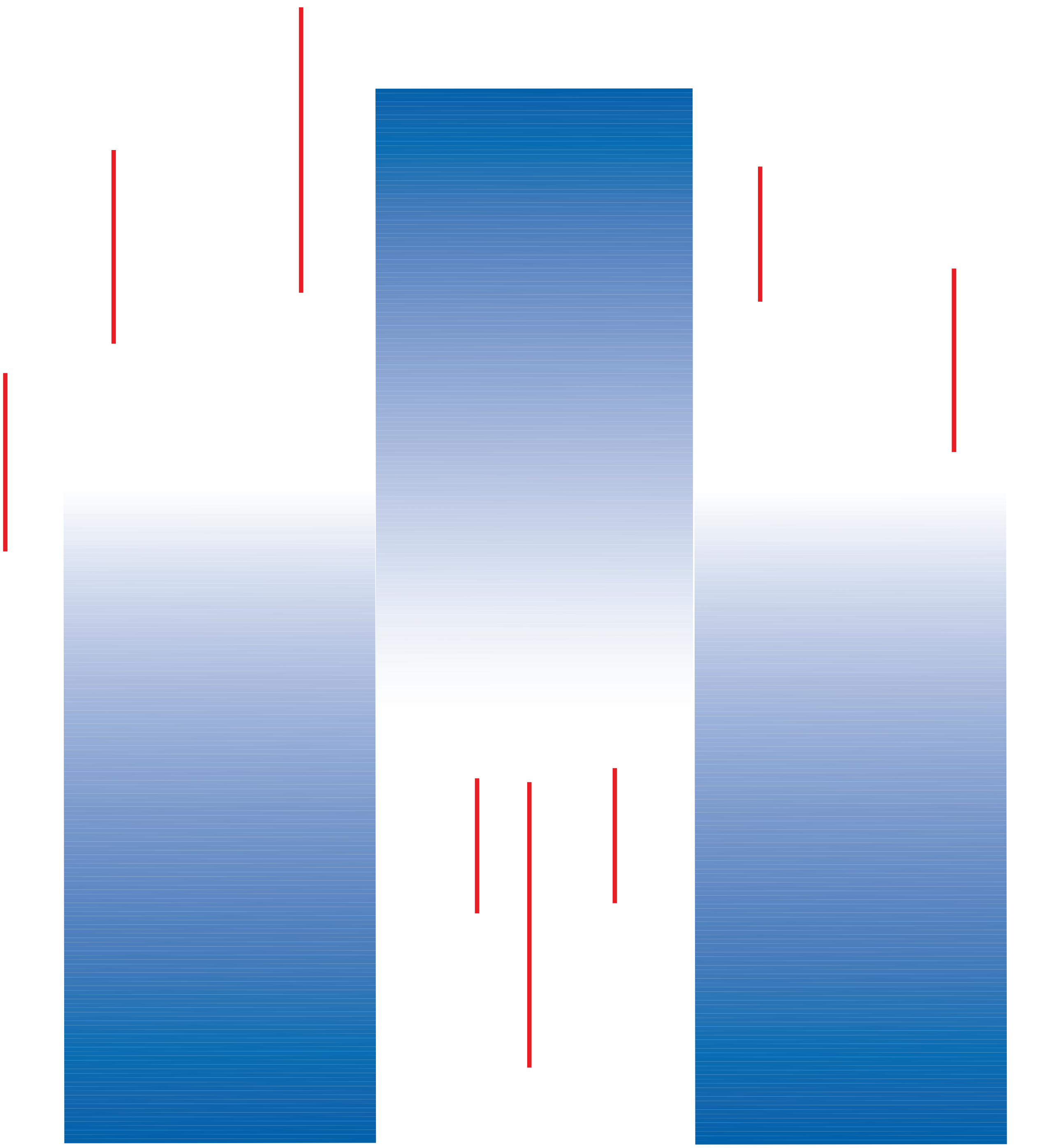
VACUUM AND PRESSURE SYSTEMS
HANDBOOK

(revised electronic edition)



VACUUM AND PRESSURE SYSTEMS
HANDBOOK

Gast Manufacturing, Inc
A Unit of IDEX Corporation
PO BOX 97
2300 M-139
Benton Harbor, MI 49023-0097
Ph: 616/926-6171
Fax: 616/927-0860
http://www.gastmfg.com/
marketing@gastmfg.com

INTRODUCTION

This handbook examines two interrelated segments of pneumatic power: pressure and vacuum. That is, the development and utilization of air pressure and vacuum to meet specific work needs.

In terms of system equipment, this translates largely (but not exclusively) into the proper selection and sizing of commercial air compressors and vacuum pumps. The focus here is on smaller units–those appropriate to powering an individual machine or, at most, a small shop.

The basic principles are equally applicable to the much larger units supplying compressed air or vacuum as a utility to an entire plant, but this book makes no attempt to deal with the practical aspects of selecting and utilizing these large–sometimes very large–machines. The emphasis throughout is on practical approaches to designing small pneumatic power systems for today's complex and sophisticated needs.

While this handbook treats vacuum and pressure systems in separate sections, keep in mind that some applications require both vacuum and pressure. For example, in thermoforming plastic cups, first air pressure, then vacuum is required. Instead of installing separate systems, a combination compressor/ vacuum pump can provide pneumatic power for both functions.

When a unit is used to provide pressure and vacuum simultaneously rather than sequentially, however, the loads must be carefully balanced. Separate units are usually preferred for this type of application.

Handbook Sections

Whatever the application, the correct pneumatic power system can help meet today's increasing demands for greater machine productivity, reliability and operator safety and convenience. The information in this handbook will help you attain these objectives.

The basic concepts of pressure and vacuum are covered in Section 1. Separate sections are devoted to the selection and operation of air compressors (Section 11) and vacuum pumps (Section IV).

Sections III and V cover accessories, work devices, and overall considerations for pressure and vacuum systems, respectively. Section VI covers combination compressor/vacuum pump systems. Section VII includes some representative problem/solution applications.

The appendix summarizes gas laws and related data. It also includes tables giving data on pressure loss due to friction, pipe bends and components, and on air flow through various-size orifices. A glossary appears at the end of the volume.

Contents

BASIC CONCEPTS ----- VACUUM AND PRESSURE

Pneumatics and Pneumatic Power

"Pneumatics" is the general term used to describe the mechanics of gases. And "pneumatic power" can be defined as: production or control of mechanical outputs for useful work by means of a pressurized gas in a closed circuit. In sim- pler
language, Fluid power provides a very reliable "muscle" function-ranging from', something as simple as pressure for blowing an air horn to vacuum for lifting a huge metal workpiece into a precise position on a worktable.

Although the definition allows for any gas, air is almost always used in practi- cal industrial systems. Only air is considered in the remainder of the book.

Pneumatic power is one of the two branches of "fluid power." The other is "hydraulic power," in which the working fluid is a liquid. In pneumatic systems, unlike hydraulic systems, the pressures can be positive or negative.

In pneumatic systems, the pressure differentials necessary to do work are produced by air compressors. They push more air into the system, increasing the pressure above that of atmospheric air.

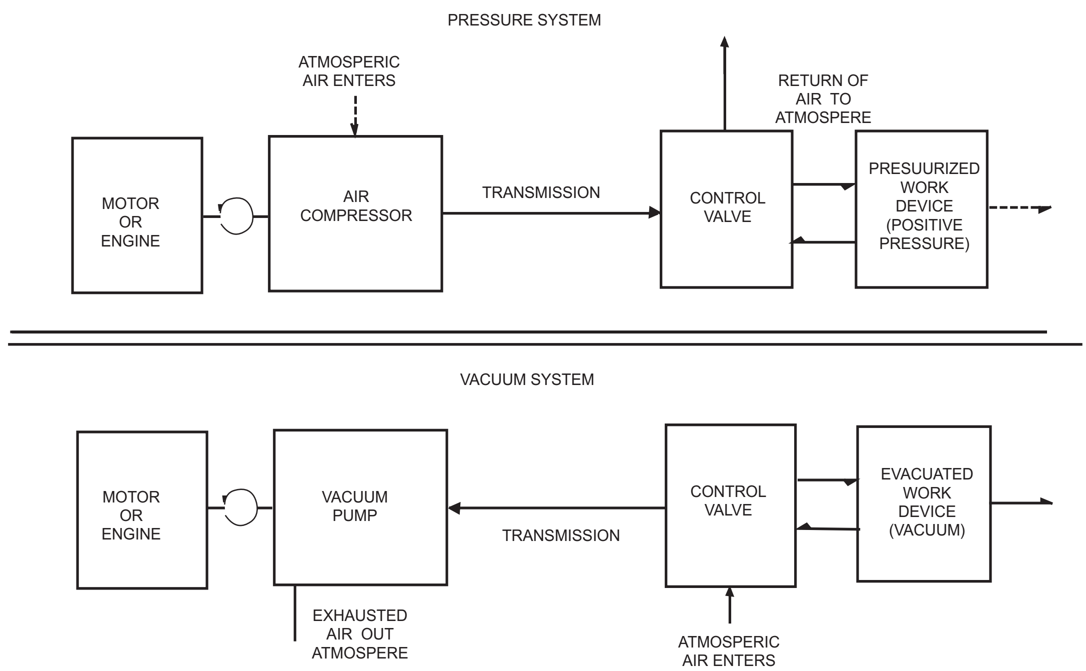
In vacuum systems, the pressure differentials are produced by vacuum pumps. These pull air out of the system, decreasing the pressure below atmo- spheric.

Even though we sometimes refer to vacuum as negative pressure, this can be misleading. In an absolute sense, pressure is always positive. Pressure can be "negative" only in relation to some other, higher, pressure. But since we are constantly surrounded by the atmosphere, it's often natural to describe below atmospheric pressures as negative.

Compressed Air and Vacuum Systems- Fig. 1 compares the basic operation of compressed air and vacuum systems. In both systems a prime mover such as an electric motor or gasoline engine operates an air compressor or vacuum pump, converting electrical or chemical energy into pneumatic energy.

Note how atmospheric air enters and leaves each system, and the direction of the arrows indicating transmission of pneumatic energy. At the end of each system, an appropriate control valve and work device (air cylinder, air motor, etc.) converts the pneumatic energy into useful mechanical force or power. In either case, the air is a working fluid that is unchanged over a complete operat- ing cycle.

Figure 1



Simplified comparison of pressure and vacuum systems, showing routing of atmospheric air into, through, and out of each.

Although the pressure differentials generated are exactly opposite in vacuum and pressure systems, there is considerable similarity in the equipment used. Air compressors and vacuum pumps use the same basic mechanisms.

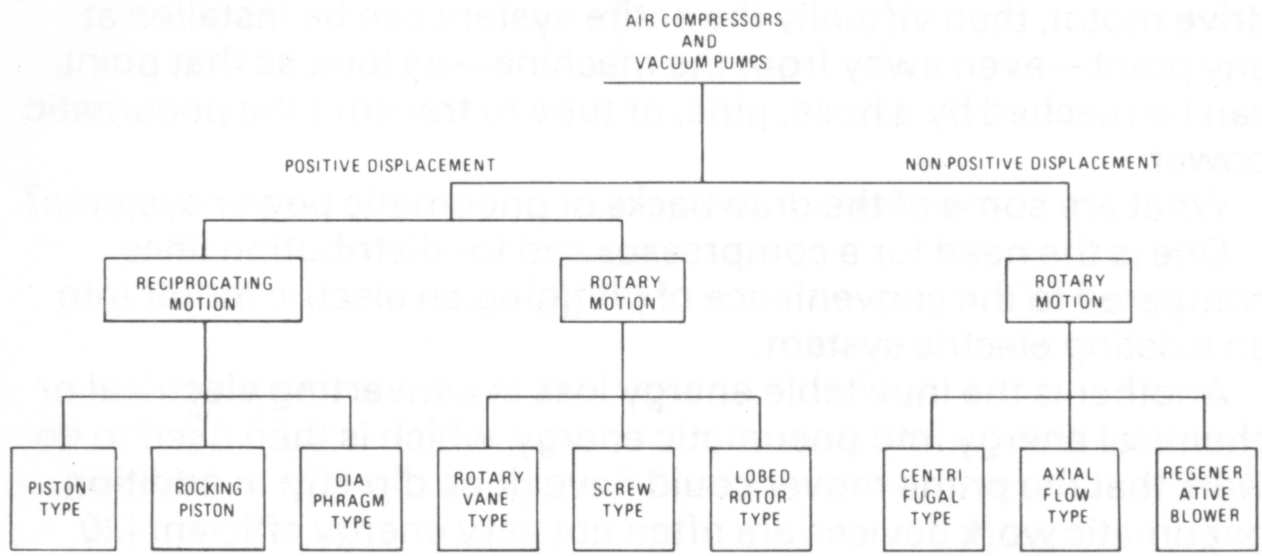
In principle they can be considered the same machine but with the inlet and outlet ports reversed. That is, each takes air at a lower inlet pressure and transforms it to air at a higher outlet pressure. But in a compressor the inlet is usually at atmospheric pressure and the outlet is connected to the system; in a vacuum pump it is the outlet that is at atmospheric pressure.

Sometimes compressors and vacuum pumps are assembled in part from the same interchangeable stock of components. Valving, porting, and oilers usually differ, however.

Another major difference is in the drive power needed. Depending on its pressure rating, an air compressor may require from 150 to 400 percent more power than a vacuum pump of the same open-capacity rating.

System Categories – A clear understanding of the basic types of compressor and vacuum pump systems and their relationships should be helpful. These are summarized in Fig. 2. Keep in mind that, in all cases, a power-driven device transforms air at some initial intake pressure to air at a higher outlet pressure.

Figure 2



Hierarchy of air compressor and vacuum pump types

Basic Advantages

Why are pneumatic power systems so popular in such a wide range of work functions? Electronic systems certainly have a much faster response to control signals. Mechanical systems can be more economical. Hydraulic systems can be more powerful.

The answer lies in the unusual combination of advantages pneumatic systems offer. A basic advantage is their high efficiency. For example, a relatively small compressor can fill a large storage tank to meet intermittent high demands for compressed air. Unlike hydraulic systems, no return lines are required.

Other advantages include: high reliability, mainly because of fewer moving parts; compactness; forces, torques and speeds readily variable over a widely useful range; easy control and coordination with other machine/system functions; low cost; easy installation and maintenance; and the availability of a wide range of standard sizes and capacities.

Another, often decisive, advantage in some applications is that air devices create no sparks in explosive atmospheres. They can also be used under wet conditions with no electrical shock hazard.

