent choice.

5) Liquid Ring Compressors:

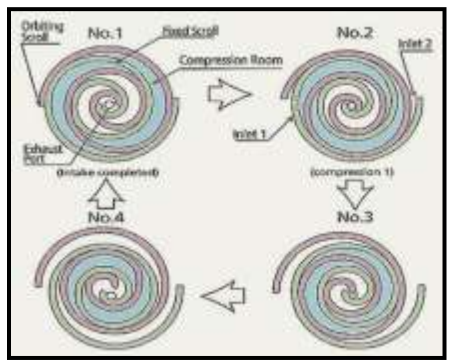
Also called Liquid Ring Pumps are rotating positive displacement equipment typically used as vacuum compressors but can also be used as vacuum pumps. The function of a Liquid Ring Compressor is similar to a Rotary Vane pump. The difference is that the vanes are an integral part of the rotor to form the compression chamber seal. They are an inherently low friction design, with the rotor being the only moving part.



The Liquid Ring Compressor compresses the gas by rotating a vanned impeller within an eccentric to a cylindrical casing. A liquid (usually water) is fed into the pump and, by centrifugal acceleration, forms a moving cylindrical ring against the inside of the casing. This liquid ring creates a series of seals in the space between the impeller vanes, which form compression chambers.

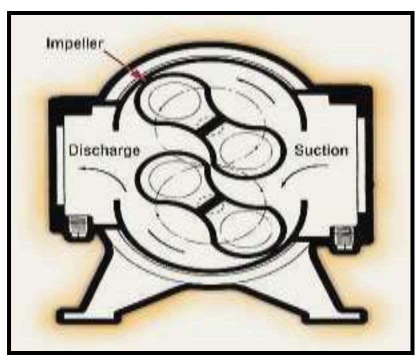
The eccentricity between the impeller's axis of rotation and the casing geometric axis results in a cyclic variation of the volume enclosed by the vanes and the ring. Gas, often air, is drawn into the pump via an inlet port in the end of the casing. The gas is trapped in the compression chambers formed by the impeller vanes and the liquid ring. The reduction in volume caused by the impeller rotation compresses the gas, flows to the discharge port in the end of the casing.

6) Scroll Compressors:



A Scroll Compressor (also called *Spiral Compressor*, *Scroll Pump* and *Scroll Vacuum Pump*) is a device for compressing air or refrigerant gas, using two interleaving scrolls to pump, compress or pressurize fluids such as liquids and gases. The vane geometry may be involute, Archimedean spiral, or hybrid curves. It is used in air conditioning equipment, as an automobile supercharger and can be also used to generate mechanical work from the expansion of a fluid, instead of the more traditional rotary, reciprocating, and wobble-plate compressors

7) Lobe Compressors:

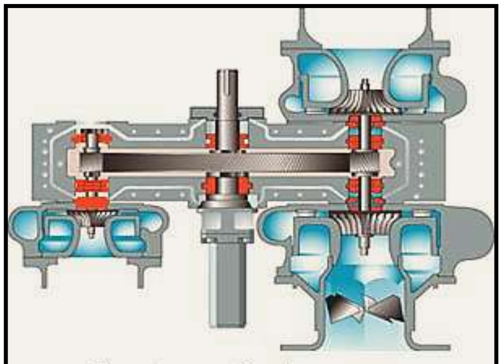


Rotary Lobe Air Compressor features two mating lobe-type rotors mounted in a case. The lobes are gear driven at close clearance, but without metal-to-metal contact. The suction is located where the cavity made by the lobes is largest. As the lobes rotate the cavity size is reduced causing compression of the gas within the case wall. The compression continues until the discharge pressure is reached at the point, where the gas exits the compressor at a higher pressure.

8) Centrifugal Compressors:

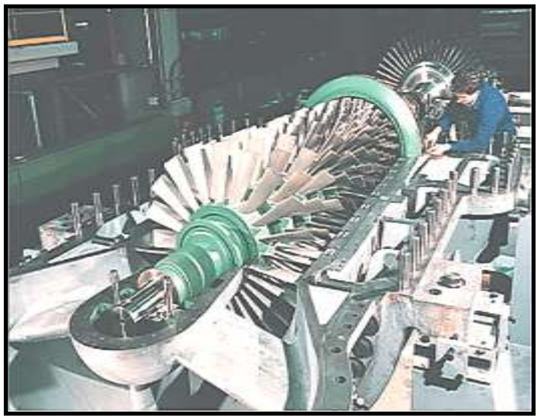
The Centrifugal Air Compressor is a dynamic type which depends on transfer of energy from a rotating impeller from about 200 HP on up to several thousand HP for specialized applications. First and foremost, these compressors need an application that requires a near constant flow of air, because, unlike a Rotary

Screw Compressor, it cannot be unloaded. Centrifugal Compressors produce high-pressure discharge by converting angular momentum imparted by the rotating impeller (dynamic displacement). In order to compress efficiently, centrifugal compressors rotate at higher speeds than the other types of compressors. These types of compressors are also designed for higher capacity because the flow through the compressor is continuous. The Centrifugal Air Compressor is an oil-free compressor by design. The oil lubricated running gear is separated from the air by shaft seals and atmospheric vents.


Three stage centrifugal compressor

9) Axial Compressors:

Axial Compressors are rotating, airfoil-based compressors, in which the working fluid flows parallel to the axis of rotation and produce a continuous flow of compressed gas, with high efficiency and large mass flow capacity in relation to their cross-section.



Axial compressors are widely used in gas turbines, such as jet engines, high speed ship engines, and small scale power stations. They are also used in industrial applications, such as large volume air separation plants, blast furnace air, fluid catalytic cracking air, and propane dehydrogenation.

III. Nm³/h and SCFM:

Generally, the capacity of a compressor is given in **Nm³/h (Normal Cubic Meter per Hour) or SCFM (Standard Cubic Feet per Minute)**. The weather conditions (atmospheric pressure, air temperature and relative humidity) and altitude conditions are necessary to know, before installing any type of compressor.

Nm³/h – refers to:

- Atmospheric Pressure at Sea Level = **1.033 kg/cm² abs.**
- Temperature = **273°K (0ºC).**
- Relative Humidity = **0% (dry).**

SCFM – refers to:

- Atmospheric Pressure at Sea Level = **14.7 psi abs.**
- Temperature = **60°F (15.6°C).**
- Relative Humidity = **0% (dry).**

	Nm³/h	SCFM
Altitude	Sea Level	Sea Level
Pressure	1.033 kg/cm² (abs)	14.7 psi (abs)
Temperature	0°	60°F (15°C)
Relative Humidity	0%	0%

So, to **design and install a compressor** it is necessary to **calculate the volumetric conversion** using the following formulas:

$$\text{Nm}^3/\text{h} = \text{m}^3/\text{h}.\ \frac{273}{273 + t1} \cdot \frac{P1 - (Rh \times Pv)}{1.013} = \text{(metric)}$$

Where**:**

t1: Inlet temperature of air (°C);
P1: Inlet pressure in the compressor unit admission (kg/cm² abs.), **according to Altitude – See Table II**;
Rh: Local relative humidity (%);
Pv: Partial pressure of saturated vapor (bar abs.), **according to Temperature – See Table I**.

$$\text{SCFM} = \text{CFM}.\ \frac{520}{460 + t_1} \cdot \frac{P_1 - (Rh \times Pv)}{14.7} = \text{(imperial)}$$

Where**:**

t_1: Inlet temperature of air (°F);
P_1: Inlet pressure in the compressor unit admission (psia), **according to Altitude – See Table II;**
Rh: Local relative humidity (%);
Pv: Partial pressure of saturated vapor (psia), **according to Temperature – See Table I.**

Air Conditions:

Typical nomenclature for most parts of the world using "**metric or imperial**" units of volume. Normal conditions are measured at "**0ºC or 32ºF**". Gas volume is directly proportional to temperature and inversely proportional to pressure. The most nomenclature used is in metric conditions, such as, absolute temperature = **272ºC + temperature, ºC**. The standards conditions defined by NIST (National Institute of Standards and Technology), commonly used for testing and documentation of compressor capacities are:

NIST Standards:

Volume air at T1 = V1; Volume air at T2 = V2;
T1 = "Standard" conditions at 20ºC (typical metric value);
T2 = "Normal" conditions at 0ºC (typical metric value).

$$\textbf{V2 air} = V1\ \frac{(273^o + T2)}{(273^o + T1)} =$$

Example :

$$\textbf{V2 air} = \frac{V1\ (273 + 0)}{(273 + 20)} = \mathbf{0.9317}$$

Note: V2 air is the volume of air at 0ºC, which is less than volume of air at 20ºC. The "**normal**" volume, (V1 x 0.9317) is designated as "**standard**" volume at 20˚C.

Table I – Pressure of Saturated Vapor (psia / kg/cm^2 abs) – Partial:

Temperature		Pressure - Abs		Temperature		Pressure - Abs		Temperature		Pressure - Abs	
°F	°C	lb/pol^2	kg/cm^2	°F	°C	lb/pol^2	kg/cm^2	°F	°C	lb/pol^2	kg/cm^2
32	→ 0,0	0.088	0.00619	65	18,3	0.305	0.02144	98	36,7	0.893	0.06279
33	0,6	0.092	0.00647	66	18,9	0.316	0.02222	99	37,2	0.921	0.06475
34	1,1	0.096	0.00675	67	19,4	0.327	0.02299				
35	1,7	0.100	0.00703	68	→ 20,0	0.339	0.02383	100	37,8	0.949	0.06672
36	2,2	0.104	0.00731	69	20,6	0.350	0.02461	101	38,3	0.978	0.06871
37	2,8	0.108	0.00759					102	38,8	1.007	0.07080
38	3,3	0.112	0.00787	70	21,1	0.363	0.02552	103	39,4	1.038	0.07298
39	3,9	0.117	0.00822	71	21,7	0.375	0.02637	104	→ 40,0	1.069	0.07516
				72	22,2	0.388	0.02728	105	40,6	1.101	0.07741
40	4,4	0.121	0.00851	73	22,8	0.401	0.02820	106	41,1	1.134	0.07973
41	→ 5,0	0.126	0.00886	74	23,3	0.415	0.02918	107	41,7	1.168	0.08212
42	5,6	0.131	0.00921	75	23,9	0.429	0.03016	108	42,2	1.202	0.08451
43	6,1	0.136	0.00956	76	24,4	0.444	0.03122	109	42,8	1.238	0.08704
44	6,7	0.142	0.00998	77	→ 25,0	0.459	0.03227				
45	7,2	0.147	0.01033	78	25,6	0.474	0.03333	110	43,3	1.274	0.08957
46	7,8	0.153	0.01076	79	26,1	0.490	0.03445	111	43,9	1.312	0.09224
47	8,3	0.159	0.01118					112	44,4	1.350	0.09492
48	8,9	0.165	0.01160	80	26,7	0.506	0.03558	113	→ 45,0	1.389	0.09766
49	9,4	0.171	0.01202	81	27,2	0.523	0.03677	114	45,6	1.429	0.10047
				82	27,8	0.541	0.03804	115	46,1	1.470	0.10335
50	→ 10,0	0.178	0.01251	83	28,3	0.558	0.03923	116	46,7	1.513	0.10638
51	10,6	0.184	0.01294	84	28,9	0.577	0.04057	117	47,2	1.556	0.10940
52	11,1	0.191	0.01343	85	29,4	0.595	0.04183	118	47,8	1.600	0.11249
53	11,7	0.199	0.01399	86	→ 30,0	0.615	0.04324	119	48,3	1.645	0.11566
54	12,2	0.206	0.01448	87	30,6	0.635	0.04465				
55	12,8	0.214	0.01505	88	31,1	0.655	0.04605	120	48,9	1.692	0.11896
56	13,3	0.222	0.01561	89	31,7	0.676	0.04753	121	49,4	1.740	0.12234
57	13,9	0.230	0.01617					122	→ 50,0	1.788	0.12571
58	14,4	0.238	0.01673	90	32,2	0.698	0.04908	123	50,6	1.838	0.12923
59	→ 15,0	0.247	0.01737	91	32,8	0.720	0.05062	124	51,1	1.889	0.13281
				92	33,3	0.743	0.05224	125	51,7	1.942	0.13654
60	15,6	0.256	0.01800	93	33,9	0.766	0.05386	126	52,2	1.995	0.14027
61	16,1	0.265	0.01863	94	34,4	0.790	0.05554	127	52,8	2.050	0.14413
62	16,7	0.275	0.01933	95	→ 35,0	0.815	0.05730	128	53,3	2.106	0.14807
63	17,2	0.285	0.02004	96	35,6	0.840	0.05906	129	53,9	2.163	0.15208
64	17,8	0.295	0.02074	97	36,1	0.866	0.06089				

lb/pol2 = psi

Table II - Altitude and Air Pressure:

Altitude		Pressure			
Feet	Meter	in.Hg	torr	psia	Kg/cm² abs
0	0	29.92	759.97	14.7	1.033
500	152	29.39	746.51	14.43	1.015
1000	305	28.86	733.04	14.17	0.956
1500	457	28.34	719.84	13.91	0.978
2000	610	27.82	706.63	13.66	0.960
2500	762	27.32	693.93	13.41	0.943
3000	914	26.82	681.23	13.17	0.926
3500	1067	26.33	668.78	12.93	0.909
4000	1219	25.84	656.34	12.69	0.892
4500	1372	25.37	644.4	12.46	0.876
5000	1524	24.9	632.46	12.23	0.860
5500	1676	24.44	620.78	12	0.843
6000	1829	23.98	609.09	11.77	0.828
6500	1981	23.53	597.66	11.55	0.812
7000	2134	23.09	586.49	11.34	0.797
7500	2286	22.66	575.56	11.13	0.782
8000	2438	22.23	564.64	10.91	0.767
8500	2591	21.81	553.97	10.71	0.753
9000	2743	21.39	543.31	10.5	0.738
9500	2896	20.98	532.89	10.3	0.724
10000	3048	20.58	522.73	10.1	0.710
10500	3200	20.19	512.83	9.91	0.697
11000	3353	19.8	502.92	9.72	0.683
11500	3505	19.41	493.01	9.53	0.670
12000	3658	19.03	483.36	9.34	0.656
12500	3810	18.66	473.96	9.16	0.644
13000	3962	18.3	464.82	8.99	0.632
13500	4115	17.94	455.68	8.81	0.619
14000	4267	17.58	446.53	8.63	0.606
14500	4420	17.24	437.9	8.46	0.595
15000	4572	16.89	429.01	8.29	0.583
15500	4724	16.56	420.62	8.13	0.572
16000	4877	16.22	411.99	7.96	0.559
16500	5029	15.9	403.86	7.81	0.549
17000	5182	15.58	395.73	7.65	0.538
17500	5334	15.26	387.6	7.47	0.525
18000	5486	14.95	379.73	7.34	0.516
18500	5639	14.65	372.11	7.19	0.505
19000	5791	14.35	364.49	7.05	0.495
20000	6096	13.76	349.5	6.76	0.475

Notes:

- The **air density** varies with **pressure** (the Ideal Gas Law) and the **altitude** above sea level. The absolute pressure at sea level is approximately ***1.033 kg/cm² (14.7 psia, 760 mmHg, 101.325 kPa,)*** with a variation is *+/- 5%*.

- The **air pressure** varies with altitude:

1. **STP** - Standard Temperature and Pressure = At **0°C** pressure is **1.033 kg/cm² (101.325 kPa).**
2. **NTP** - Normal Temperature and Pressure. Commonly used for testing and documentation of fan capacities at ***20°C* and 1. 0360 kg/cm2 (*101.6 kPa, 14.735 psia or 30 in Hg at 68°F)***.

Example:

Check the **m³/h** or **FAD** (Free Air Delivery) installation of a Screw Compressor capacity = **1400 Nm³/h** (870 SCFM) for the following local conditions:

- **Altitude** = 763 m above sea level (2500 ft)
- **Temperature** = 26°C (79°F)
- **Relative Humidity** = 80%

Applying formula for Normal- m³/h (Metric):

$$Nm^3/h = m^3/h . \frac{273}{273 + t1} . \frac{P1 - (Rh \times Pv)}{1.033} =$$

Where:

Altitude 763 m = According to **Table II** - P_1 = **0.943 kg/cm³ abs**;
Pv = According to **Table I** - Pressure of Saturated Vapor - 26°C = **0.03445 kg/m² abs**;
Rh = **80%**;
t_1 = **26°C**.

Then to calculate in **m³/h or FAD** (Free Air Delivery), above sea level is:

$$1400 = m^3/h . \frac{273}{273 + 26} . \frac{0.943 - (0.8 \times 0.03445)}{1.033} =$$

$$m^3/h = \frac{1400}{0.80913} = 1730\ m^3/h$$

Then, **1400 Nm³/h** (870 SCFM) corresponds to **1730 m³/h** (1018 CFM) FAD (Free Air Delivery).

Note: Therefore is essential to verify the actual use of compressed air when preparing the specification of a compressor, i.e., the flow must be specified in free air, **m³/h**, **Nm³/h, CFM or SCFM**. Whenever the capacity of a compressor is expressed in **Nm³/h or SCFM**, we need to know the weather conditions (atmospheric pressure, air temperature and relative humidity) and **altitude conditions** of the location where is installed the compressor.

Applying formula (Imperial):

$$SCFM = CFM.\ \frac{520}{460 + t1}\ .\ \frac{P1 - (Rh \times Pv)}{14.7} =$$

Where:

Altitude 2500 ft = According to **Table II** - P_1 = **13.41 psia**;
Pv = According to **Table I** - Pressure of Saturated Vapor - 79°F = **0.490 psia**;
Rh = 80%;
t_1 = 79°F.

$$870 = CFM.\ \frac{520}{460 + 79}\ .\ \frac{13.41 - (0.8 \times 0.490)}{14.7} =$$

870 = **CFM** x 0.96474 x 0.8856 = **1018 CFM**

Then, **870 SCFM** (1400 Nm³/h) corresponds to **1018 CFM** (1730 m³/h) or FAD (Free Air Delivery).