It is often advantageous to add pneumatic power to machines that have electricity as their primary power source. This may be done to economically provide supplementary functions such as automatic clamping, locking, closing, opening, etc., of various components or devices. The design problems involved are usually not difficult to solve, and equipment selection procedures are simple and straightforward. Installation is simple, too.

When the air compressor or vacuum pump is driven by a power takeoff from the machine it is being added to, the mounting location may be critical (although accessory air components may be placed almost anywhere). But if the unit is provided with its own drive motor, then virtually the entire system can be installed at any point–even away from the machine–as long as that point can be reached by a hose, pipe, or tube to transmit the pneumatic power.

What are some of the drawbacks of pneumatic power systems?

One is the need for a compressor and for distribution lines, compared to the convenience of plugging an electric motor into an existing electric system.

Another is the inevitable energy loss in converting electrical or chemical energy into pneumatic energy, which is then used to do work that the prime mover could have done directly. In addition, pneumatic work devices are often not very energy efficient (20 percent efficiency is typical of air motors, for example). But in variable-load applications, this is offset by pneumatic devices drawing only the power actually needed. Most electric motors, by contrast, draw almost the same power regardless of load.

And, of course, reasonably sized pneumatic devices cannot exert the forces and torques that hydraulic devices can. In most applications, a horsepower rating somewhere in the tens represents the crossover point at which the increasing size and cost of pneumatic devices begin to exceed the basic cost of a hydraulic power generation and transmission system.

In summary, then, electric, hydraulic, and pneumatic systems each have their place. But the advantages of pneumatic power make it the system of choice in many applications.

Pressure Levels and Terminology

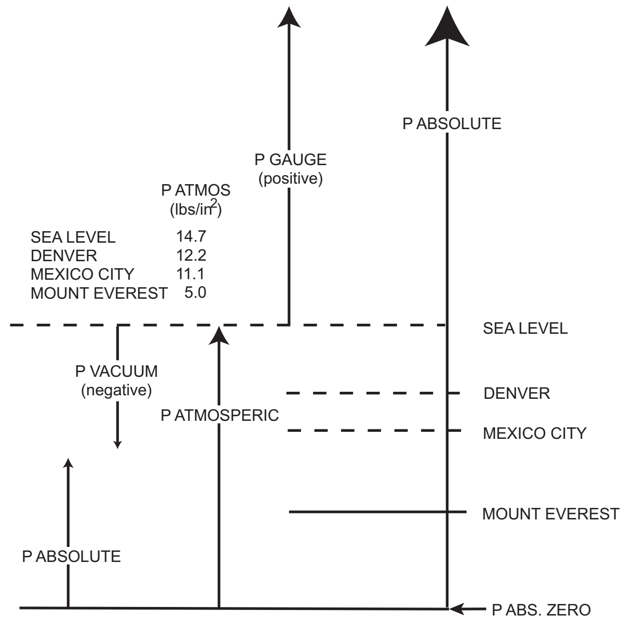
Fig. 3 summarizes the basic relationships and definitions needed to understand the pressure terminology used in this handbook.

Atmospheric Pressure–The atmosphere that surrounds the earth can be considered a reservoir of low-pressure air. Its weight exerts a pressure that varies with temperature, humidity, and altitude.

For thousands of years, air was considered weightless. This is understandable, since the net atmospheric pressure exerted on us is zero. The air in our lungs and the blood in our cardiovascular system has an outward pressure equal to (or perhaps slightly greater than) the inward pressure of the outside air. Since we feel no pressure, we are unaware of the air's weight.

The weight of the earth's atmosphere pressing on each unit of surface constitutes atmospheric pressure, which is 14.7 psi (1101,300 Pa or 0.1013 MPa) at sea level. This pressure is called one atmosphere. In other commonly used units, one

Figure 3



Summary of basic pressure measurement relationships.

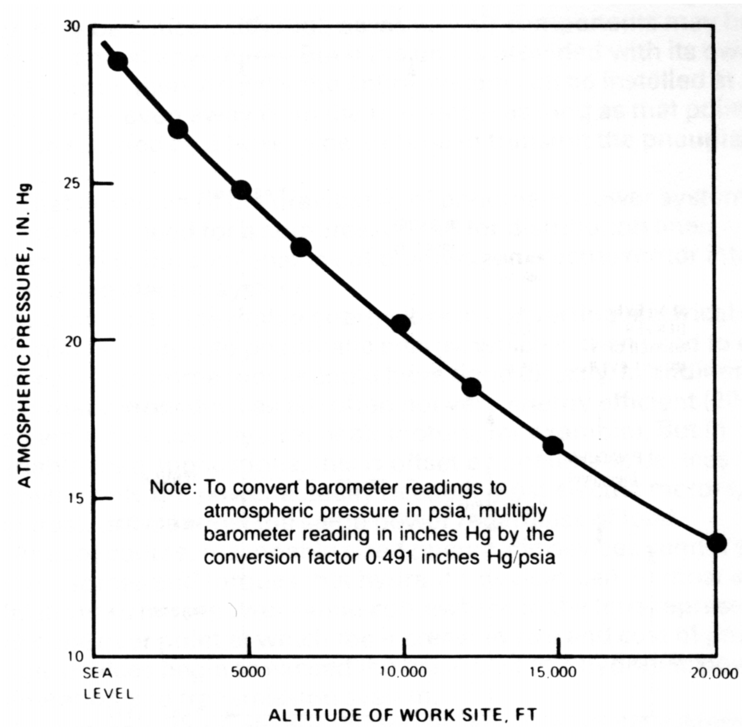
atmosphere equals 29.92 inches of mercury (in. Hg), 760 mm Hg (or 760 torr), and 1.013 bar 0 bar= 0.1 MPa).

Since atmospheric pressure results from the weight of the overlying air, it is less at higher altitudes. As Fig. 3 shows, atmospheric pressure in Denver, Colorado (altitude 5,280 feet), is only 12.2 psi. And in Mexico City, Mexico (altitude 7,800 feet), it is 11.1 psi. On top of Mount Everest, the pressure has fallen to one-third of an atmosphere. Fig. 4 shows this in a different way.

Atmospheric pressure also varies from time to time at a single location, due to the movement of weather patterns. While these changes in barometric pressure are usually less than one-half inch of mercury, they need to be taken into account when precise measurements are required.

Gauge Pressure–Atmospheric pressure serves as a reference level for other types of pressure measurements. One of these is "gauge pressure."

As Fig. 3 shows, gauge pressure is either positive or negative, depending on its level above



Graph shows the effect of altitude on atmospheric pressure. For each thousand feet of elevation, atmospheric pressure In reduced by approximately 1 in. Hg.

or below the atmospheric pressure reference. For example, an ordinary tire gauge showing 30 pounds (actually, 30 psi) is showing the excess pressure above atmospheric. In other words, what the gauge shows is the difference between atmospheric pressure and the pressure of the air pumped into the tire. Gauge pressures can be either positive (above atmospheric) or negative (below atmospheric). Atmospheric pressure represents zero gauge pressure.

Absolute Pressure – A different reference level is used to obtain a value for "absolute pressure." This is pressure measured above a perfect vacuum. It is composed of the sum of the gauge pressure (positive or negative) and the atmospheric pressure. Where there might be confusion, gauge and absolute pressures are distinguished by adding the letter "g" or "a," respectively, to the abbreviation for the units ("psig" or "psia").

To obtain the absolute pressure, simply add the value of atmospheric pressure (which averages 14.7 psi at sea level) to the gauge pressure reading. To find the current value of atmospheric pressure in psia at a given location, multiply the barometer reading in in. Hg by 0.491. This conversion factor arises from the fact that a cube of mercury with one inch sides weighs 0.491 pound and thus exerts a pressure of 0.491 psi.

Using the simple tire pressure example, the absolute pressure –including the atmospheric pressure–exerted by the air within the tire is 44.7 psia (30 psig plus 14.7 psi). Thus, the absolute pressure is 14.7 psi more than would be read on a tire-pressure gauge. Absolute pressure must be used in virtually all calculations involving pressure ratios.

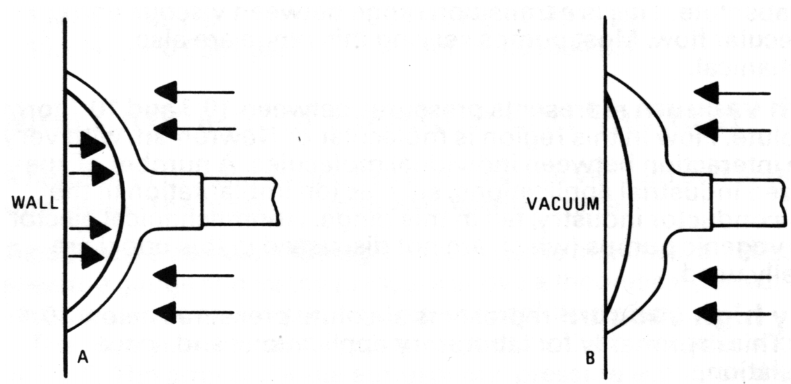
Vacuum–Vacuum is a pressure lower than atmospheric. Except in outer space, vacuums occur only in closed systems.

In the simplest terms, any reduction in atmospheric pressure in a closed system may be called a partial vacuum. In effect, vacuum is the pressure differential produced by evacuating air from the system.

To illustrate the basic concept of using the atmosphere to create a vacuum, Fig. 5 shows a simple example: a rubber suction cup pressed against a smooth wall. It remains there because a vacuum has been created.

In drawing A, before the cup is pressed against the wall, the opposing arrows denote balanced atmospheric pressure. There is

Figure 5



In (A), a suction pad is placed against a wall; no vacuum is generated – the atmospheric forces are balanced. In (B), a vacuum isproduced by expelling air from between the pad and the wall; the pad clings to the wall because outside pressure is greater than the inside pressure.

no vacuum yet because the air pressure inside the cup is equal to the air pressure outside. Both values are at atmospheric pressure.

This balance changes when the cup is pressed against the wall. Drawing B shows that a vacuum now exists in the remaining open space. After most of the air has been expelled, partial expansion of the cup leaves less air per unit volume inside the cup than outside. With less thrust against the cup's inside surface, the pressures are now unbalanced. Outside atmospheric pressure forces the cup against the wall. And since pressure is also holding the edge of the cup firmly to the wall, no air can leak in to relieve the partial vacuum inside.

In a vacuum system more sophisticated than a suction cup, the enclosed space would be a valve actuator or some appropriate work device. A vacuum pump would be used to reduce atmospheric pressure in the closed space. The same principle would apply, however.

By removing air from one side of an air-tight barrier of some sort, atmospheric pressure can act against the other side. Just as with the suction cup, this action creates a pressure differential between the closed system and the open atmosphere. The pressure differential can be used to do work.

For example, in liquid packaging (bottling), reducing the pressure in a bottle (the enclosed space) makes the filling operation go much faster because the liquid or other material is literally pulled into the bottle, rather than simply failing by gravity.

Vacuum is usually divided into four levels:

Low vacuum represents pressures above one torr absolute. Flow in this range is viscous, as represented by most common fluids. Mechanical vacuum pumps are used for low vacuum, and represent the large majority of pumps in industrial practice.

Medium vacuum represents pressures between 1 and 10 torr absolute. This is a transition range between viscous and molecular flow. Most pumps serving this range are also mechanical.

High vacuum represents pressures between 10-1 and 10-' torr absolute. Flow in this region is molecular or Newtonian, with very little interaction between individual molecules. A number of specialized industrial applications, such as ion implantation in the semiconductor industry, fall in this range. Nonmechanical ejector or cryogenic pumps (which are not discussed in this book) are usually used.

Very high vacuum represents absolute pressures below 10-1 torr. This is primarily for laboratory applications and space simulation.

Keep in mind that a "perfect" vacuum-that is, a space with no molecules or atoms-is a purely theoretical condition. Only in interstellar space is this condition approached at all closely, and even there a few atoms per cubic meter will be found. In practice, all vacuums are partial.

Units of Pressure/Vacuum Measurement

In the physical sciences, pressure is usually defined as the perpendicular force per unit area, or the stress at a point within a confined fluid. This force per unit area acting on a surface is expressed in metric units as Newtons per square meter (N/m^2) or Pascals (Pa). The corresponding expression in the English system is pounds per square inch (psi); remember that the pound is a unit of weight, or force, not of mass.

However, many other units are still commonly used for pressure and vacuum measurements. This is understandable because each offers specific advantages in some instances.

For example, "standard atmosphere" as a unit of measurement relates directly to a physical feature of the Earth's surface (strictly speaking, only at sea level). This feature may be the most prominent aspect of some situations. And the bar combines a value near one atmosphere with the simplicity of metric units. It is defined as 10^5 Pascals.

In some vacuum applications the most salient fact may be the fraction of air originally present that has been evacuated (percent vacuum). And when mercury manometers were the usual instrument for vacuum/pressure measurements, the directly observable unit–requiring no further calculations–was the length of the mercury column in inches or millimeters. (Torr is the modern name for mm Hg.)

Table 1 shows the equivalences for selected values of the common units. Other values can be calculated using the conversion factors:

1 atm = 14.70 psi	1 bar= 14.50 psi
1 MPa=145 psi	1 in. Hg = 0.4912 psi
1 in. H_20 = 0.0361 psi	1 torr = 0.01934 psi

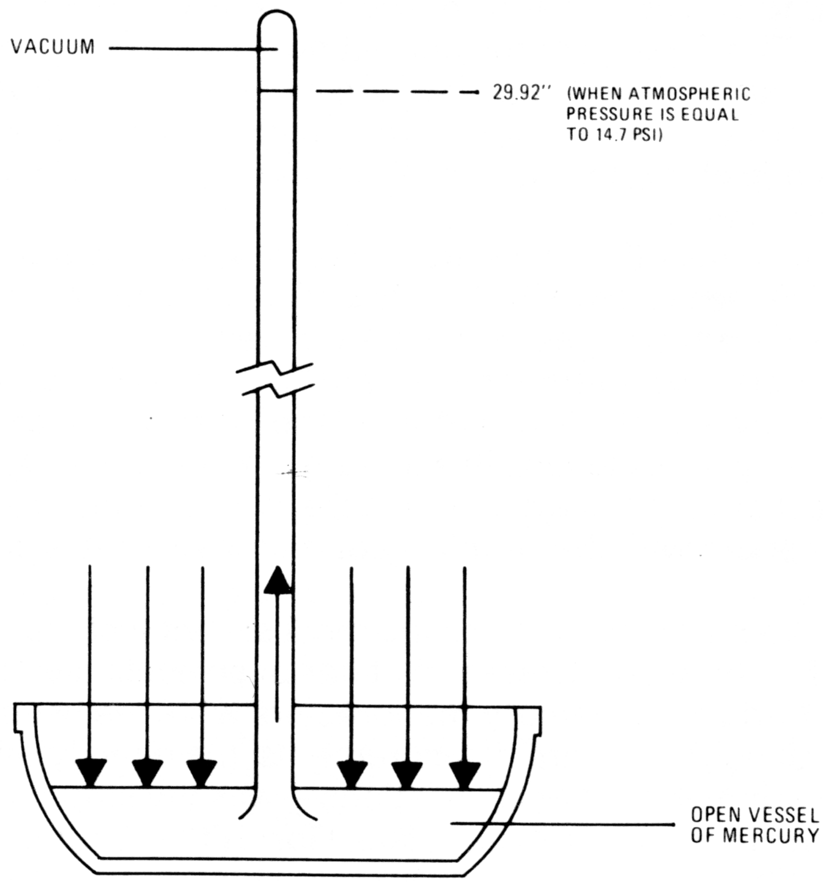
Measurement of Pressure and Vacuum

A number of devices are available to measure vacuum and pressure levels. The most common are described here.