IV. AIR DRYERS:

After-coolers are heat exchangers that utilize either water or ambient air to cool the compressed air. An after-cooler discharging compressed air at **100°F passes 67 gallons of water with 1,000 SCFM** each 24 hours. To avoid these problems, compressed air systems have purification devices available to remove the water vapor and other contaminants. As the water and lubricant vapors within the compressed air cool, a significant amount condenses into liquid.

Refrigerated Air Dryers: Remove moisture from the compressed air through a mechanical refrigeration system to cool the compressed air and condense water and lubricant vapor. Most refrigerated dryers cool the compressed air to a temperature of approximately **35°F**, resulting in a pressure dew point range of **33°F - 39°F**.

Desiccant Dryers: Utilize chemicals beads, called desiccant, to adsorb water vapor from compressed air. Silica gel, activated alumina and molecular sieve are the most common desiccants used. (Silica gel or activated alumina is the preferred desiccants for compressed air dryers). The desiccant provides an average **-40°F** pressure dew point performance. Molecular sieve is usually only used in combination with silica gel or activated alumina on **-100°F** pressure dew point applications.

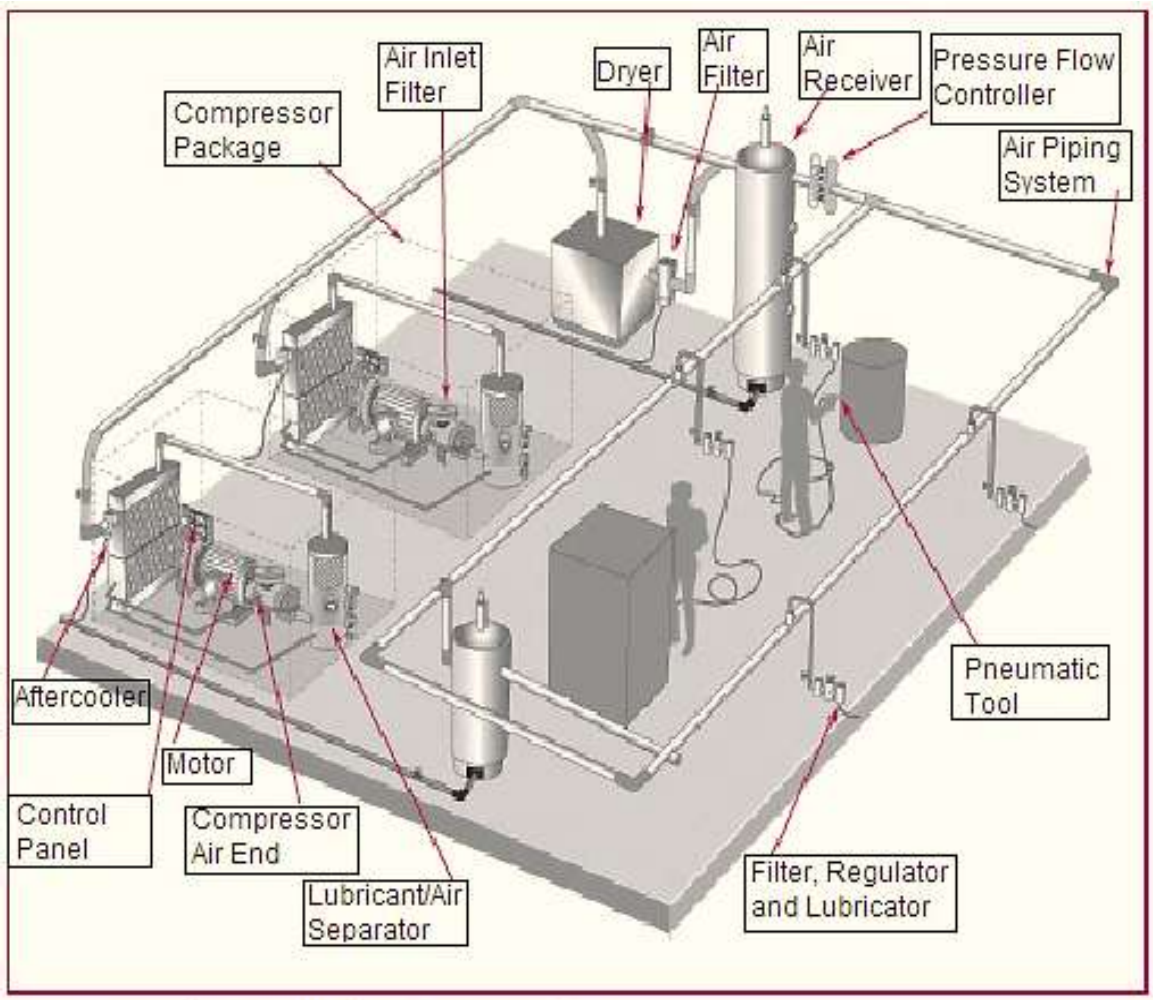
Deliquescent Air Dryers: Utilize an absorptive type chemical, called a desiccant, to provide a **20°F to 25°F** dew point suppression below the temperature of the compressed air entering the dryer. While deliquescent dryers are typically used in applications such as sandblasting and logging operations, they are not recommended for industrial applications since the dried compressed air exiting the dryer may contain small amounts of the effluent which may be corrosive to downstream equipment.

Coalescing Filters: Coalescing filters are the most common form of air purification; however, these types of filters can only remove previously condensed liquids. These filters remove water and lubricants from compressed air and are installed downstream in a refrigerated air dryer system or upstream in a desiccant dryer system.

Note: It´s necessary to keep in mind that this range is also the simplest achievable system with a refrigerated design, as the condensate begins to freeze at **32°F**.

Manufacturers typically require filter replacements, when the pressure drop reaches 10 psi, which is approximately 6 to 12 months of operation. Coalescing filters will also remove particulate contamination; however, this will increase the pressure drop across the filter and shorten the filter element life.

Obs.: Filters are rated according to liquid particle retention size (micron) and efficiency, such as 0.50 micron and 99.99% D.O.P. (Dispersed Oil Particulate) efficient, or 0.01 micron and 99.9999% D.O.P. HEPA (High Efficiency Particulate Air) or ULPA (Ultra Low Penetration Air) testing is the process in which the integrity of your filter is tested through the introduction of particulates.



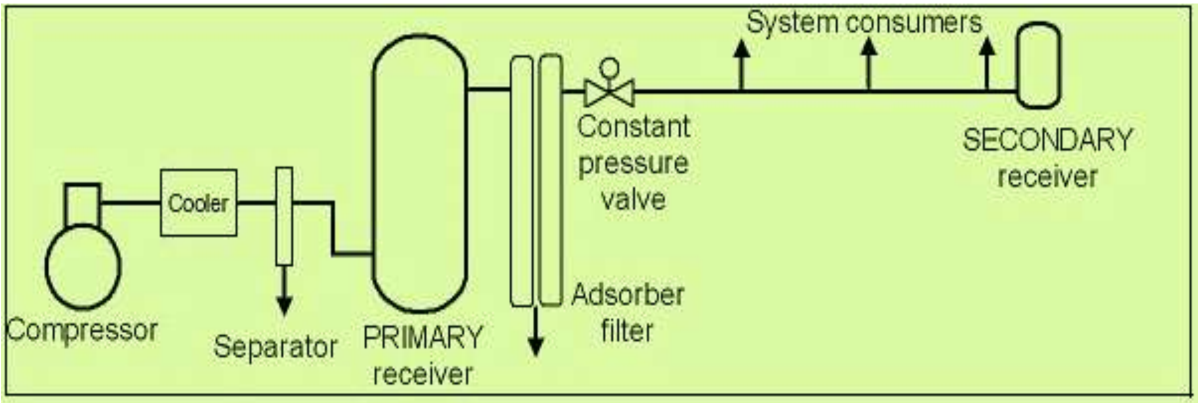
Components of a Typical Industrial Compressed Air System.

V. AIR TANKS OR AIR RECEIVERS:

An air receiver is essential to every compressed air system to act as a buffer and a storage medium between the compressor and the consumption system. There are in principal two different air receivers in a compressed air system:

1) **Primary Air Receiver:** Also designated as air tank, is located near the compressor, after the aftercooler but before filtration and drying equipment.

2) **Secondary Air Receiver:** Located close to points of larger intermittent air consumptions.



Since the maximum capacity of an air compressor also always exceeds the minimum air consumption in the system - the compressor must modulate its capacity during normal work, often by using simple strategies as on/off modulating or more advanced strategies, using frequency drives and inverters. Air receivers in compressed air systems serve the important purposes of:

- Equalizing the pressure variation from the start/stop and modulating sequence of the compressor;
- Storage of air volume equalizing the variation in consumption and demand from the system;
- Collecting condensate and water in the air after the compressor.

Sizing an Air Receiver: In general it is possible to calculate the maximum consumption in the system by summarizing the demand of each consumer. The summarized consumption must be multiplied with a usage factor ranging 0.1 - 1 depending on the system. If the consumption process requires 100 psig and the compressor is set to 100 psig, it´s not necessary storage and buffer. Any increased demand that makes a pressure drop **below 100 psig** makes the compressor controls respond automatically, by increasing the volume compressed. If the compressors operate at 110 psig the difference between 110 psig and 100 psig accounts for the air stored in the receiver. If the demand increases, the pressure can drop 10 psig before the minimum requirement. Pressure and flow controllers are used after the receiver for stabilizing downstream pressure to 100 psig and flattening demand peaks. The pipe system also makes the purpose of a buffered volume.

The air receiver must in general be sized according to:

- ✓ The variation in the consumption demand;
- ✓ The compressor size and the modulation strategy.

The time needed to fill an air receiver may be calculated with the formula:

$$t = \frac{V \times (P_1 - P_2)}{Q \times Pa} =$$

Where:

V = Volume of the receiver tank (cu ft);
t = Time for the receiver to go from upper to lower pressure limits (min);
Q = Free air needed (CFM);
Pa = Atmospheric pressure (14.7 psia);
P1 = Maximum tank pressure (psia);
P2 = Minimum tank pressure (psia).

Note: When a compressor pumps **1 CFM**, in theory, means that it also **sucked in 1 CFM** of "**free air**" (air at atmospheric pressure). It is also common to size air receivers:

- 1 gallon of air for each ACFM (Actual Cubic Feet per Minute);
- 4 gallons of air per compressor HP (Horse Power).

Example:

For a tank 3.3 ft³ capacity, a compressor "cuts-in" at 85 psi and "cuts out" at 102 psi and the compressor adds 1.1 atm of pressure during each cycle of 35 minutes.

Then, a tank 3.3 ft³ x 1.1 atm = **3.6 cubic feet per 35 seconds**;

In minutes:

3.6 cubic feet x 60/35 = **6.2 CFM (at 85 psi**).

Table III - Recommended Air Receiver Capacity:

Airflow Capacity		Recommended Receiver Volume		
(cfm)	(m3/h)	(cu ft)	(gal)	(m3)
100	170	13	100	0.4
200	340	27	200	0.8
300	510	40	300	1.1
400	680	54	400	1.5
500	850	67	500	1.9
750	1275	101	750	2.9
1000	1700	134	1000	3.8
1500	2550	201	1500	5.7
2000	3400	268	2000	7.6
3000	5100	402	3000	11.4
4000	6800	536	4000	15.2
5000	8500	670	5000	19.0
7500	12750	1005	7500	28.5
10000	17000	1340	10000	38.0

Compressor Power		Recommended Receiver Volume		
(hp)	(kW)	(cu ft)	(gal)	(m3)
5	3.7	3	20	0.1
7.5	5.6	4	30	0.1
10	7.5	5	40	0.2
15	11.2	8	60	0.2
20	14.9	11	80	0.3
25	18.7	13	100	0.4
30	22.4	16	120	0.5
40	29.8	21	160	0.6
50	37.3	27	200	0.8
60	44.8	32	240	0.9
75	56.0	40	300	1.1
100	74.6	54	400	1.5
125	93.3	67	500	1.9
200	149.2	107	800	3.0
350	261.1	188	1400	5.3
450	335.7	241	1800	6.8
500	373.0	268	2000	7.6

Air Tank Sizes:

The air compressor receivers should have **1 US gallon** of capacity for **every CFM** of compressor output. Therefore, since a **25 HP compressor** can theoretically generate about **100 CFM at 90 PSI**, the receiver for that compressor should be **100 gallons in size**.

Note: In general - **1 HP compress ~5 CFM (0.14 m³/h) at pressure 120 psi (8.5 kg/cm² = 8.3 bar) or, 1 HP compress ~4 CFM at 90 psi.**

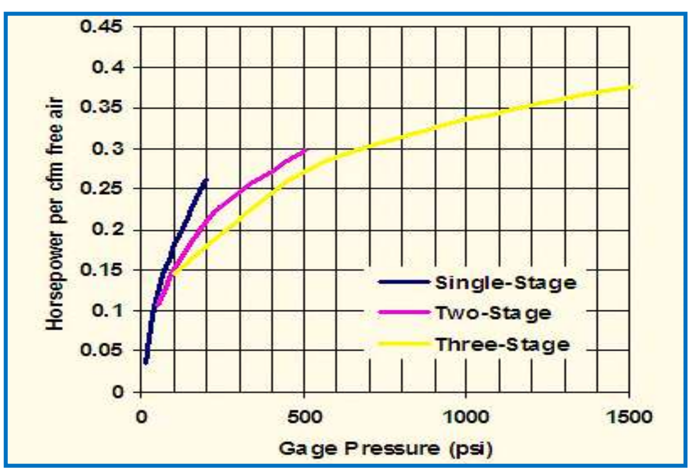
Table IV – Air Tanks (Air Receivers) Sizes:

Tank Size	Tank Size	Gauge Pressure on Tank *(psig)*			
(inches)	*(gallons)*	0	100	150	200
12 x 24	10	1.3	11	15	19
14 x 36	20	2.7	21	30	39
16 x 36	30	4.0	31	45	59
20 x 48	60	8.0	62	90	117
20 x 63	80	11	83	120	156
24 x 68	120	16	125	180	234

Common Conversions:

1 Nm³/h *= 1 SCFM x 1.6077;*
1 m³/h *= 1 CFM x 0.5885;*
1 SCFM *= 1 Nm³/h x 0.622;*
1 CFM *= 1 m³/h x 1.699;*
1 CFM *= 1 m³/min x 0.028;*
1 psig *= 6.684 kPa = 0.069 bar = 0.0703 kg/cm²;*
1 Gallon (U.S.) *= 0.03785 m^3 = 3.785 dm^3 (liter) = 231 in³ = 0.1337 ft³;*
1 ft^3 *= 1 m³ x 0.028;*
1 FPM *= 1 m/min x 0.305.*

Reciprocating Compressor HP: The theoretical **horsepower** (HP) required to **compress 1.0 cubic foot of free air** (atmospheric pressure) for a **single-staged, two-staged and three-staged** compressors, are indicated in the diagram below. In general **plus 15 - 20%** friction:



Note: For compressors **smaller than 10 HP**, is necessary to read the specifications for these particular units to determine the flow and pressure rates or use the "*guestimate*" of **2 CFM at 90 PSI per HP** of electric motor. The power required to adiabatic compression of air can be expressed as (imperial units):

$$HP = \frac{144.\,N.\,P_1.V.k}{33000\,(k-1)} \times \frac{P_2}{P_1}^{k-1/k} - 1$$

Where:

- **HP** = Horsepower;
- **N** = Number of compression stages;
- **k** = **1.41** = adiabatic expansion coefficient;
- P_1 = Absolute initial atmospheric pressure (psi) - (**14.7 psi at sea level**);
- P_2 = Absolute final pressure after compression (psi);
- **V** = Volume of air at atmospheric pressure (cfm).

The power required to adiabatic compression of air can be expressed as **(metric units)**:

$$kW = 1.634.Q.P1.\,\frac{k}{k-1} \times \frac{P_2}{P_1}^{k-1/k} - 1$$

Where:

- **kW** = Kilowatt;
- **Q** = Volume of air - m³/min;
- P_1 = Inlet pressure (mPa) – (1.033 kg/cm² abs at sea level);
- P_2= Outlet pressure after compression (kg/cm²);
- **k** = 1.41 = Adiabatic expansion coefficient.

The chart below shows air consumption for a 1" stroke cylinder, in a variety of standard bore sizes, and with a specific cycle rate.

Table V – Air Requirements for Pneumatic Cylinders:

Air Cylinder – 1" Stroke Requirements at 90 psi		
Cylinder bore size - in	**Air volume - ft³**	**CFM required for 10 extensions & 10 retractions**
1.0	0.79	0.0079
2.0	3.14	0.04
2.5	4.9	0.06
3.25	8.29	0.10
4.0	12.5	0.15
5.0	19.6	0.23
6.0	28.2	0.33
8.0	50.2	0.60
10.0	78.5	0.91

Power (HP) to Operate Air Compressors: The table below is shown for **single-stage, two-stage, and three-stage piston-type compressors**, assuming their efficiency to be about 85%. To **convert SCFM into HP** see the appropriate column in the table below. Since isothermal and adiabatic compressions are both theoretical conditions, this table were calculated for compression conditions about halfway between these two theoretical extremes. Inlet air is assumed to be about room temperature.

Table VI – Capacity for Piston-Type Air Compressors:

1-Stage Compressor		2-Stage Compressor		3-Stage Compressor	
PSI	HP*	PSI	HP*	PSI	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

Note: For example, if a cylinder consumption has been calculated to be **24 SCFM** and the installed compressor is a **two - stage** model, the **HP needed at 90 PSI will be: N = 24 x 0.156 = 3.7HP.**

Example:

Check the **table** to find the power of 1-stage compressor, considering that **1 HP compresses 5 CFM at pressure 120 psi**,.