Absolute Pressure Gauge – As its name indicates, an absolute pressure gauge shows the pressure above a theoretical perfect vacuum condition. It thus provides an absolute reading.

The most basic absolute pressure gauge is the barometer shown in Fig. 6. The arrows denote atmospheric pressure acting on the surface of mercury in a dish. This pressure is transmitted in all directions within the body of mercury, including pressure upward into the tube. The height of the column supported this way directly measures the current atmospheric pressure.

There is no inherent requirement that only atmospheric pressure be measured. When the apparatus is arranged so that some other pressure acts on the mercury surface, that pressure can be measured equally well. It does not



A basic mercury barometer. The bottom of the tube is immersed in a pool of mercury that is exposed to atmospheric pressure (arrows). This pushes the mercury up into the tube until the downward pressure of its weight is exactly equal to the pressure exerted by the atmosphere.

matter whether the pressure is above or below atmospheric; the only requirement is that the tube be long enough to accommodate the mercury column.

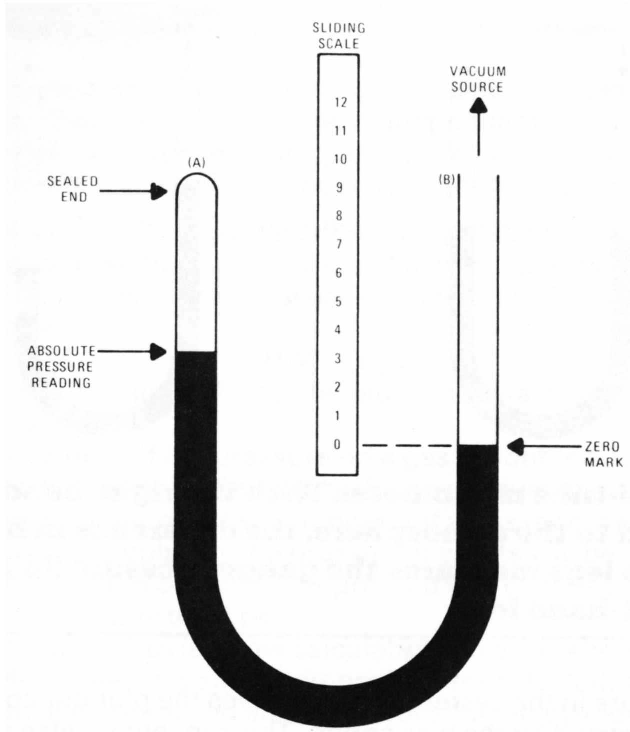
Another type of absolute pressure gauge is used for vacuum measurements only. This gauge (Fig. 7) has the same U shape as the manometer (next page), but leg A is sealed. Mercury fills this end when the gauge is not in use.

When leg B is connected to a vacuum source, the mercury level in leg A is pulled down. The sliding scale is then placed so that the zero mark is opposite the level in leg B. Matching the level in leg A against the scale then gives the absolute pressure directly in in. Hg.

Mercury U-Tube Manometer – A manometer indicates the difference between two pressures. If one is atmospheric pressure, the result is a direct reading of positive or negative gauge pressure.

In its simplest form, the device is a U-tube about half-filled with mercury (Fig. 8)

Figure 7



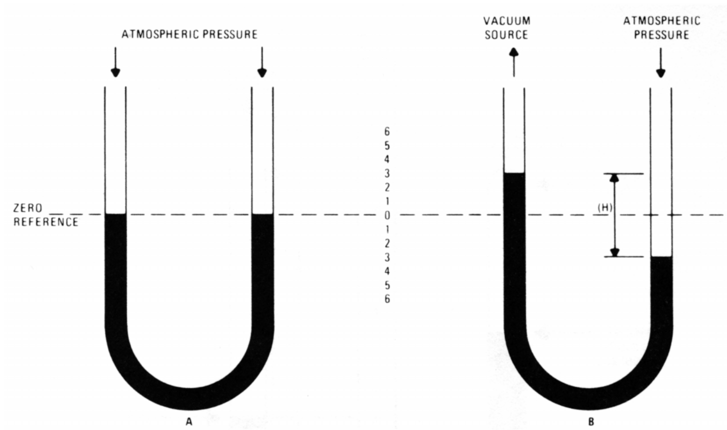
Absolute pressure gauge (for vacuum only) uses a sliding scale to measure the difference in height between mercury in the two legs. Since the mercury completely fills the left-hand leg when the right-hand leg In connected to the atmosphere, the absolute pressure on this leg Is always close to zero. Hence, the difference In height of the two legs measures the absolute pressure (in In. Hg) on the right-hand leg.

With both ends of the tube open to the atmosphere, the liquid is at the same height in each leg. This is a zero value because the atmospheric pressure is equal and has balanced the two columns.

But when a vacuum source is applied to one leg, the mercury rises in that leg and falls in the other leg. The total difference, h (2 + 2 = 4 in this instance), in height between the two new levels is the gauge pressure–in this case, negative.

Had a positive pressure been applied to the left leg, then the mercury would have fallen there and risen in the right leg. Again, the total difference in height between the two new levels would represent the gauge pressure.

Plunger Gauge–A plunger gauge consists of a plunger connected to system pressure, a bias spring, and a calibrated indicator. An auto tire gauge would be an example.



Mercury U-tube manometer. With the right-hand log connected to the atmosphere, the difference in height of the two legs measures the gauge pressure (in In. Hg) on the left-hand leg.

As pressure in the system rises, it moves the plunger against the force exerted by the bias spring. This movement also moves the indicator to show the appropriate pressure on the scale.

With suitable calibration and a spring that can work in extension as well as compression, a plunger gauge can be used for either positive or negative pressure.

Bourdon Gauge–This is the most widely used instrument for measuring both positive pressure and vacuum. Measurement is based on the deformation of an elastic element (a curved tube) by the pressure being measured. The radius of curvature increases with increasing positive pressure and decreases with increasing vacuum. The resulting deflection is indicated by a pointer on a calibrated dial through a ratchet linkage.

Similar gauges may be based on the deformation of diaphragms or other flexible barriers.

McLeod Gauge–For extremely accurate measurements of very low pressures (high vacuums), the McLeod vacuum gauge is the most widely used device. It's also used to calibrate other types of gauges.

This design uses Boyle's Law (see below) to determine pressure in a system. A sample of the gas is isolated in the gauge and reduced in volume by a known amount. This causes a proportional increase in pressure, which, in turn, produces a readable difference in the height of a mercury column.

The McLeod gauge is rarely used for industrial applications. It is better suited to laboratory studies where very high vacuums are involved.

Pressure/Volume/ Temperature Relationships

Pressure, volume, and temperature are basic measurable properties of air. They are not completely independent properties but are interrelated in specific, simple ways. When designing a pneumatic system, it is helpful to understand these relationships.

Boyle's Law–This law describes compression. It states that, at a fixed temperature, the volume of a given quantity of gas varies inversely with the pressure exerted on it. To state this as an equation:

$$P1V1 = P2V2$$

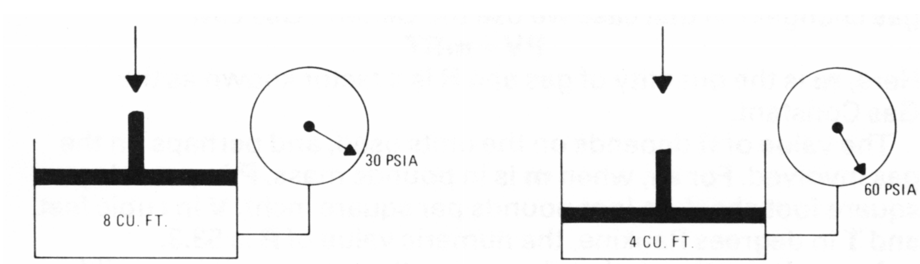
where the subscripts refer to the initial and final states, respectively.

In other words, if the pressure on a gas is doubled, then the volume will be reduced by one-half. The product of the two quantities remains constant. Similarly, if a gas is compressed to half its previous volume, the pressure it exerts will be doubled. In Fig. 9, when 8 cu ft of gas is compressed to 4 cu ft, the 30 psia reading will double to 60 psia.

Designers use Boyle's Law calculations in a variety of situations: when selecting an air compressor, for calculating the consumption of compressed air in reciprocating air cylinders, and for determining the length of time required for storing air. Boyle's Law, however, may not always be practical because of temperature changes. Temperature increases with compression, and Charles' Law then applies.

Charles' Law–This law states that, at constant pressure, the volume of a gas varies directly with its absolute temperature. Absolute temperature is defined on a scale where zero represents the temperature at which all thermal motion

Figure 9



Boyle's Law: Pressure doubles when the volume of a closed container is decreased by one-half.

Figure 10

Δ TEMPERATURE	=	Δ VOLUME (CONSTANT PRESSURE)
Δ TEMPERATURE	=	Δ PRESSURE (CONTANT VOLUME)

Basic relationships of Charles' Law

-460^0F). The two absolute temperature scales in common use are the Kelvin scale, which uses the same degree as the Celsius scale, and the Rankine scale, which uses the Fahrenheit degree.

Expressed as an equation, Charles' Law states:

$$P_1/T_1 = P_2/T_2$$

where, again, the subscripts refer to the initial and final states.

Fig. 10 summarizes the basic concepts of Charles' Law. When the temperature of a gas is increased, the volume changes proportionately (as long as the pressure does not change). The same relationship holds with temperature and pressure, as long as volume does not change.

Combined Gas Law-What if both temperature and pressure are changed at the same time? Another important mathematical description, the Combined Gas Law, can be derived from Boyle's and Charles' Laws. It states that:

$$P_1V_1/T_1 = P_2V_2/T_2$$

This makes it possible to calculate any one of the three quantities -pressure, volume, and temperature-as long as the other two are known.

General Gas Law (Equation of State for an Ideal Gas) -All the above laws compare the state of a given quantity of gas in one condition with that in another. But what if the quantity of gas changes? In that case we use the General Gas Law:

$$PV=mRT$$

Here, **m** is the quantity of gas and **R** is a factor known as the Gas Constant.

The value of **R** depends on the units used, and perhaps on the gas involved. For air, when **m** is in pounds mass, **P** in pounds per square foot absolute (not pounds per square inch), **V** in cubic feet, and **T** in degrees Rankine, the numeric value of **R** is 53.3.

As an alternative, rather than using the mass, **m**, it is possible to use the number of moles, **n**. One mole contains 6.02×10^{23} molecules and weighs the gas's molecular weight in grams. Then, when **P** is in atmospheres, **V** in liters, and **T** in degrees Celsius, R has the value 0.08207. When moles are used, the value of **R** is independent of the gas.

Table 1

Equivalence Table for Pressure/Vacuum Measurements
(pressure reduction of 2 torr corresponds to 0.232% vacuum)

torr mmHg	mbar (10–"MPa)	psi	inches Hg	atm	% vacuum
760	1013	14.696(14.7)	29.92	1.0	0.0
750	1000 (1 bar)	14.5	29.5	0.987	1.3
735.6	981	14.2	28.9	0.968	1.9
700	934	13.5	27.6	0–921	7.9
600	800	11.6	23.6	0.789	21
500	667	9.7	19.7	0 .658	34
400	533	7.7	15.7	0.526	47
380	507	7.3	15.0	0.500	50
300	400	5.8	11.8	0.395	61
200	267	3.9	7.85	0.264	74
100	133.3	1.93	3.94	0.132	87
90	120	1.74	3.54	0.118	88
80	106.8	1.55	3.15	0.105	89.5
70	93.4	1.35	2.76	0.0921	90.8
60	80	1.16	2.36	0.0789	92.1
51.7	68.8	1.00	2.03	0.068	93.0
50	66.7	0.97	1. 97	0. 0658	93.5
40	53.3	0 . 77	1.57	0.0526	94.8
30	40.0	0.58	1.18	0.0395	96.1
25.4	33.8	0.4912	1.00	0.034	96.6
20	26.7	0.39	0.785	0.0264	97.4
10	13.33	0.193	0.394	0.0132	98.7
7.6	10.13	0.147	0. 299	0.01	99.0
1	1.33	0.01934	0.03937	0.00132	99.868
Micron (10^{-3} torr)					
750	1. 00	0.0145	0.0295	0.000987	99.9
100	0.133	0.00193	0.00394	0.000132	99.99
10	0.0133	0.000193	0.000394	0.0000132	99.999
1	0.00133	0.0000193	0.0000394	0.00000132	99.9999
0.1	0.00133	0.00000193	0.00000394	0.000000132	99.99999

AIR COMPRESSORS

To keep you oriented as we discuss pressure, the basic concepts are treated in this sequence: generation, transmission, storage, and utilization of compressed air in a pneumatic system. This section covers air compressors, the devices that generate air pressure.

Pressure Generation: Compression

The pressure exerted by a confined gas results from rapid and repeated bombardment of the container walls by the enormous number of gas molecules present. The pressure can be increased by increasing the number or force of the collisions. Increasing the temperature does this by speeding up the molecules (Charles' Law). Another way is to increase the average number of molecules in a given volume. This is compression. It can be done by either decreasing the volume (Boyle's Law) or increasing the amount of gas.

Liquids and solids can be compressed only with difficulty. But gases are easily compressed because their molecules are relatively far apart and move freely and randomly within a confined space.

Compression decreases the volume available to each molecule. This means that each particle has a shorter distance to travel before colliding with another particle or the wall. Thus, proportionately more collisions occur in a given span of time, resulting in a higher pressure.

Compression Work Requirements

An air compressor does most of its work during the compression stroke. This adds energy to the air by increasing its pressure. Compression also generates heat, however, and the amount of work required to compress a quantity of air to a given pressure depends on how fast this heat is removed. The compression work done will lie between the theoretical work requirements of two processes:

- Adiabatic–a process having no cooling; the heat remains in the air, causing a pressure rise that increases compression work requirements to a maximum value.
- Isothermal–a process that provides perfect cooling; thus, there is no change in air temperature and the work required for compression is held to a minImum.

The difference in the amount of work required to compress air to 100 psi by these two processes is about 36 percent. Most industrial air compressors are near adiabatic, since the process is too fast to allow much heat to escape through the compressor casing.

Air Compressors: Basic Operation

An air compressor operates by converting mechanical energy into pneumatic energy via compression. The input energy could come from a drive motor, gasoline engine, or power takeoff.

The ordinary hand bellows used by early smelters and blacksmiths was a simple type of air compressor. It admitted air through large holes as it expanded. As the bellows were compressed, it expelled air through a small nozzle, thus increasing the pressure inside the bellows and the velocity of the expelled air.

Modern compressors use pistons, vanes, and other pumping mechanisms to draw air from the atmosphere, compress it, and discharge it into a receiver or pressure system. Table 2 summarizes the capabilities of various compressor types.

The most basic types of air compressors are designated as .positive displacement" and "nonpositive displacement" (sometimes called "dynamic"). The characteristic action of a positive displacement compressor is thus a distinct volumetric change-a literal displacement action by which successive volumes of air are confined within a closed chamber of fixed volume and the pressure is gradually increased by reducing the volume of the space.

The forces are static-that is, the pumping rate is essentially constant, given a fixed operating speed. The principle is the same as the action of a piston/cylinder assembly in a simple hand pump.

Positive Displacement Compressors

Positive displacement compressors generally provide the most economic solution for systems requiring relatively high pressures. Their chief disadvantage is that the displacing mechanism provides lower mass flow rates than nonpositive displacement compressors (see pages 29-33).

Pressure Characteristics - A compressor with a positive displacement pumping mechanism has these important pressure characteristics:

- The pressure against which the compressor works rises to higher and higher values as pumping continues. It must be limited by some external pressure control device.
- The rate of free air delivery is highest at 0 psig and very gradually drops to lower values as pressure increases.
- The amount of heat generated progressively rises as pressure increases, causing substantial increases in temperature of both the air handled and the compressor structure.

Types of Positive Displacement Compressors

Positive displacement compressors are divided into those which compress air with a reciprocating motion and those which compress air with a rotary motion. The principal types of positive displacement compressors are the piston, diaphragm, rocking piston, rotary vane, lobed rotor, and rotary screw.

Reciprocating Piston -This design (Fig. 11) is widely used in commercial air compressors because of its high pressure capabilities, flexibility, and ability to rapidly dissipate heat of compression. And it is oil-less.

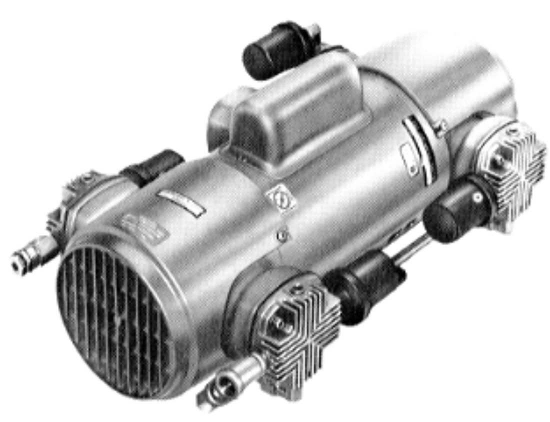
Compression is accomplished by the reciprocating movement of a piston within a cylinder (Fig. 12). This motion alternately fills the cylinder and then compresses the air. A connecting rod transforms the rotary motion of the crankshaft into reciprocating piston motion in the cylinder. Depending on the application, the rotating crank (or eccentric) is driven at constant speed by a suitable prime mover. Separate inlet and discharge valves react to variations in pressure produced by the piston movement.

As Fig. 12 shows, the suction stroke begins with the piston at the valve side of the cylinder, in a position providing minimum (or clearance) volume. As the piston moves to a maximum volume position, outside air flows into the cylinder through the inlet valve. The discharge valve remains closed during this stroke.

During the compression stroke, the piston moves in the opposite direction, decreasing the volume of air as the piston returns to the minimum position.

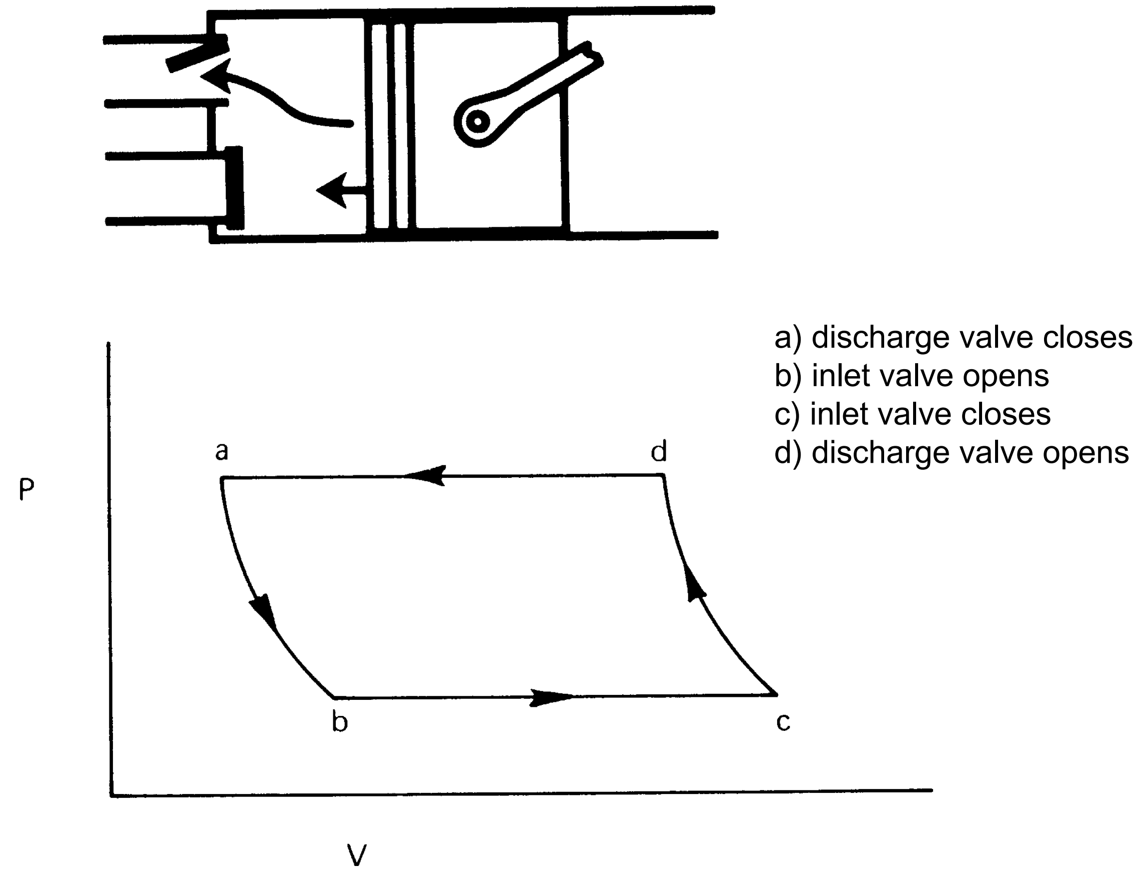
During this action, the spring-loaded inlet and discharge valves are automatically activated by pressure differentials. That is, during the suction stroke, the piston motion reduces the pressure in the cylinder below atmospheric pressure. The inlet valve then opens against the pressures of its spring and allows air to flow into the cylinder.

Figure 11



Typical reciprocating piston air compressor

Figure 12



Reciprocating motion of the piston compresses air with each revolution of the crankshaft.

When the piston begins its return (compression) stroke, the inlet valve spring closes the inlet valve because there is no pressure differential to hold the valve open. As pressure increases in the cylinder, the valve is held firmly in its seat.

The discharge valve functions similarly. When pressure in the cylinder becomes greater than the combined pressures of the valve spring and the delivery pipe, the valve opens and the compressed air flows into the system.

In short, the inlet valve is opened by reduced pressure, and the discharge valve is opened by increased pressure.

Some piston compressors are double-acting. As the piston travels in a given direction, air is compressed on one side while suction is produced on the other side. On the return stroke the same thing happens with the sides reversed. In a single-acting compressor, by contrast, only one side of the piston is active.

Single-acting compressors are generally considered light-duty machines, regardless of whether they operate continuously or intermittently. Larger double-acting compressors (usually watercooled) are considered heavy-duty machines capable of continuous operation.

Sizes of reciprocating piston compressors range from less than 1 hp to 6000 hp. Good part-load efficiency makes them very useful where wide variations in capacity are needed.

Their disadvantages? Reciprocating piston compressors inherently generate inertial forces that shake the machine. Thus, a rigid frame, fixed to a solid foundation, is often required. Also, these machines deliver a pulsating flow of air that may be objectionable under some conditions. Properly sized pulsation damping chambers or receiver tanks, however, will eliminate such problems.

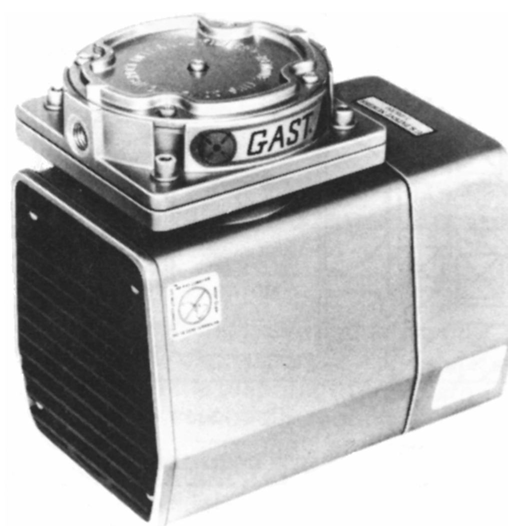
In general, the reciprocating piston compressor is best suited to compression of relatively small volumes of air to high pressures.

Diaphragm - The diaphragm design (Fig. 13) is a modification of the reciprocating piston principle. An outstanding characteristic of the diaphragm design is that the basic compressing mechanism does not require a sliding seal between moving parts. A diaphragm compressor is also oil-less and it is therefore often selected when no oil contamination of the line or atmosphere can be tolerated.

Compression is performed by the flexing of a diaphragm back and forth in a closed chamber. Fig. 14 indicates how this flexing action is generated by the motion of a connecting rod under the diaphragm. Only a short stroke is required to produce pressure effects similar to those produced by a reciprocating piston in a cylinder.

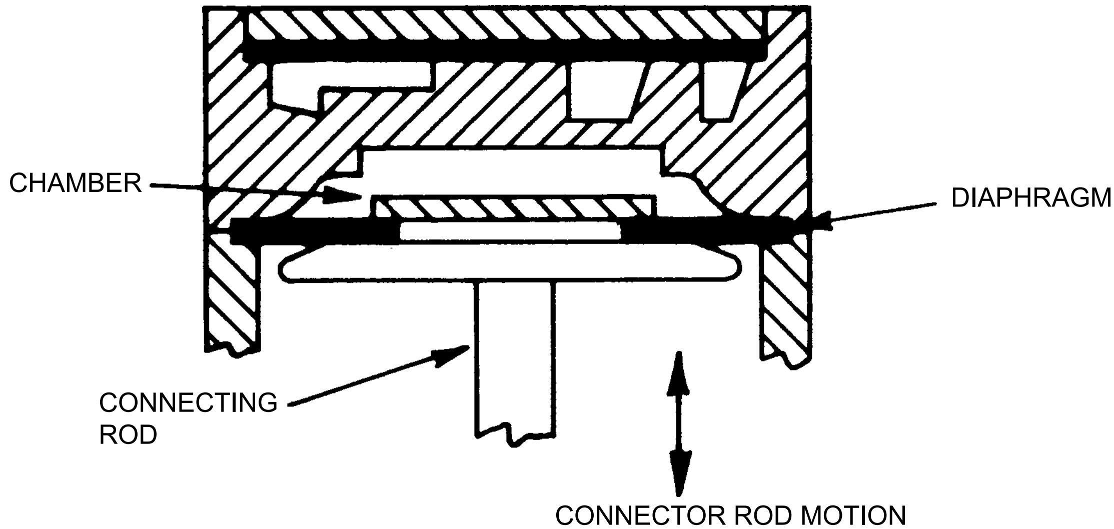
Intake and discharge valves convert the volume changes produced by the reciprocating movement into pumping action. The reed-type valves work like those in the piston design.

Figure 13



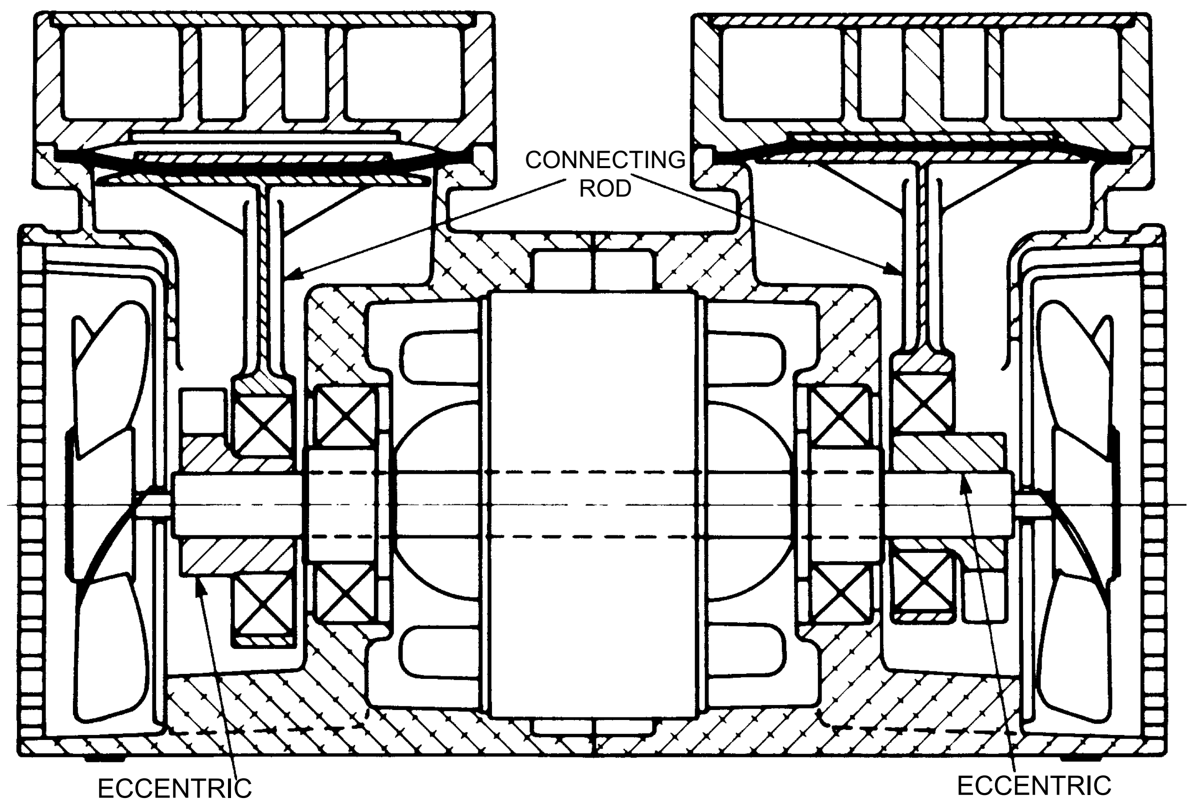
Typical diaphragm compressor. The heavy-duty diaphragm is made of heat-resistant elastomer with fabric reinforcement.