Solution:

For **1-stage compressor** and 120 psi the table shows = **0.196 HP**, so:

N = 5 CFM x 0.196 = **0.98 HP (~ 1 HP)**

Notes:

1. Compressors from ***1 to 50 HP*** are typically for **Reciprocating Compressors**.

2. Compressors ***100 HP*** **and above** are typically **Rotary Screw or Centrifugal Compressors**.

3. **Positive displacement** compressors (Reciprocating, Rotary Screw) are **isentropic** machines.

4. **Dynamic compressors** (centrifugal or axial compressors) are **polytropic** machines.

5. Air compressors can **generate** over **20 gallons of water in an 8-hour** operating period. If not removed, the moisture and contaminants can cause premature **failure** of piping and pumping equipment.

Rating of Air Compressors: There is no universal standard for rating air compressors, air equipment and tools. The most common terms are:

CFM:
- CFM (Cubic Feet per Minute) is the U.S or imperial method of describing the volume flow rate of compressed air. It must be defined further to also take account of pressure, temperature and relative humidity.

SCFM:
- SCFM (Standard CFM) is the standard flow in CFM (ft^3/min) measured at some reference point according to sea level, but converted back to normal air conditions in the installation plant (Standard Reference Atmosphere - 14.7 psia, 60°F and 0% Relative Humidity).

ICFM:
- ICFM (Inlet CFM) is used to measure the air flow in CFM (ft^3/min) as it sucks the air or enters the air compressor intake.

ACFM:
- ACFM (Actual CFM) is used to measure air flow in CFM at some reference point at local conditions. This is the actual volume flow rate in the compressor outlet.

FAD:
- FAD (Free Air Delivery) is the real quantity measurement of air at the discharge of the compressor. The units are in CFM in the U.S or imperial system and m^3/min in the SI system, measured according to the ambient standard conditions ISO 1217 – 1, bar abs., and 20°C.

European References:

ANR:
- The ANR (Atmosphere Normale de Reference) is quantity of air at ambient conditions **1.013 bar absolute** (= 1.033 kg/cm^2 abs), **20°C** and **65% RH** (Relative Humidity).

Nl/min:
- Is the flow in **liters/min** measured at some reference point but converted to standard or normal air conditions **1.013 bar absolute** (1.033 kg/cm^2 abs), **0°C** and **0% RH** (Relative Humidity).

ISO 1217:
- The standard reference of air at ambient conditions - temperature **20°C**, pressure **1 bar abs.**, relative humidity **0%**, cooling air/water **20°C**, and working pressure of **7 bar** absolute at outlet.

VI. MEASUREMENT OF FREE AIR DELIVERY (FAD):

The capacity of a compressor **is the full rated volume of flow of air/gas compressed** and delivered under conditions of total temperature, total pressure, and composition at the compressor inlet meaning the **actual flow rate**. This is called **Free Air Delivery (FAD)** i.e., air at atmospheric conditions at any specific location. The power and air wastage depends on the percentage deviation of FAD, with a periodic assessment of each compressor, which has to be carried out to check its actual capacity. If the deviations are more than 10%, corrective measures should be taken.

The ideal method of a compressor capacity measurement is called the **Nozzle Method** wherein a calibrated nozzle is used to give pressure load velocity at the generated compressed air. Flow is assessed, based on the air temperature, pressure, constant velocity, etc. Nevertheless, the easier way of determining the Free Air Delivery of a compressor is by the **Pump up Method**, also known as receiver filling method. Although it is less accurate, this test can be adopted where the **Nozzle Method** is more difficult to be performed, due the complex arrangement of the test equipment.

Pump up Test:

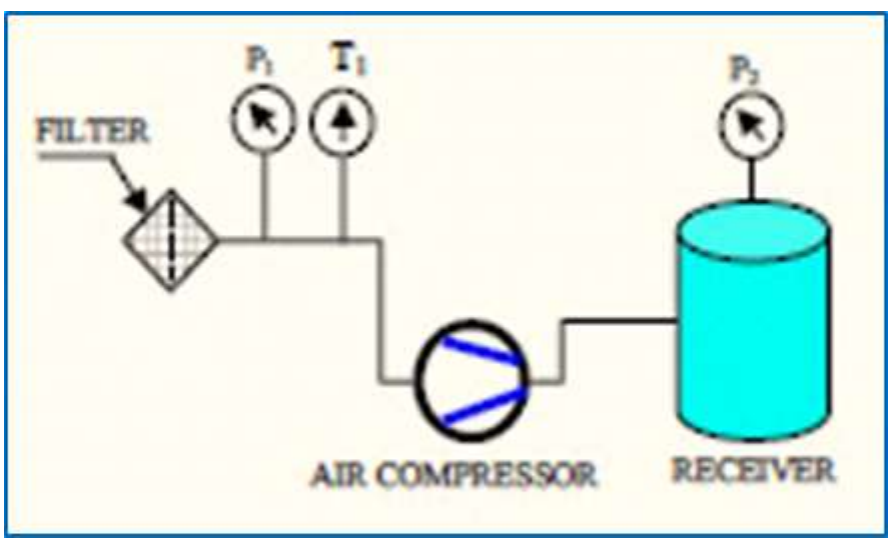
Open the water drain valve and fully drain out water of the air receiver and the pipeline. Make sure that the water trap line is tightly closed, before starting the test.

- Start the compressor and activate a stopwatch;
- Note the time taken at normal operational pressure P_2 (in the receiver) and the initial pressure P_1;
- Calculate the capacity as per the formulae given below:

$$Q = \frac{(P_2 - P_1) \times V}{P_a \times t} =$$

Where:

P_2 = Final pressure after filling (kg/cm² abs.);
P_1 = Initial pressure (kg/cm² abs) after bleeding;
P_a = Atmospheric pressure (1.033 kg/cm² abs);
V = Storage volume in m□, which includes the air receiver and delivery piping;
t = Time take to pump up pressure to P_2, in minutes;



Example:

A **Reciprocating Compressor** has the following data below. Calculate the Free Air Delivery (FAD).

- Piston displacement = **16.88 m³/min**;
- Theoretical compressor capacity = **14.75 m³/min @ 7 kg/cm²**;
- Air receiver volume = **7.79 m³;**
- Additional volume (piping) = **0.4974 m³;**
- Total volume = **8.322 m³;**
- Pump-up time = **4.021 min**;
- Initial pressure P_1 = **0.5 kg/cm²** ;
- Final pressure P_2 = **7.03 kg/cm²;**
- Atmospheric pressure P_0 = **1.033 kg/cm².abs**.

$$Q = FAD\ (m^3/min) = \frac{(P2 - P1) \times \text{Total Volume}}{\text{Atm. Pressure} \times \text{Pump-up time}} =$$

$$Q = FAD\ (m^3/min) = \frac{(7.03 - 0.5) \times 8.322}{(1.033 \times 4.021)} = \mathbf{13.08\ m^3/min\ (462\ CFM)}$$

Note: The capacity fall rating is approximately **13% (referent to 14.75 m³/min)**. This indicates the compressor performance is below and **not delivering air** according to its specification.

Remember:

- **1 m³/min** = *CFM x 35.31;*
- **1 m³/h =** Nm³/h x ~1.23;
- **1 SCFM =** CFM x ~1.17;
- ***1 CFM =** 1 m³/h x 1.699.*

Example:

Evaluate **the previous Reciprocating Compressor** capacity, but with data in U.S. units.

To calculate in **imperial units** the equation below is the same previously used with **metric units** to estimate the real capacity of a compressor:

$$Q = \frac{V.\ (P_2 - P_1)}{P_a \times t} =$$

Where**:**

V = Receiver volume (ft^3);
Q = Free air flow (SCFM);
P_a = atmospheric pressure (psia);
P_1 = maximum pressure (psig);
P_2 = minimum pressure (psig);
t = time for receiver from max. pressure (P_1) to min. pressure (P_2), (min.).

So,

- **V** = 294 cu.ft (8.322 m^3);
- $\mathbf{P_1}$ = 7.11 psig (0.5 kg/cm^2.abs);
- $\mathbf{P_2}$ = 100 psig (7.03 kg/cm^2.abs);
- **t** = Time taken to fill receiver from P_1 to P_2 = 4.021 min.;
- $\mathbf{P_o}$ = Atmospheric pressure 14.7 psia (1.033 kg/cm^2.abs).

Then:

$$Q = \frac{294. (100 - 7.11)}{(14.7 \times 4.021)} = 462 \text{ CFM}$$ (corresponding to **13.08 m^3/min**).

Evaluating the Capacity of a Compressor:

1. The air compressor that will be tested for capacity is isolated from the rest of the system, by operating an isolating non-return valve.

2. The compressor drive motor is shut-off.

3. The receiver connected to this air compressor is emptied.

4. The motor is re-started.

5. The pressure in the receiver begins to rise. Initial pressure, let´s say **2 kg/cm^2 (28.44 psi)**, is noted. **The stopwatch is started** at this moment.

6. The **stopwatch is stopped** when receiver pressure has risen to, let´s say, **8 kg/cm^2 (113.78 psi).**

7. Time elapsed is noted in **minutes**.

Effects on Storage on Pressure Differential: For storage air in receivers there must be a pressure differential and an allowable pressure band. Without an allowable pressure differential, there is no storage. The usable pressure differential and the air receiver size determine the available storage. In U.S units, a receiver has a given **volume**, normally measured in **gallons**, except in large sizes, where they are measured in **cubic feet**. There are **7.48 gallons in a cubic foot**. The amount of free air in the receiver depends upon the pressure. Remember: At sea level, the atmospheric pressure is **14.7 psia (0 psig)**.