Figure 14



Cross-section shows diaphragm flexing in response to up/down motion of connecting rod.

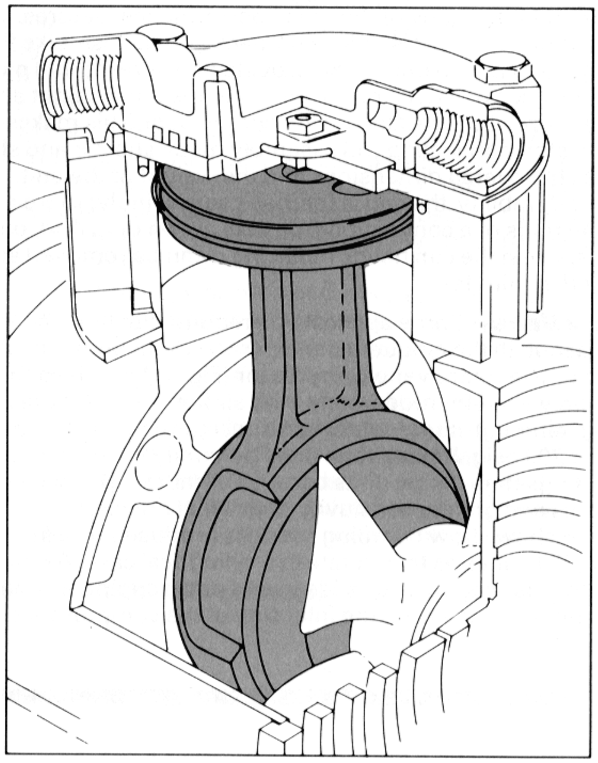
Figure 15



Dual-chamber diaphragm compressor.

Fig. 15 shows a dual-chamber machine. The contour of the diaphragm in the separate chambers indicates different stroke positions at the same instant.

The pressure capabilities of the diaphragm compressor are less than those of the piston type, but usually exceed those of the rotary vane type.



The rocking piston principle can be viewed as a combination of the reciprocating piston and diaphragm Ideas.

Rocking Piston - The rocking piston principle (Fig. 16) is another variation of reciprocal compression. In fact, it can be viewed as a combination of the diaphragm and piston principles.

The rocking piston pump essentially mounts a piston rigidly (no wrist pin) on top of the diaphragm unit's eccentric connecting rod. This piston is surmounted by a cup made of Teflon , for instance. The cup functions both as a seal-equivalent to the rings of a piston compressor-and as a guide member for the rod. It expands as the piston travels upward, thus maintaining contact with the cylinder walls and compensating for the rocking motion.

® Teflon is a registered trademark of DuPont.

The rocking piston compressor not only combines the mechanical features of the reciprocating piston and diaphragm types, but it also combines many of their best performance features. Like the diaphragm type, it is quiet, compact, and oil-less. Like the reciprocating piston unit, it can provide pressures to 100 psi.

The absence of a wrist pin is the key to the light weight and compact size of the rocking piston compressor. This makes the entire piston-connecting rod assembly much shorter and sharply reduces the overall dimensions and the weight of the unit.

As for durability, the cup is (perhaps surprisingly) more durable than the rings of a conventional oil-less piston unit. And, on Gast models, when the cup needs replacing it can be removed and replaced in minutes.

Rotary Vane - Some applications require that there be little or no pulsation in the air output, and perhaps a minimum of vibration also. The rotary vane compressor (Fig. 17) provides this. It is commonly used for moderately high air flows at pressures under 30 psig, although some rotary vane designs can provide pressures of 200 psig. Rotary vane units generally have lower pressure ratings than piston units because of more difficult sealing problems and greater sensitivity to thermal effects.

Fig. 18 shows how pumping action is produced by a series of sliding, flat vanes as they rotate in a cylindrical case. As the rotor turns, the individual vanes slide in and out, trapping a quantity of air and moving it from the inlet side of the compressor to the outlet side.

Figure 17

Typical rotary vane air compressor.

There are no valves in the rotary vane design. The entire flow of air into and out of the individual compartments is controlled by the movement of the vanes across separate inlet and discharge ports.

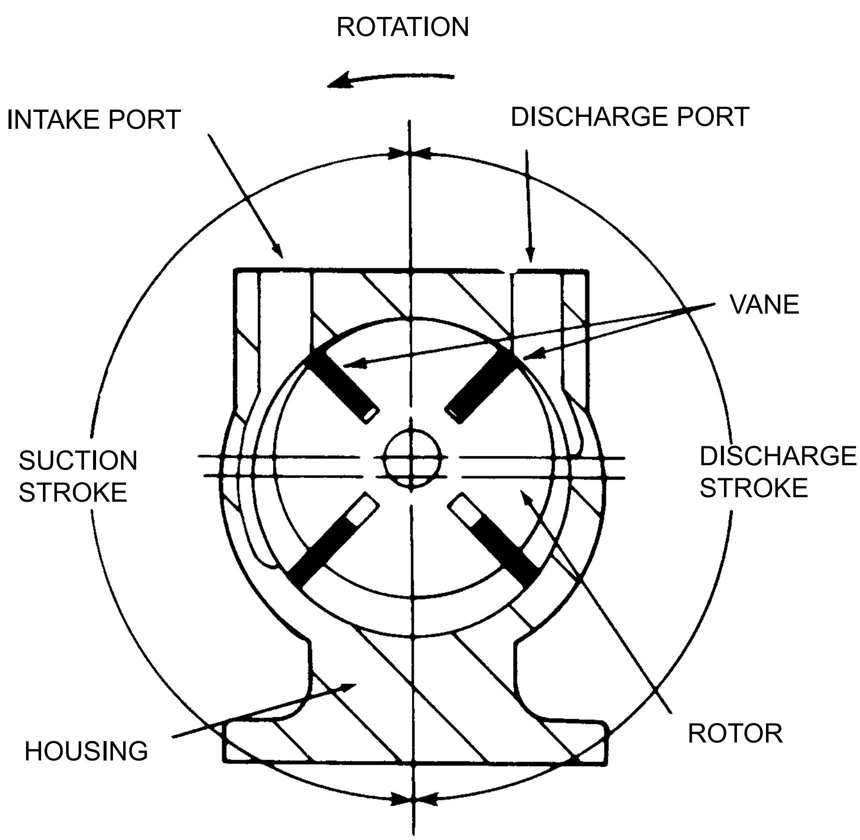
The rotor is mounted eccentrically -that is, not in the center of the casing. As the rotor rotates, the vanes are flung outwards and held against the body bore by centrifugal and pressure-loading forces. This creates a series of air compartments of unequal volume (because of the rotor's eccentricity). The compartments formed between adjacent vanes gradually become larger during the suction part of the cycle, and air is drawn into the compartment from the inlet port.

During the discharge portion of the cycle, the compartment volumes gradually become smaller, compressing the air. When a rotating compartment reaches the discharge port, the compressed air escapes to the delivery system.

The suction and exhaust flows are relatively free of pulsation because the inlet and discharge ports do not have valves, and the air is moved continuously rather than intermittently.

Rotary vane compressors have certain significant advantages. In addition to providing smooth, pulse-free air flow without receiver tanks, they are compact (or, equivalently, offer high flow

Figure 18



In a rotary-vane compressor, the eccentrically mounted rotor creates smaller compression compartments as the vanes are pushed in by chamber walls.

capacities for a given size), are simple and economical to install and operate, have low starting and running torque requirements, and produce little noise or vibration.

Rotary Screw and Lobed Rotor - Two other types of positive displacement compressors are the rotary screw and lobed rotor. Neither is as widely used, especially in smaller sizes, as are rotary vane and piston compressors.

Rotary screw compressors are used when nearly pulseless high-volume air is required. The compression mechanism is composed of two meshing rotors that have helical contours. When the rotors are driven at the same speed, air is trapped between the lobes as the screws turn. The volume between the advancing rotor helix and the endplate diminishes, forming continuous cavities until the end of the helix passes over the discharge port.

In a lobed rotor compressor, a pair of mating lobes on separate shafts rotate in opposite directions to trap incoming air and compress it against the casing. Lobed rotor units provide very high air flows at pressures between those of nonpositive displacement compressors and other types of positive displacement units.

Multistage Compression

Compression may be accomplished in one or more stages. That is, air can be compressed once or several times before it reaches the compressor outlet and is delivered to the system devices. Each stage provides a proportional increase in the output pressure.

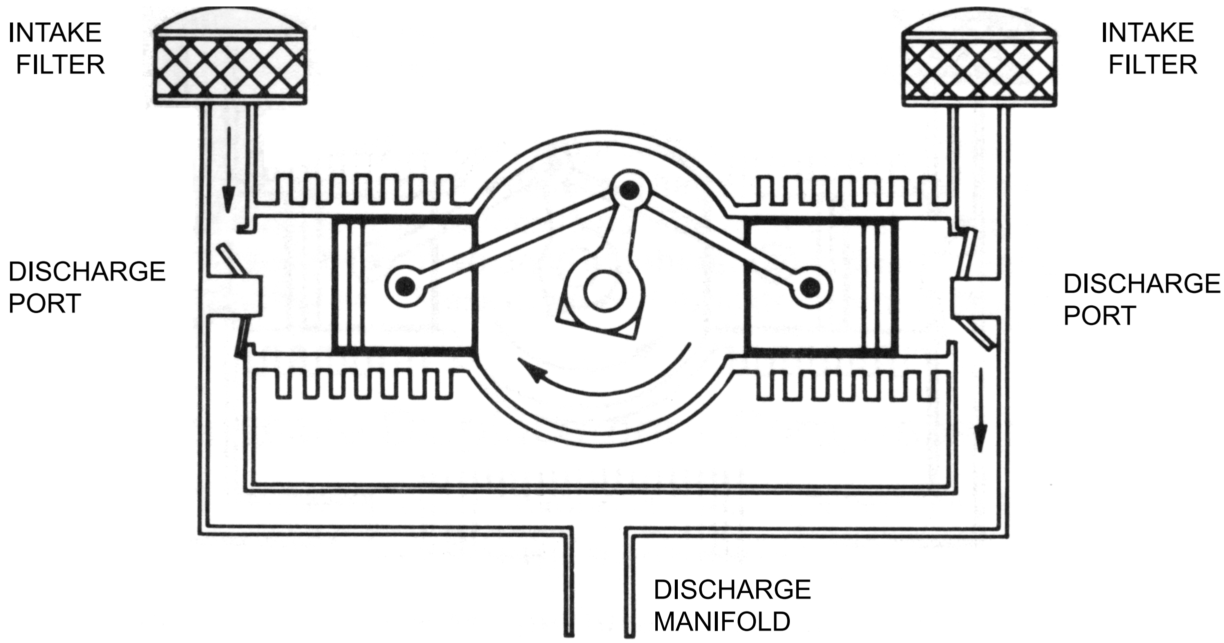
Positive displacement compressors have the advantage of providing relatively large pressure changes in a single stage, and very large pressure changes in a few stages. However, the pressure output of nonpositive displacement compressors can also be raised by staging.

Single Stage - Fig. 19 is another way of illustrating how the compression process is carried out in a single pass through a pumping chamber. This piston-type compressor has two cylinders, but the compression action occurs in a single stage. The cylinders are connected in parallel between the atmosphere and the discharge manifold.

The normal maximum pressure rating for single-stage compressors is about 100 psig. Operation above this level increases the heat of compression (caused by leakage and recompression) to levels that could harm the compressor and the overall system.

Multiple Stage-In multiple-stage compression, the gas moves from one chamber to another. This sequential action provides the final pressure.

For general utility and process purposes, two-stage compression is usually justified when the compression ratio (R,) exceeds six. When Rc exceeds 20, compression is



SINGLE STAGE TWO CYLINDER COMPRESSOR

Basic operation of a single stage/two cylinder air compressor.

usually accomplished in three stages. To put this in pressure units, the upper limit for utility two-stage compressors is between 280 and 300 psig. A gauge pressure of 500 psi has an R_C value of 35.

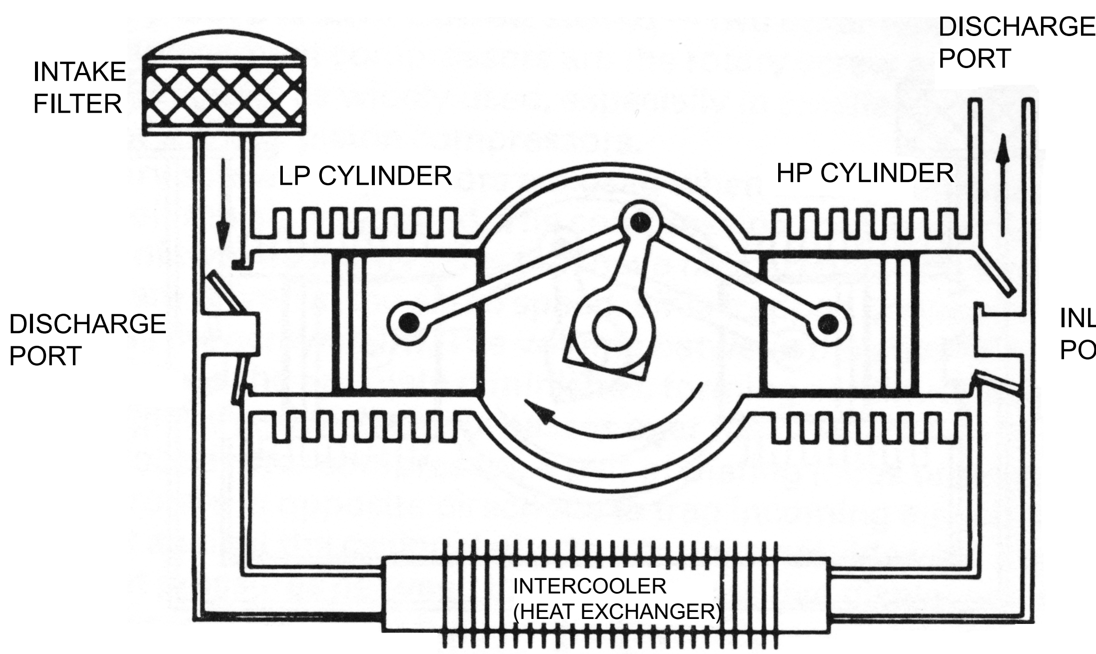
Some multistage compressors eliminate the problem of increased heat of compression above 100 psig. This is done by:

- Compressing the air to an intermediate pressure in the large-diameter low-pressure cylinder.
- Removing a portion of the heat of compression before the air is fed to the next stage (this is known as "intercooling" and is normally done by an air-cooled or water-cooled heat exchanger).
- Further compressing the air to final pressure in a smaller high-pressure cylinder.

As Fig. 20 shows, these two cylinders are connected in series through the intercooler (compare with Fig. 19). Intercooling greatly decreases both the total temperature rise of the compressed air and the amount of work required for its compression. But the added cost of an intercooler cannot always be justified on a small compressor.

Some two-stage compressors have three cylinders: two low-pressure cylinders connected to one high-pressure cylinder through an intercooler.

Figure 20



TWO STAGE TWO CYLINDER COMPRESSOR

Basic operation of two stage/two cylinder air compressor

Lubrication and Exhaust Air Quality

Contamination in the air can affect many applications. A laboratory process, for example, powered by compressed air may be extremely sensitive to moisture, oil, or dust particles. Or in such places as food processing plants, even the air exhausted from the pneumatic system may have to be entirely free of oil vapor and contaminants.

A variety of filters, generally expensive, have been developed to solve such problems. An alternative is to use an oil-less air compressor.

Oil-Less Compressors - Compressors designed with "dry" self-lubricating materials, such as graphite or Teflon, produce oil-free air both in the line and at the exhaust. They effectively eliminate the presence of air/oil vapors in applications where even a very fine oil mist can cause contamination, stains, deterioration, or a safety hazard.

Oil-less pneumatic systems are particularly useful in the food, textile, paper, pharmaceutical, and chemical industries. And since no maintenance lubrication is required, these units can be mounted in the best, rather than the easiest to reach, location.

Oil-Lubricated Compressors - if, for some reason, an oilless air compressor is not practical in an application where contarnination is prohibited, then an oil-lubricated unit must be used and equipped with appropriate filters to remove the oil after the air is compressed.

In an oil-lubricated compressor, a thin film of oil is maintained between the walls of the pumping chamber and the pistons, vanes or other moving parts. Siphon or wick-type lubricators are used in light-duty operations. Pressure type lubricators are used in heavy-duty or continuous-duty applications.

In general, oil-lubricated compressors have higher pressure ratings than oil-free compressors. They also run cooler and may therefore have longer service lives. The relative value of these factors versus the convenience of inherently oil-free operation dictate whether an oil-less or an oil-lubricated compressor should be used.

Nonpositive Displacement Compressors

Also called "dynamic," "continuous-flow," and "velocity-type" compressors, this category comprises machines that use changes in kinetic energy to create pressure gradients.

Kinetic energy is the energy that a body possesses by virtue of its motion. A fluid's kinetic energy can be increased either by rotating it at high speed or by providing an impulse in the direction of flow.

Unlike the positive displacement compressor, in which distinct volumes of air are isolated and compressed, a nonpositive displacement compressor does not provide a constant-volume flow rate over a range of discharge pressures. This is because the compartments are not isolated from each other and leakage between them increases as pressure rises.

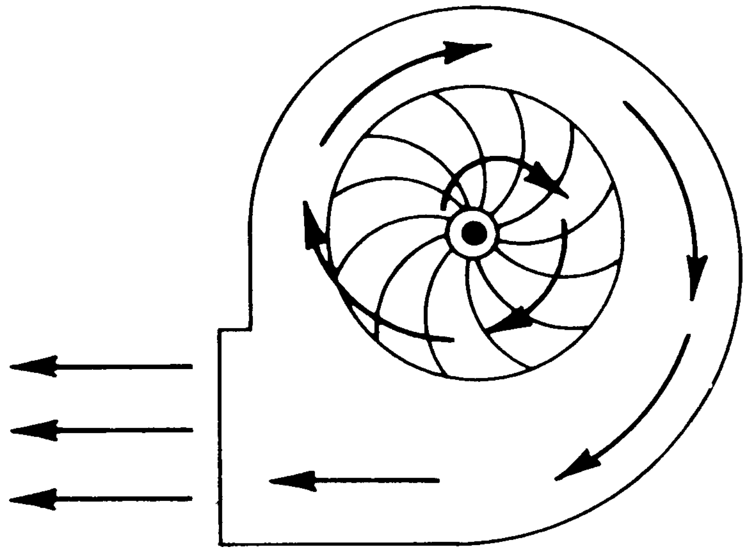
Initial acceleration of the air produces a negative (suction) pressure at the inlet port, drawing air in. Partial deceleration of air at the discharge port converts some of the kinetic energy to pressure. Speed of the rotating impeller determines the pressure change. Higher pressure differences require either faster impeller speeds or additional stages.

The most important advantage of nonpositive displacement machines is their ability to provide very high mass flow rates. On the other hand, multiple stages are required to provide pressures above 4 or 5 psi and such machines are cost effective only for flow rates above 80-100 cfm.

Nonpositive displacement devices are sometimes called fans or blowers rather than compressors. By some definitions, a fan provides less than 0.5 psi pressure and a blower between 0.5 and 10 psi. The distinction is frequently blurred in common use, however.

The three common types of nonpositive displacement compressors are centrifugal, axial, and peripheral (or regenerative). These names derive from the direction of air flow through their compression chambers.

Figure 21



In a centrifugal blower, a rotating impeller sweeps air radially along the casing to the outlet.

Centrifugal- Centrifugal compressors a re best suited to the continuous movement of large air volumes through small pressure ranges. Fig. 21 shows the basic operation. Air leaving a rotating impeller passes radially outward to the casing. Centrifugal action builds up velocity and pressure levels.

In its simplest form, a centrifugal compressor consists of a high-speed rotating impeller that receives air through an inlet nozzle at the center. The impeller vanes are fixed (unlike those in the rotary vane design). They throw the air centrifugally outward toward the casing, increasing its velocity and energy. Here, an outlet discharges the air into a stationary passageway known as a "diffuser." The diffuser reduces the air velocity, thus raising the pressure. Beyond the diffuser, the velocity may be further reduced and pressure increased by a "collector."

Staging can yield higher pressures. Staging is accomplished by directing the output from the diffuser of one stage into the nozzle of the next.

Because the flow from the impeller is continuous, a smooth, surge-free output is obtained. Furthermore, discharge pressure depends only on impeller speed. It is nearly constant, despite variations in flow, over the stable operating range.

But this can be a drawback if the demand falls far enough below the rated flow, allowing system pressure to build up. The compressor continues to deliver air at about the same pressure until the back-pressure exceeds that developed by the compressor. The result is "surge"-a reversal of flow. This reversal immediately allows the back-pressure to go down, and regular compression is resumed.

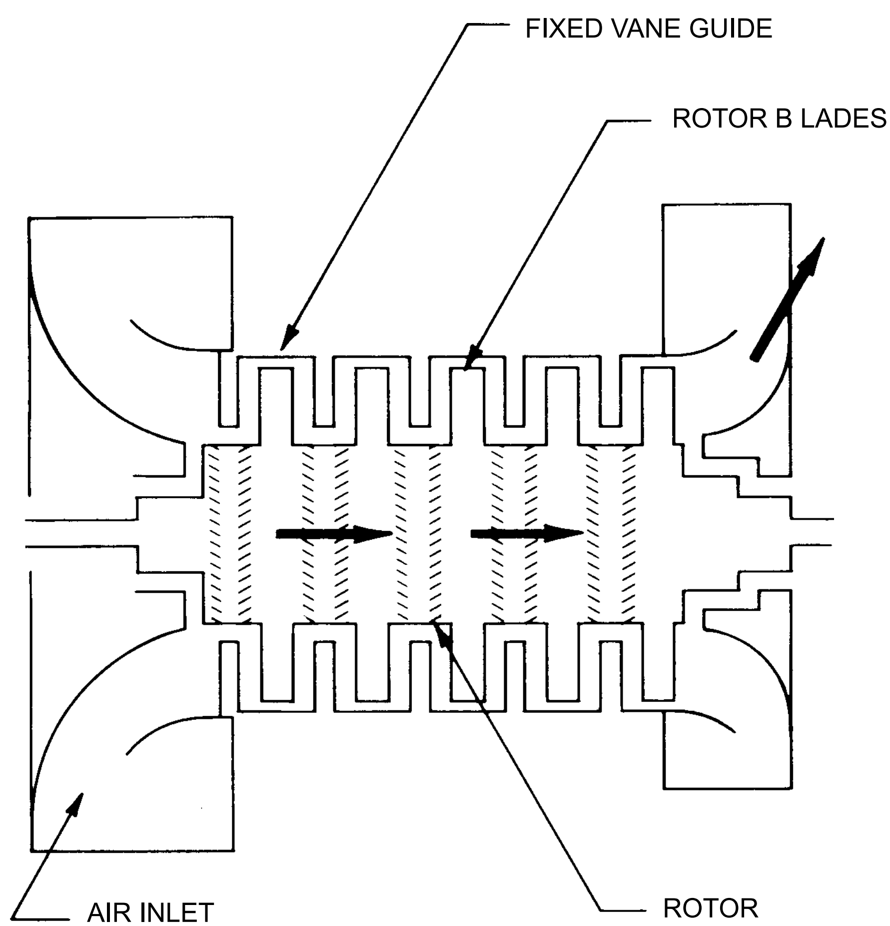
Surge can be prevented if flow remains above a limit established for each design. Various models have minimum operating flows between 45 and 90 percent of rated capacity.

Centrifugal compressors are available in both small and very large sizes. Units with up to six stages and supplying 30,000 cfm of air are commercially available. Operating speeds are very high compared with other types-up to 20,000 rpm in standard applications.

Axial Flow -This category is generally used for ultrahigh flow applications (30,000 to 1,000,000 cfm). Air flow is through a duct, primarily in a direction parallel to the axis of rotation. In multistage versions, this flow channeling is provided by the fixed guide vanes or stator blades positioned between each stage (Fig. 22).

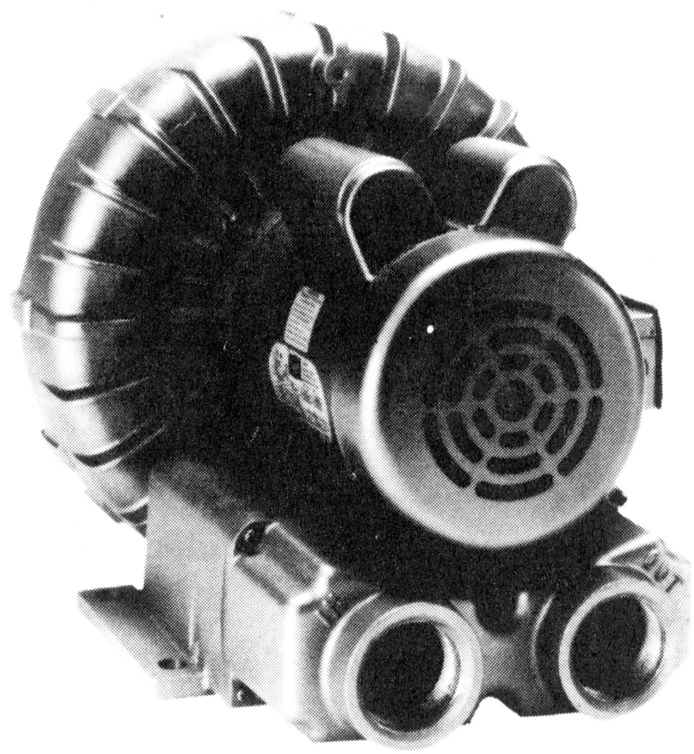
An axial flow compressor requires about a third the floor space of a centrifugal design, and it weighs about a third as much. Below capacities of 100,000 cfm, though, the axial design is seldom competitive in price.

Figure 22



Air flows (arrows) through multistage axial flow blower. The fixed guide vanes between each stage keep air flow parallel to the axis of rotation.

Figure 23



Typical peripheral (regenerative) blower provides equivalent of multistage compression in a single revolution of the impeller.

Peripheral (Regenerative) -These units (Fig. 23) provide somewhat higher pressures than do the other dynamic designs. With some units, a single-stage regenerative action provides pressures similar to those obtained with multistage centrifugal compressors.

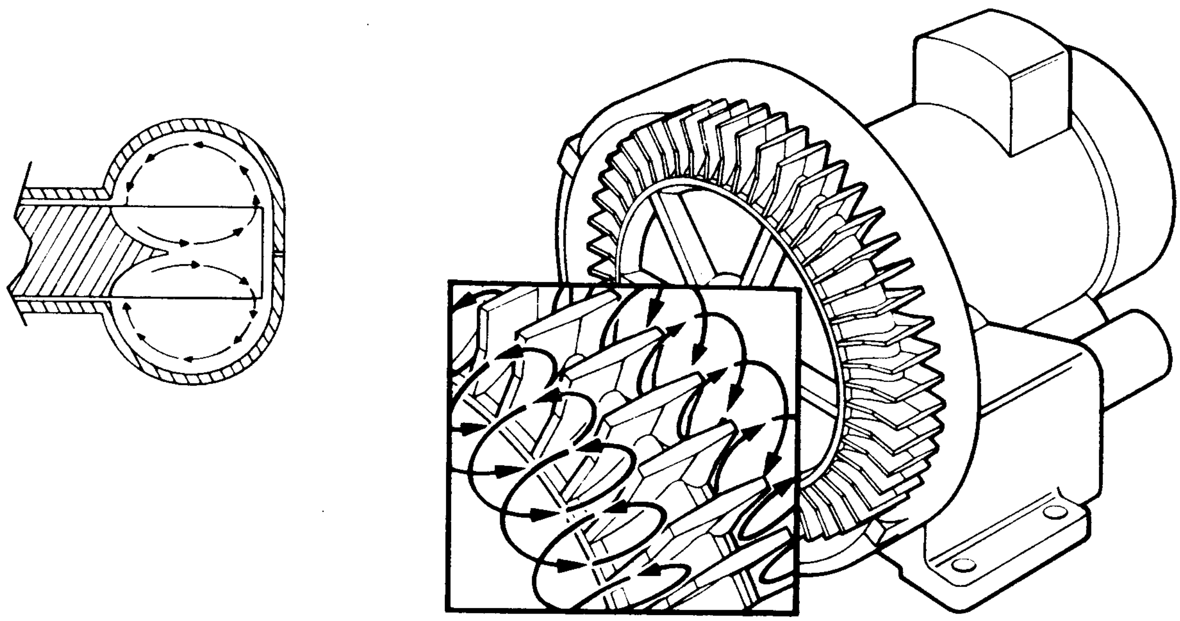
The compression space consists of a hollow, circular ring between the tips of the impeller blades and the walls of the peripheral passage. See Fig. 24. In operation, the rotating impeller draws air from the inlet port into the compression space. The air moves radially outward to the curved housing by centrifugal force.