Example:

If air in the receiver has been compressed to **100 psig**, the absolute pressure is:

P = (100 + 14.7) = **114.7 psia**

So, a **1,000 gallon** receiver **at 100 psig** will have the capacity equivalent of:

$$Q = \frac{(1{,}000 \times 114.7)}{(7.48 \times 14.7)} = 1{,}043 \text{ ft}^3 \text{ of free air (FAD).}$$

Obs.: That´s is why, the storage air in receivers there must be a pressure differential and an allowable pressure band.

Examples:

1) An industrial plant has a **1,000 gallon** air receiver that works with an allowable pressure differential of **10 psi (100 to 90 psig).** The available compressed air in storage would be:

$$Q = \frac{V \times (P_2 - P_1)}{(7.48 \times P_a)} =$$

$$Q = \frac{1{,}000.\ (100 - 90)}{(7.48 \times 14.7)} = \mathbf{91\ ft^3}$$

2) The **time required** for the pressure to fall from **100 to 90 psig** is proportional to the rate of demand (presuming no supply during the demand event). For a demand rate of **200 CFM** of free air, the time would be:

$$t = \frac{V.\ (P_2 - P_1)}{(Q \times P_a)} =$$

Where:

V = Receiver volume = 134 ft³;
P_1 = Initial pressure = 100 psig;
P_2 = Final pressure = 90 psig;
Q = Free Air Delivery = 200 CFM;
P_a = Atmospheric pressure = 14.7 psia;
t = Time in minutes = (?).

$$t = \frac{134 \times (100 - 90)}{(200 \times 14.7)} = \mathbf{0.456\ minutes}$$

Note: Similarly, the pressure would take **one minute** to fall from **100 psig to 78 psig**. It´s necessary to be careful not to raise the compressor operating pressure in order to increase storage. Considering the receiver in **gallons** (U.S. units) the formula for computing **CFM based on pump-up time** is as follows:

$$CFM = \frac{Gallons \times (P_1 - P_2)}{7.48 \times P_0 \times t}$$

Where:

P_1 = Air receiver final pressure, psig;
P_2 = Air receiver starting pressure, psig;
P_0 = Atmospheric pressure (14.7 psia);
t = Time to fill the air receiver, min.;
Gallons = Air receiver capacity, gal.

Note: The unit of measurement of **P_2 and P_1 is psig** or whatever initial pressure can be chosen, can be between **100 psig - 50 psig**.

Example:

Evaluate **the previous Reciprocating Compressor** capacity with the following data:

- **V** = 2198.4 gallons (8.322 m^3);
- $\mathbf{P_1}$ = 7.11 psig (0.5 kg/cm^2.a);
- $\mathbf{P_2}$ = 100 psig (7.03 kg/cm^2.a);
- **t** = Time taken to fill receiver from P_1 to P_2 = 4.021 min.;
- $\mathbf{P_o}$ = Atmospheric pressure 14.7 psia (1.033 kg/cm^2.abs.).

$$\mathbf{CFM} = \frac{(100 - 7.11) \times 2198.4}{7.48 \times 14.7 \times 4.021} = \mathbf{462\ CFM}$$

Relationship - Fans and Air Compressors: To calculate a **fan CFM** it´s necessary to know the **fan blade diameter and pitch** (blade tilt or attack angle) as well, the RPM (Revolution/minute). Designers of air-movement mechanisms consider the basic relationship between fans or impellers speed in **RPM** and the volumetric air flow in **CFM** (cubic feet per minute) in air compressors. While these machines have different operating principles, their common characteristic is that each revolution of their spinning input shaft will result in a certain amount of volumetric air (or gas) flow when they are operating.

Example:

Let us take a small three-bladed plastic fan with a **1-foot diameter** (0.5 ft. radius) and an **8-inch** effective **pitch**. The fan is running at **1,200 rpm**. Calculate the linear velocity and respective flow of the air through the running fan:

Solution: This means that **each revolution** of the running fan blows **1-foot-diameter of column of air** coming through the fan with **8 inches pitch,** without efficiency losses.

Column of air = 8 inches x 1200 rpm = **9600 inches per minute;**
Column of air = 9600 inches per minute/12 feet = **800 feet per minute;**

To calculate the **CFM** (volumetric flow of air) **at 1,200 rpm**:

CFM = (3.1416) x fan radius² (0.5^2 feet) x the column length in feet.
CFM = (3.1416) x 0.25 feet x 800 feet **= 628.32 CFM at 1200 rpm.**

Example:

A Reciprocating Compressor works with an air net displacement of **10 in³** per revolution, and works normally **at 600 rpm**. The **cubic feet per minute of compressed air** have a **10 - 1 compression ratio**. Define the compressor application and calculate the incoming volumetric air flow.

Solution: If the compressor takes in **10 in³ of air** for each revolution, then:

CFM = 600 RPM x 10 in³/1728 = 3.47 CFM.

Since the compression ratio is **10 - 1**, the compressed air is: **CFM =** 3.47/10 = **0.347 CFM**.

Remember: 12^3 inches = **1728 ft**.

High Pressure Off-line Storage: In some compressed air systems, there is a very large demand that happens frequently. Large air compressors can often take several minutes before effective work.

Example:

A plant had three **3,000 CFM** centrifugal air compressors. **Two** of them supplied the plant; the **third** was left on **standby**. A few times longer, one compressor shut down unexpectedly. This caused the operating pressure to fall below the required **80 psig**, resulting in equipment shutdowns that crippled the plant processes for hours.

- One option was to install **another air storage** to cover the unexpected loss of one air compressor and other was to design a high pressure spare air storage, and metering it into the system. Normally, it took approximately **one minute** for the **3,000 CFM standby** centrifugal compressor to start producing air. **The startup time** for a compressor with a **10 psig** differential would require:

$$Q = \frac{1\text{ min} \times 3000\text{ CFM} \times 14.7\text{ psia}}{10\text{ psig}} = \mathbf{4{,}410\ ft^3}$$

$$Q = 4{,}410\text{ ft}^3 \times 7.48 = \mathbf{32{,}987\ gallons}.$$

- The manufacturer elected to **use high-pressure off-line** storage. Then, a small, high-pressure air compressor capable of providing **300 psig** was installed, as well as, another **1,500 gallon air receiver** and the required accessories to reduce the pressure to **80 psig**, then:

$$V_1 = \frac{1500\text{ gallon}}{7.48} = \mathbf{200.53\ ft^3}$$ **-** Using the formula: $$Q = \frac{V.(P_2 - P_1)}{P_a \times t} =$$

$$V_2 = \frac{200.53 \times (300\text{ psig} - 80\text{ psig})}{14.7\text{ psia} \times 1\text{ min.}} = \mathbf{3001\ ft^3}$$

Note: The system **worked** as designed and shutdown problems were eliminated. A complete understanding of a compressed air system is essential to implementing storage solutions.

VII. LEAK DETECTION:

Leaks can be a significant source of **wasted energy** in an industrial compressed air system, sometimes wasting 20-30% of a compressor's output. A number of measurements may be taken to determine the average time it takes **to load and unload** the compressor. The air leaks will cause the compressor to have a constant **cycle on and off**, as the pressure drops from air escaping through the system. The total leakage (percentage) can be calculated as follows:

$$\textbf{Leakage (\%)} = \frac{(T \times 100)}{(T + t)} =$$

Where:

T = On-load time, min;
t = Off-load time, min;

Leakage test in air compressed systems can be calculated as follows:

$$\textbf{Leakage (FAD)} = \frac{V \times (P1 - P2)}{(T \times 14.7) \times 1.25} =$$

Where:

V = Total volume, ft³;
P1 = Inlet Pressure, psig;
P2 = Exhaust pressure, psig;
T = Load time, min.;

Note: The **1.25 factor** corrects the leakage and normalize the pressure differential in a system.

Ultrasonic Acoustic Detector: Since **air leaks are almost impossible** to see, other methods must be used to locate them. The best way to detect leaks is to use an **ultrasonic acoustic detector**, which can recognize the high-frequency hissing sounds associated with air leaks. These portable units consist of directional microphones, amplifiers, and audio filters, and usually have either visual indicators or earphones to detect leaks. A simpler method is to apply soapy water with a paint brush to suspect areas. Although reliable, this method can be time consuming.

Lubrication: Lubricants are made to cool, seal, and lubricate machine moving parts for enhanced performance and avoid wear. Important considerations for compressor lubricants include proper application and compatibility with downstream equipment, including piping, hoses, and seals. A **lubricator** may be installed near a point of use to lubricate items, such as pneumatic tools. The lubricator generally comes combined with a **filter and a pressure regulator** to form what is commonly called as **FRL** (filter-regulator-lubricator). The lubricant should be specified by equipment manufacturer to the point-of-use.

VIII. PIPING PRESSURE DROP:

Excessive pressure drop will result in poor results in lower operating pressure, at the points where excessive air energy are necessary. Any type of obstruction or roughness diagnosis will cause resistance to air flow at different points of the system, and requires correction measurements, but sometimes are difficult to identify the cause of pressure drops.

Piping layout on industrial plants should be reasonably complete, with good checking for space, clearances, interference, and pressure drops in equipment that require working air. When pressure drop **tables** are used, it is necessary to find the equivalent length of the pipeline, from the compressor to the farthest point of the piping system.

Pressure Drop Measurements: Measuring the actual pipe length is the first step. In addition, the effects of accessories and connections must also be considered.

- Determine the actual pressure drop that will occur only in the piping system. Generally accepted practice is to allow 10% of the proposed system pressure for pipe friction loss.
- It is a good practice to oversize the piping to allow for future growth, as well as, the addition of process equipment.
- Size the piping using the appropriate charts, having calculated the volume per minute and the allowable friction loss in each section of the piping being sized.
- The temperature used to calculate the friction loss is **60ºF (16ºC)**.
- Pressure drop in compressed air can be expressed as equivalent length in meters of pipeline.
- Equivalent length in metric units:

Table VII – Equivalent Length of Piping (8 bar -116 psi) - and 5% pressure drop:

Flow rate			Equivalent length									
			164 ft	328 ft	492 ft	984 ft	1640 ft	2460 ft	3280 ft	4265 ft	5249 ft	6561 ft
Nm3/h	Nl/min	cfm	50 m	100 m	150 m	300 m	500 m	750 m	1000 m	1300 m	1600 m	2000 m
10	167	6	16,5	16,5	16,5	16,5	16,5	16,5	16,5	25	25	25
30	500	18	16,5	16,5	16,5	25	25	25	25	25	25	40
50	833	29	16,5	25	25	25	25	25	40	40	40	40
70	1167	41	25	25	25	25	40	40	40	40	40	40
100	1667	59	25	25	25	40	40	40	40	40	40	63
150	2500	88	25	40	40	40	40	40	40	63	63	63
250	4167	147	40	40	40	40	63	63	63	63	63	63
350	5833	206	40	40	40	63	63	63	63	63	63	76
500	8333	294	40	40	63	63	63	63	63	76	76	76
750	12500	441	40	63	63	63	63	76	76	76	76	100
1000	16667	589	63	63	63	63	76	76	76	100	100	100
1250	20833	736	63	63	63	76	76	100	100	100	100	100
1500	25000	883	63	63	63	76	100	100	100	100	100	100*
1750	29167	1030	63	63	76	76	100	100	100	100	100*	100*
2000	33333	1177	63	76	76	100	100	100	100	100*	100*	100*
2500	41667	1471	63	76	76	100	100	100*	100*	100*	100*	100*
3000	50000	1766	76	76	76	100	100	100*	100*	100*	100*	100*
3500	58333	2060	76	76	100	100	100*	100*	100*	100*	100*	100*
4000	66667	2354	76	100	100	100	100*	100*	100*	100*	100*	100*
4500	75000	2649	76	100	100	100*	100*	100*	100*	100*	100*	100*
5000	83333	2943	76	100	100	100*	100*	100*	100*	100*	100*	100*
5500	91667	3237	100	100	100	100*	100*	100*	100*	100*	100*	100*
6000	100000	3531	100	100	100*	100*	100*	100*	100*	100*	100*	100*

Obs.: *Pressure drop more than 5%.

The pressure in a pipeline can also be calculated using following formula **(metric units)**:

Pressure Drop, kg/cm² =

$$\text{Pressure Drop, kg/cm}^2 = \frac{7.57 \times (Q^{1.85}) \times L \times (10^4)}{(d^5) \times P} =$$

Where**:**

Q = Air flow in m³/min (FAD);
L = Length of pipeline (m);
d = inside diameter of pipe (mm);
P = Initial pressure, kg/cm².

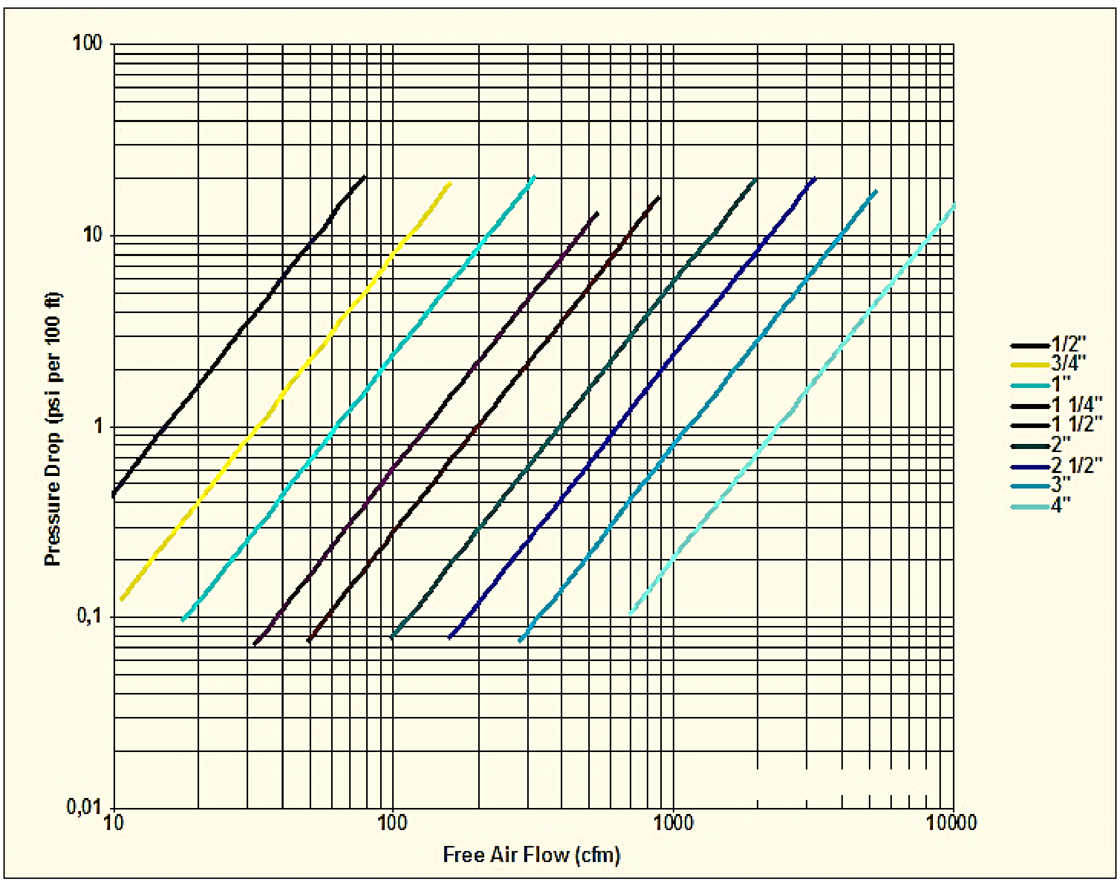
The points to be kept in mind while designing a distribution system are:

- Low pressure drop between the compressor installation and the end use point;
- Minimum leakage;
- Minimum number of joints, bends, fillings in the pipeline;
- Proper design and layout of the pipeline

Factors to be considered about pressure drops:

- A general guide for selection of **pipe sizes** according to general standards:
- The pressure drop **should not exceed 45 psi (~3 kg/cm²)** at the farthest end of the line. For large pipelines, the **pressure drop up to 8 psi ~0.5 kg/cm²)** may be acceptable.
- Typical instrument air pressure in a chemical, manufacturing plant or oil plants, is approximately **6 bar (90 psi)**, but maximum ratings are seldom over **8 bar (120 psi)**.
- There are other uses and facilities, such as hospitals, that may require instrument air pressure up to **10 bar (150 psi)** and tighter specifications for air quality.

Recommended pressure drops with pipelines from ½" up to 4" - 100 psi:



The compressor room piping header should be sized so that the air velocity does not exceed **20 ft/s to 25 ft/s**, thus allowing a future expansion.

Distribution header piping leaving the compressor room should be sized to allow an air velocity not to exceed **30 ft/sec**, to minimize pressure drop. The required pipe diameter and length of the pipe line are given in report if the pressure drop exceeds the allowed pressure drop.

Some systems operate at an elevated pressure of **100 psi** at full load when the machinery and tools can **operate efficiently** at a lower air pressure of **90 – 70 psi**. The **extra 10 – 30 psi** would be responsible for approximately 5% -15% of the plant's increased energy costs.