The action is called "regenerative" because a certain amount of air slips past each impeller blade during rotation and returns to the base of a succeeding blade for reacceleration. The effect is like the pressure buildup in a multistage blower, and higher pressures can be generated.

Single-stage peripheral blowers are available in capacities up to several hundred cfm and can generate pressures close to 5 psig. Multistage versions are also available.

A significant advantage for peripheral blowers is that they are highly immune to operating conditions that might otherwise cause blockage of inlet and discharge flows. They also provide oil-less operation and continuous pulse-free flow.

Figure 24



The arrows show the route air takes through a regenerative blower.

Compressor Controls and Cycling

Before we discuss compressor selection, it is necessary to look at how pressure is regulated in a pneumatic system.

One way is to use a pressure relief valve between the compressor and the receiving tank. In this case, the compressor runs continuously against the system pressure.

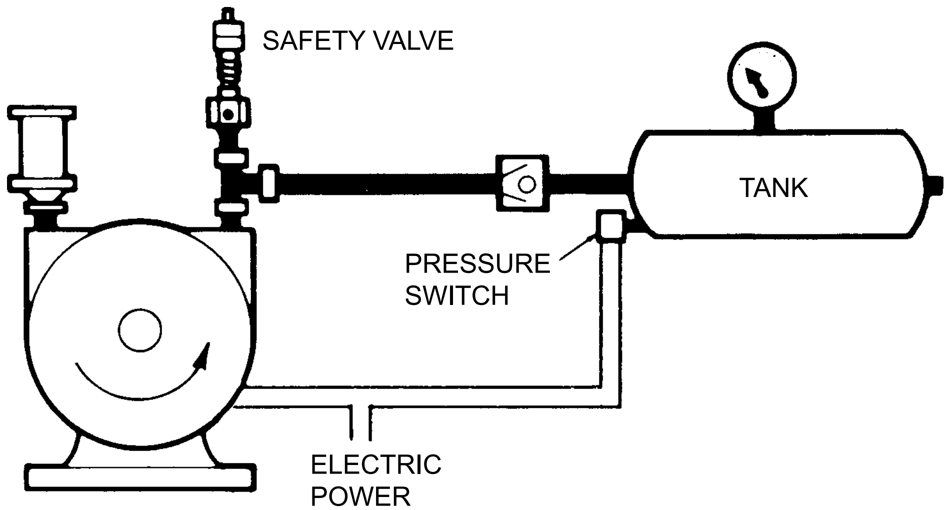
Alternatively, the compressor can be set to automatically turn off when the pressure reaches a preset maximum and on again when it reaches a minimum. This on/off cycling gives the compressor a chance to cool down and allows some models to be used at pressures higher than their continuous duty rating.

Yet another alternative, known as load/unload cycling, diverts the compressor output to the atmosphere when the set pressure is reached. This limits the work the compressor must do and the resulting heat buildup without imposing high starting torques on the motor.

On/Off Cycling - Fig. 25 shows how an electrical pressure switch, installed on the receiver tank, provides on/off cycling of the compressor. The switch starts and stops the drive motor as the pressure reaches preset levels. The normal range between .cut-out" and "cut-in" levels is 15 to 20 psi. The compressor in this type of system is said to have an intermittent duty cycle. As mentioned elsewhere, pressure ratings of many compressors are higher for intermittent than for continuous duty.

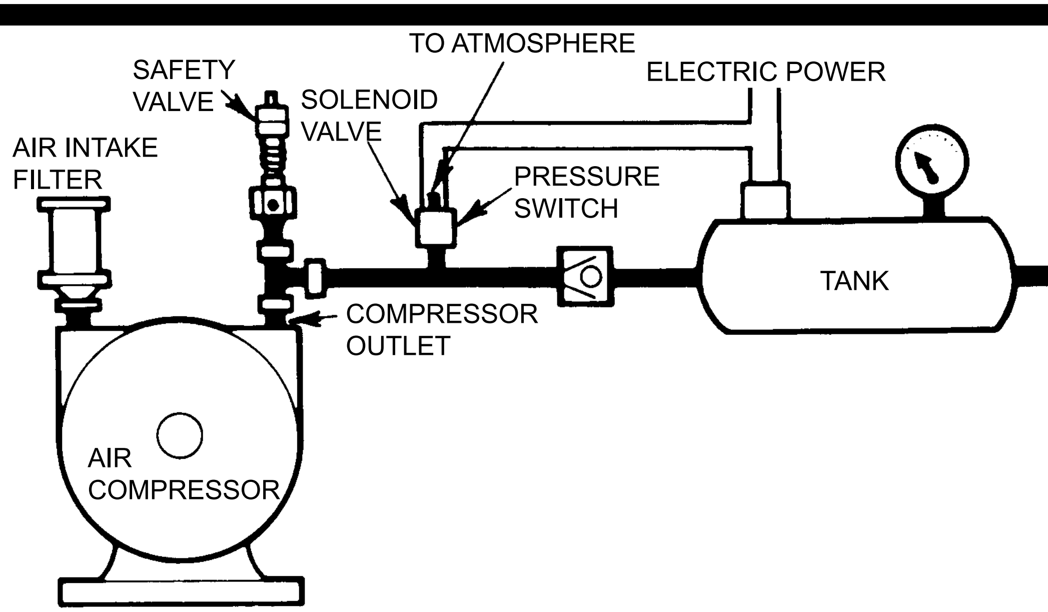
A safety valve is always used with this type of control to provide independent protection against overpressure.

Figure 25



A pressure switch can control receiver tank pressure by on/off cycling of compressor drive motor. This is completely independent of downstream pressure.

Figure 26



A pressure switch at the receiver tank can actuate a solenoid valve, providing automatic venting at a predetermined pressure limit.

Load/Unload Cycling - An electrical pressure switch can also be used to provide overpressure protection when it is not possible or advisable to start and stop the drive unit (Fig. 26). Examples are compressors driven by gasoline engines or power takeoffs.

As in the preceding system, the pressure switch is mounted at the receiver tank. But instead of being connected to a drive device, the switch is connected to a solenoid valve mounted in the delivery line,

At a predetermined upper ("unload") pressure limit, the pressure switch de-energizes the solenoid valve. The valve opens and vents output air to the atmosphere. The valve remains de-energized until the pressure switch senses that the pressure has dropped to a lower ("load") limit, when the valve closes.

Overpressure protection is similar to that provided by a pressure relief valve, except that the pressure control is at the receiver tank rather than at the compressor. The venting action reduces compressor work and heat to a minimal level. It also provides an internal flow of cool air through the compressor, supplementing the external forced draft provided by the fan.

A load/unload system requires the use of a safety valve to provide independent overpressure protection for the power supply system.

Selection of Compressor Type

In selecting an air compressor, the designer must determine how much pressure and air flow is required to meet specific application needs. He must also determine the drive power requirements for the compressor and how they will be met. Other considerations include cost, space, weight limitations, and possible needs for oil- free or pulseless air. Only when all these have been reviewed does the designer have enough information to select the type and size of compressor, plus the other components needed to complete the system.

Maximum Pressure Rating

The primary criterion for evaluating performance of a compressor is its maximum pressure rating. This is defined as the maximum pressure at which the compressor can deliver air to the system in commercial operation.

For any compressor, the physical design sets limits (because of such factors as air leakage and drive power limitations) on the pressure that can be generated. But in many cases, it is heat build-up that determines the actual pressure rating. The easier the compressor is to cool, the higher the pressure rating. This is also why many compressors have continuous-duty ratings that are considerably lower than their intermittent-duty ratings.

Basically, the selection of the appropriate compressor type is determined by comparing the maximum pressure requirements of the application with the maximum pressure rating of available compressor types.

As Table 3 shows, an application where the system pressure requirement is relatively low will give the designer a greater variety of compressor types from which to choose. As the system pressure requirement increases, the number of available alternatives diminishes-sometimes to a point where only a single compressor is applicable. Sometimes it may be a rather costly unit and have relatively high power requirements per cubic foot of air delivered.

The maximum system pressure required for a given application depends on the operating conditions, a requirement that isn't as simple as it sounds because several factors may be involved.

For example, if the system pressure is controlled by pressure venting through a relief valve, the valve will normally be set so that the maximum system pressure equals the highest pressure required by any operating device. Using this type of control, however, requires the compressor to work harder and longer at maximum pressure than with other control techniques.

But if system pressure is controlled by automatic on/off or load/ unload cycling, then the maximum system pressure will be the cutoff pressure of the cycling switch. This may be 15 to 20 psig higher than the highest working pressure required by any single operating device.

The working pressure requirements of individual devices can be determined from handbook formulas, performance curves, catalog data, or actual tests made with prototype systems. Normally, the working pressure needed will depend on the size of the actuator (work device) used. Sometimes system pressure requirements can be reduced when it is practical to use a larger actuator. That is, the required force can be generated by a lower pressure if it acts over a larger area. Overall system costs usually determine such decisions.

In some cases, the required working pressure may depend on the "pressure heads" needed for specific air flow velocities or rates.

Air Flow

While flow capacity is the primary factor in selecting a given size compressor, it also enters when selecting a compressor type. This is because different types of compressors tend to offer trade-offs between air flow and pressure capability. Not only do dynamic compressors provide higher flows and lower pressures than do positive displacement units, but the same is often true within each group. Rotary vane compressors, for example, tend to provide more air but less pressure than do comparably sized piston compressors.

Thus, after determining the required air pressure, the usual procedure is to select the type of compressor that will provide maximum air flow at that pressure, subject to other constraints. At a minimum, it is necessary to be sure that units offering the required flow are available in the type of compressor selected. Table 4 lists typical capacities for various types of compressors.

Compressors are available either with an integrally mounted motor or configured for a separate drive.

If capacity is a problem, it sometimes can be solved by changing the pressure vs. flow requirement to allow use of a different and less expensive type of compressor. In any case, this potential solution should be considered before final selection is made.

Type of Drive

Air compressors are furnished with or without an integrally mounted motor (Fig. 27). This allows the compressor to utilize whatever power may be available. For example, a separate gasoline-engine drive may be required in a remote outdoor location where there is no electric power.

Separate-Drive Compressors -These units are used in applications requiring drive by variable step-down belt drive systems, or by power secured from power takeoffs, gasoline engines, or special electric motors. The drive unit and compressor are generally connected either by a flexible coupling or by a drive pulley containing a built-in cooling fan.

The most significant advantage of a separate drive model is the cost saving realized when an existing power source is available. In addition, a belt drive sometimes makes possible higher operational speeds.

If a separate drive system is chosen, the designer must make sure it can provide both the rated speed and horsepower required by the type and size of compressor selected at its maximum operating pressure. Table 5 lists the horsepower/speed ranges for Gast separate-drive piston and rotary vane compressors. The speed of the piston units, for example, can be varied from 1000 to 2000 rpm by changing the size of the drive pulley.

Power requirements listed in the compressor manufacturer's catalog should assure adequate power for rated speed operation at any pressure up to and including the compressor's maximum pressure rating.

The drive speed determines the amount of air delivered. If the compressor is operated at less than the rated speed given in the manufacturer's catalog, both air flow and the horsepower required will be reduced in the same proportion. The torque requirement will remain unchanged, but only with positive displacement compressors.

In general, operation of separate-drive compressors at abnormally low speeds or at higher than rated speeds can cause problems. Always consult the manufacturer if it will be necessary for the system to deviate considerably from the rated speed.

Motor-Mounted Compressors - Motor-mounted compressors are literally built up around the drive motor shaft. The stationary elements of the compressor are securely anchored to the motor frame. The rotor (or eccentric) that generates the pumping action is installed on the motor shaft. Therefore there is no need for base-plate mounting and power transmission components. This approach provides an extremely compact, lowcost motor/compressor package to accommodate a range of capacities.

If electric power is available, a motor-mounted compressor will quickly solve the entire drive problem at reasonable cost. The only drive system problems are those associated with supplying and controlling the electricity. In general, motor-mounted compressors are more compact and cost less than the combined cost of a separate drive, base and motor.

Table 6 summarizes some available motor-mounted compressors (Gast units are taken as typical examples). Horsepower values are included to indicate the range of power requirements.

Selecting Electric Motor Drives

Once a motor-mounted compressor has been decided on, it is necessary to select the precise motor to be used. A variety of types are available, either as standard units or on custom order.

Motor Types -Several different types of electric motors are available, each with its own advantages and drawbacks. These include the following.

Shaded Pole - offers low starting torque at low cost. Usually used in direct drive applications for small units.

Permanent Split Capacitor (PSC) -Performance and applications are similar to shaded pole motors but the PSC is more efficient, with lower line current and higher horsepower capabilities.

Split-Phase - Offers moderate starting torque but high breakdown torque. Used on easy-starting applications. Small motors may have an external relay in place of the usual centrifugal starting switch.

Capacitor Start -In many respects, the capacitor start motor is similar to the split-phase motor. The main difference is the use of a capacitor in series with the start winding, providing higher starting torque.

Three-Phase -These motors operate on three-phase power only. They offer high starting and breakdown torque, high efficiency, medium starting current, simple, rugged design, and long life.

Direct Current (DC) -These are generally used only if DC is available. They are usually used with batteries.

Frequency -most motors available in this country are designed to operate at 60 Hz. At this frequency they will run at either 1725 or 3450 rpm, depending on the number of poles wound into the motor at manufacture. Except for specialized (and relatively expensive) adjustable-speed motors, a motor's speed is fixed by its design and cannot be changed.

Some motors are dual-frequency, designed to operate either at 60 Hz or at the European standard of 50 Hz. It is important to remember that when one of these motors is operated at 50 Hz, both its speed and the pneumatic output of the unit it is driving will decline by 17 percent.