Table VIII – Recommended Pipe Sizes

Recommended piping sizes with sch. *40* steel pipes, air pressure *100 psi (6.9 bar)*							
Free Air Flow *(cfm)*	**Length of Pipe *(feet)***						
	25	**50**	**75**	**100**	**150**	**200**	**250**
5	1/2	1/2	1/2	1/2	1/2	1/2	1/2
10	1/2	1/2	1/2	1/2	1/2	1/2	1/2
15	1/2	1/2	1/2	1/2	3/4	3/4	3/4
20	1/2	1/2	3/4	3/4	3/4	3/4	3/4
25	1/2	3/4	3/4	3/4	3/4	1	1
30	1/2	3/4	3/4	3/4	1	1	1
35	3/4	3/4	3/4	3/4	1	1	1
40	3/4	3/4	3/4	1	1	1	1
45	3/4	3/4	1	1	1	1	1 1/4
50	3/4	1	1	1	1	1 1/4	1 1/4
55	3/4	1	1	1	1 1/4	1 1/4	1 1/4
60	3/4	1	1	1	1 1/4	1 1/4	1 1/4
65	3/4	1	1	1	1 1/4	1 1/4	1 1/4
70	3/4	1	1	1 1/4	1 1/4	1 1/4	1 1/4
75 - 80	1	1	1	1 1/4	1 1/4	1 1/4	1 1/4
90 - 100	1	1	1 1/4	1 1/4	1 1/4	1 1/4	1 1/2

Rules Of Thumb:

1. Air compressors are normally rated to deliver **4 to 5 CFM per HP at 100 psig** discharge pressure.

2. A **50 HP** compressor rejects approximately **126,000 BTU per hour** for heat recovery.

3. The water vapor content at **100¼ °F** of saturated compressed air is about **2 gallons per hour** for each **100 CFM** of compressor capacity.

4. Every **20¼ °F** temperature drop in saturated compressed air at constant pressure, **50%** of the water vapor condenses to liquid.

6. At **100 psig, every 20¼ °F** increase in saturated air the temperature, doubles the amount of moisture in the air.

5. Most water-cooled after coolers will require about **3 GPM per 100 CFM** of compressed air at discharge air temperature at **100 psig**.

6. Every **2 psig change** in pressure equals **1% change in HP**.

Piping: Most compressed air systems use **galvanized, black steel or stainless steel piping** schedule 80 for sizes 2 inches and smaller and schedule 40 for sizes over 2 inches. The **schedule 40 piping** is suitable for pressures in the 175 psig range. Pipe fittings shall be galvanized or black steel or stainless steel, to match piping used.

Copper compressed air piping or tubing shall be Type K or Type L. When copper pipe or tubing is used, brazed joints shall be used for connections. **Fiberglass** reinforced plastic (FRP) may also be used within the following limitations:

• 150 psig maximum pressure, up to 200°F.
• 75 psig maximum pressure, up to 250°F.

The **PVC piping** is relatively inexpensive, easy to install, lightweight, and corrosion resistant. However, PVC is not recommended due it has one major drawback, it is brittle. An inadvertent impact could cause the piping to shatter, endangering surrounding personnel.

Sizing of compressed air piping is based on the allowable velocity of compressed air in the pipeline, keeping a check on the pressure drop. In compressed line **if the pressure drop is high**, the operating pressure at the generation end has to be increased to match with the requirement. This will result in increased power consumption of the compressor. The **recommended velocity** for interconnecting piping and main headers is **20 to 25 fps**.

Table IV - Approximate discharge **temperatures** (before after cooling) at **80¼ °F** ambient:

Compressor	100 psig	160 psig	200 psig
Single Stage	510	615	-
Two Stages	325	365	395
Rotary (Oil cooled)	180 - 200	190 - 205	200 - 215

Inter and After-Coolers: Intercoolers are heat exchangers that remove the heat of compression between the stages of compression. Intercooling affects the overall efficiency of the machine as mechanical energy is applied to a gas for compression, the temperature of the gas increases.

After-coolers are installed after the final stage of compression to reduce the air temperature. As the air temperature is reduced, water vapor in the air is condensed, separated, collected, and drained from the system. Most of the condensate from a compressor with intercooling is removed in the intercooler(s), and the remainder in the after-cooler.

Useful conversions:

1 m^3/min = 1000 liter/min = 16.7 l/s = 35.31 ft^3/min.
1 kg/cm² = 0.98 bar = 14.22 psi = ~100 kPa = 0.1 mPa
1 kPa = 0.01 bar = 0.145 psi = 0.0102 kg/cm²
1 mPa = 10 bar = 10.2 kg/cm² = 145 psi.

Temperature conversions:

C° = $\frac{F - 32}{1.8}$ =

F° = 1.8C° + 32 =

IX. DENSITY AND ALTITUDE:

Although the concept of density altitude is commonly used to describe the effect on aircraft and engine performance, the underlying property of interest is actually the air density. For example, the lift of an aircraft wing, the aerodynamic drag and the thrust of a propeller blade are all directly proportional to the air density. The down force of a racecar spoiler is also directly proportional to the air density.

Density altitude has been a convenient yardstick for pilots to compare the performance of aircraft at various altitudes, but it is in fact the air density which is the fundamentally important quantity, and density altitude is simply one way to express the air density. The 1976 International Standard Atmosphere (which is used as the basis for these Density Altitude calculations) is **mostly described in metric SI units**.

Air Density Calculations:

To begin to understand the calculation of air density, consider the ideal gas law:

P*V = n*R*T =

Where**:**

P = Pressure;

V = Volume;
n = Number of moles;
R = Gas constant;
T = Temperature.

Density is simply the number of molecules of the ideal gas in a certain volume, in this case a molar volume, which may be mathematically expressed as:

D = n / V =

Where:

D = density;
n = number of molecules;
V = volume.

Then, by combining the previous two equations, the expression for the density becomes:

$$D = \frac{P...}{R \times T} =$$

Where:

D = Density, kg/m^3;
P = Pressure, Pascals;
R = Gas Constant, J/(kg°K) = 287.05 for dry air;
T = Temperature, °K = (°C + 273.15).

Temperature in ºF into Rankine, ºR:

ºR = (ºF + 459.67) or **ºR = (ºF + 460) =**
Table IX - Density and Specific Weights in SI Units:

Table IX - Density and Specific Weights: SI Units:

Temperature - *t* - (ºC)	Density - *ρ* - (kg/m³)	Specif. Weight - *γ* - (N/m³)
0	1.293	12.67
5	1.269	12.45
10	1.247	12.23
15	1.225	12.01
20	1.204	11.81
25	1.184	11.61
30	1.165	11.43
40	1.127	11.05
50	1.109	10.88
60	1.060	10.40
70	1.029	10.09
80	0.9996	9.803
90	0.9721	9.533
100	0.9461	9.278

Table X - Density and Specific Weights: Imperial Units:

Temperature - *t* - (ºF)	Density - *ρ* - $(slugs/ft^3) \times 10^{-3}$	Specif. Weight - *γ* - $(lb/ft^3) \times 10^{-2}$
0	2.683	8.633
10	2.626	8.449
20	2.571	8.273
30	2.519	8.104
40	2.469	7.942
50	2.420	7.786
60	2.373	7.636
70	2.329	7.492
80	2.286	7.353
90	2.244	7.219
100	2.204	7.090
120	2.128	6.846
140	2.057	6.617
160	1.990	6.404
180	1.928	6.204
200	1.870	6.016
300	1.624	5.224

Density Altitude with Relative Humidity: The National Weather Service information shows the hourly **dew point**, **relative humidity** and **altimeter** setting for US locations, in both English and Metric units.

The **International Standard Atmosphere** standard conditions for zero density altitude are:

- Altitude **0 meters (0 ft)**, Air Temperature **15°C (59°F)**, Air Pressure **1013.25 mPa (29.921 in Hg)**;
- Relative Humidity **0 % (0 absolute dew point)**;
- **Standard Sea Level Air Density** = **1.225 kg/m³ (0.002378 slugs/ft³)**.
- **Temperature in ºC into Kelvin, ºK** = **°K = (°C + 273.15) =**

Air density is affected by the air pressure, temperature and humidity. The density of the air is reduced by decreased air pressure, increased temperatures and increased moisture. A reduction in air density reduces the engine horsepower, reduces aerodynamic lift and reduces drag.

The altitude (or elevation) is the geometric altitude above sea level where the altimeter setting, temperature and dew point have been measured. The absolute air pressure is the actual air pressure, not corrected for altitude, and is also called the station pressure. Relative density is the ratio of the actual air density to the standard sea level density, expressed as a percentage.

Example:

At **5050 feet altitude, 95°F, 29.45 inches-Hg** barometric pressure and **40% relative humidity**, the **density altitude** is calculated as **9251 feet**.

X. LINKS AND REFERENCES:

Compressed Air Sizing, Parker Hannifin Training, 2006;
Improving Compressed Air System Performance, US Department of Energy, 2003;
Compressor Handbook, Paul C. Hanlon, McGraw-Hill;
Displacement Compressors Acceptance Tests, Standard ISO 1217;
CAGI - Compressed Air and Gas Institute.

For additional technical information related to this subject, please visit the following websites:

1. The Compressed Air and Gas Institute: *www.cagi.org*
2. ASME Boiler and Pressure Vessel Code: *www.asme.org*.
3. Energy Tips – Compressed Air Tips Sheet: *www.eere.energy.gov*
4. OSHA Technical Manual: *www.osha.gov*
5. http://www.engineeringtoolbox.com;
6. http://www.not2fast.com/turbo/compression/compression.shtml.