Motor Enclosures -In general, there are two classifications for motor enclosures: open motor and totally enclosed motor. These two categories are further broken down as follows.

Dripproof -This prevents liquid drops and solid particles from entering the motor at angles up to 15^{o} from vertical.

Splashproof -This prevents liquid drops and solid particles from entering the motor at angles up to 100^{o} from vertical.

Totally Enclosed (TE) -There are no ventilation openings in the motor housing, but it is not air tight. Totally enclosed motors are used in dirty, damp, or oil-contaminated locations

Totally Enclosed, Non-Ventilated (TENV) - Unlike regular TE motors, this is not equipped with an external cooling fan. Cooling depends on convection or on a separately driven device.

Explosion Proof -This is a totally enclosed motor designed to withstand an internal explosion of specified gases or vapors without allowing flame to escape through the housing. There are different classes of explosion proof motors. The most common is class I-Group D.

Service Factors - Fractional horsepower motors commonly have service factors between 1.0 and 1.35. That is, some motors can tolerate up to a 35 percent overload on a continuous basis. The service factor is normally stamped on the motor nameplate and can be referred to easily.

Only open motors have service factors. For totally enclosed and explosion proof motors, the implied service factor is 1.0. When substituting a totally enclosed motor for an open one, the difference in service factors may require use of the next larger size.

Temperature Rise - Most motors are provided with class A insulation, which is designed for a maximum motor temperature (measured in the windings) of 221°F (105°C). The shell temperature should normally be under 180°F (82°C). Class B insulation is designed for a maximum temperature at the windings of 266°F (130°C). If the motor's temperature consistently exceeds the maximum recommended for its insulation class, its life (normally 20,000 hours) will be halved for every additional 10°C.

Motor design generally anticipates a maximum ambient temperature of 104°F (40°C). Slightly higher temperatures can be tolerated at less than maximum load.

As a motor's internal temperature changes, the speed changes even though the load remains the same. This is because higher temperatures increase the motor winding resistance. This reduces motor current, which in turn reduces the magnetic field strength in the motor. As a result, the torque falls and the speed drops in accord with the motor speed-torque curve.

If the total winding temperature does not exceed design limits and the unit is delivering rated output, a fractional horsepower motor normally operates near its rated speed. However, where high ambient temperatures require insulation rated to 130°C (class B insulation), lower speeds are the result.

This can occasionally cause problems when the motor is operating at a service factor greater than one - that is, is running slow because it is loaded beyond the full load point. If these factors are further compounded by low line voltage (speed varies nearly as the square of the voltage), the motor may stall. Under these conditions, a higher-horsepower motor will be required.

Thermal Overload Protection - Electric motors can overheat because of high ambient temperatures, continuous stall, abnormal voltages, restricted ventilation, or an overload. To minimize motor failure, a thermal overload cut-out device is used. Two basic types are available. One type is sensitive to temperature only, the other to both temperature and current.

Some overload protectors provide running as well as locked rotor protection. These meet the requirements of Underwriters Laboratories Incorporated. Others-for instance, in permanent split capacitor and shaded pole motors-provide only locked rotor protection. All provide some degree of protection at all times. However, the protection is not always strict enough to ensure normal motor life-particularly if the motor is operating close to its limits. Tighter limits, though, could lead to an excessive number of nuisance trips.

Some thermal overload devices feature automatic reset. That is, they automatically reset themselves after a cooling period. No human intervention is required-often an advantage when compressors or pumps must run unattended. But automatic reset devices should never be used when an unexpected restart might be dangerous. And if the fault still exists at automatic restart, the motor will cycle on and off until the fault is corrected.

Other Factors

Contamination - As noted earlier, oil vapors may cause contamination or deterioration of products and materials in some applications. Oil-less compressors are generally required in such cases. They also reduce maintenance costs by eliminating periodic filling of lubricators.

Available oil-less compressors include most of the piston and diaphragm types, regenerative blowers, and some of the rotary vane designs.

Pulse-free Delivery - Applications requiring continuous, pulse-free air delivery-without the extra cost and space requirements of a receiver tank-usually dictate that the compressor use a rotary pumping mechanism, such as the rotary vane design. An additional advantage of this compressor type is that noise and vibrations are considerably below those of reciprocating designs. The regenerative blower type also has pulse-free delivery and low vibration, but the high impeller speeds can generate high-pitched noise.

Mounting Space - Very often, compressor selection is further limited by available space at the installation site. For lower pressure systems, the compact rotary vane compressors usually require less space for a given free air capacity than do piston or diaphragm designs.

Compressor Size Selection

By the time all the foregoing application needs are resolved, the choice of a particular type of compressor very often has been limited to just one or two designs. Then the next step is to determine optimum compressor size.

Like other equipment decisions, compressor size selection also begins with application needs. Each application requires a specific volume of air over a specific time at a specific pressure. Then it's basically a matter of matching these specifics against the cfm and psi ratings of available air compressors.

Determining Free Air Consumption

Free air, as already defined, is air at atmospheric pressure. The free air volume is obtained by using the gas laws (Section I and Appendix) to convert volume at the actual working pressure and temperature to volume at atmospheric pressure and ambient temperature.

Three steps are required to determine the system's rate of free air consumption:

- Identify the volume of free air required by each operating device during its work cycle. This can be done by calculations based on handbook formulas, or from free air curves, catalog data or tests made with a prototype system.
- Multiply by the number of work cycles per minute.
- Total the results for all the work devices in the system. If a receiver is not used, it is also necessary to check that possible peak demands do not exceed the average demand calculated. If they do, then it will be these peak demands that govern the capacity requirements.

Effects of Receiver Recharging - if a receiver tank used in an application requires rapid on/off or load/unload cycling operations, the compressor must be sized with extra capacity so the receiver can be recharged without interrupting normal system operation.

A rule-of-thumb for determining the volume of free air required for receiver tank recharging is to multiply the receiver volume by the pressure difference (in atmospheres) between the cut-in and cut-out pressures. Then divide the result by the charging time permitted and select the compressor that will deliver that cfm at the cut-out pressure.

Effects of Initial Receiver Charging - in some intermittent systems, extra-large receivers are used to permit a longer off period. While this reduces the number of duty cycles during a given interval, the time required for initial charging of the large receiver may be too long to permit normal system operation.

A practical solution is to select a compressor with greater capacity than otherwise required simply to reduce the initial charging time. The increased capacity will also reduce the portion of the duty cycle the compressor is on.

The time required for initial receiver charging can be estimated by dividing the amount of free air to be pumped into the receiver
by an average delivery rate: rate at low pressure plus rate at high pressure divided by two.

Determining Available Compressor Capacity

In determining an air compressor's ability to meet specific system needs, rated capacity is generally determined from curves or performance tables. These show the actual free air delivery at rated speed for discharge pressures ranging from 0 psig to the maximum pressure rating. (Table 4 lists typical capacity ranges for various types of compressors.)

Keep in mind that horsepower and displacement are not suitable sizing criteria. Such factors can lead to large sizing errors because they do not provide accurate measures of the compressor's actual delivery capabilities.

For protection against problems caused by leaks, unusual operating conditions, or poor maintenance, size selection should provide some extra capacity. Generally, the compressor actually selected should have a rated free air capacity 10 to 25 percent greater than the system's actual rate of free air consumption. This precaution also allows for possible future system expansion or field modifications.

Effects of Duty Cycle - When an intermittent pressure rating is used in selecting a compressor, the restrictions on duty cycles established by the compressor manufacturer must be strictly observed. For example, intermittent pressure ratings for Gast compressors are based on a 50/50 (10-minute on/10-minute off or open) duty cycle. This 10-minute off period is a minimum necessary to allow the compressor to cool. Longer off periods can be obtained by increasing the receiver volume or by increasing the difference between cut-in and cut-out pressures of the pressure switch.

The 10-minute on period is the maximum based on temperature rise. Increasing the compressor capacity will shorten the on periods, but very short on cycles can cause problems when pressure is controlled by starting and stopping the compressor's drive motor. This is because too-frequent starts can actuate the motor's thermal overload mechanism, temporarily interrupting electrical power. This is best solved by leaving the pump on and using a solenoid valve.

Other Selection Considerations

Volumetric Efficiency -The theoretical pumping capability of a positive displacement compressor is the product of its displacement (the total volume transported by its pumping elements in one revolution) times its speed in revolutions per minute. Displacement is determined by the size and number of the pumping elements (piston chambers, vane compartments, etc.). Displacement alone should not be used as a sizing parameter, since it is a theoretical value that does not take into account pumping losses.

A pumping device's volumetric efficiency is how close it comes to delivering the calculated volume of fluid. Volumetric efficiency varies with speed, pressure, and type of pump. It is found by comparing actual delivery with computed delivery using this formula:

$$\text{Volumetric Efficiency (\%)} = \frac{\text{Free Air Delivered in cfm}}{\text{Theoretical Capability in cfm}} \times 100$$

The volumetric efficiency of an air compressor is highest at 0 psig-that is, when it is discharging to the atmosphere. Volumetric efficiency becomes progressively lower as pressure increases.

This drop reflects a loss in rated capacity at higher pressures, mainly because of increases in the pressure of air trapped in the "clearance volume" and to an increase in internal leakage or slippage. The temperature and density of the incoming air also affect the efficiency.

Drive Power Requirements - For any given compressor, the power required for compression depends on the capacity of the compressor, the pressure at which it is operated, and the efficiency of the cooling method. Additional power is needed to overcome inertia and the frictional effects of startup, as well as mechanical resistance while driving the compressor at rated speeds.

The manufacturer's drive power recommendations for a given compressor usually includes both a suggested drive speed and a horsepower requirement. The drive speed will be that at which the rated capacities are developed. The horsepower specification will be the maximum power required. This is usually the power required at maximum rated pressure but may occasionally reflect startup requirements.

The compressor manufacturer can generally provide performance curves that show the power requirements at rated speeds over a range of pressures. In some cases, curves may be available showing power requirements at different speeds for given pressures.

Adherence to these recommendations will assure satisfactory operation at any pressure within the compressor's range of operation.

Power Efficiency - Techniques to evaluate the efficiency with which a compressor uses power have been widely adopted. In general, these call for simultaneous measurements of cylinder volume and pressure, free air flow, temperatures, and input power.

Actual outputs calculated from these measurements are compared with theoretical values. This way the efficiency of the compressor, the compression process, and the overall installation can be determined.

A simple but relatively accurate comparison of the performance of different compressors is provided by the cfm of free air delivered per horsepower. This can be calculated directly from manufacturers' catalog data.

It's a simple procedure. First, find the cfm delivered at the required pressure. Divide this value by the horsepower at that pressure. This results in the actual quantity of free air per minute per installed horsepower. (Be sure to note, however, whether catalog data are based on cfm at actual pressure levels or at atmospheric pressure. The same reference level must be used for all compressors being compared.)

Since horsepower delivered is directly proportional to the product of gauge pressure and flow rate, cfm at a given pressure per input horsepower indicates power efficiency.

There is a relatively wide variation in the energy efficiency, defined as cfm of free air per horsepower, of the various compressor types, as Tables 5 and 6 indicate. This often enables the designer to reduce power requirements simply by switching to a different type of compressor.

Temperature Effects on Performance - High temperature is an air compressor's enemy. It can limit pressure capabilities, reduce delivery rates, and increase power requirements. Continued operation at high temperature accelerates wear and degrades the lubricant, causing bearing failures.

To avoid problems with high temperatures, the compressor's operating pressure should be held within the manufacturer's stated maximum pressure rating and duty cycle limitations. If the compressor must operate continuously at high pressures or temperatures, a heavy-duty water-cooled unit may be required.

Wherever possible, the compressor should be installed where its fan can draw in cool, clean air. Units powered by electric motors should not be installed where ambient temperatures are above 40°C (104°F).

In addition to the effects of a compressor's overheating, discharge of high temperature air can have a number of adverse effects on a pneumatic system: (a) reduced receiver storage capacity; (b) removal of volatile components from lubricating oil carried over into receiver and air lines; (c) increased moisture carried over into the air system. How to avoid these problems is discussed in Section III.

Table 2

Summary of Compressor characteristics

Class	Category	Type	Power Range (HP)	Pressure Range (PSI)	Advantages
Positive displacement compressors	Reciprocating	Piston Air-cooled	1/2 to 5000	10 to 15,000	Simple, lightweight
		Piston Water - cooled	10 to 5000	10 to 50,000	Efficient, heavy-duty
		Diaphragm	10 to 200	10 to 3,500	No seal, contamination-free
	Rotary	Sliding vane	10 to 500	10 to 150	Compact, high-speed
		Screw (helix)	10 to 500	10 to 150	Pulseless delivery
		Lobe, low-pressure	15 to 200	5 to 40	Compact, oil-free
		Lobe, high-pressure	7 1/2 to 3000	20 to 750	Compact high-speed
Non-positive Displacement compressors	Rotary	Centrifugal	50 to 20,000	40 to 2,000	Compact oil-free high-speed
		Axial Flow	1000-10,000	40 to 500	High-volume, high-speed
		Regenerative peripheral blower	1/4 to 20	1 to 5	Compact, oil-free high volume

Table 3

Pressure Ratings and Applicable Compressors

Required System Pressure (PSIG)		Compressor Types Available	
Continuous	Intermittent*	Preferred Type	Optional Type
100 to 175	175 to 200	Two-stage (piston) Piston (1 stage)	-
50 to 100	-	Rocking Piston (1 stage)	-
30 to 60	-	Diaphragm	Rocking Piston (1 stage)
25 to 30	25 to 30	Rotary vane (oil-lubricated)	Any of above
10 to 15	15 to 20	Rotary vane (oil-less)	Any of above
10	10	Rotary vane (oil-lubridated & oil-less)	Any of above
3.5	-	Regenerative (peripheral) blower	-

*Pressure ratings based on 10-minute or / 10-minute off duty cycle

Table 4

Capacity Ranges and Pressures

Available Compressor Types	Maximum Pressure (PSIG)		Range of Capacities (CFM Free Air at 0 PSIG)	
	Continuous	Intermittent	Smallest	Largest
Piston (1-stage)	50-100	50-100	1.3	11.0
Rocking Piston (1-stage)	10-100	10-100	0.6	3.25
Diaphragm (1-stage)	50-60	50-60	0.51	3.8
Rotary Vane (oil-lubricated)	10-25	12-30	1.3	55
Rotary Vane (oil-less)	10-15	10-20	0.35	55

Table shows pressures and range of capacities available from various types of Gast compressors

Table 5

Availability of some separate drive comrpessors

Compressor Type	Range of Capacities (CFM at 0 PSIG)	Drive Requirements H.P.	Speed
Piston (1-stage)	1.3 to 4.8 cu. ft.	0.3 to 1.1	2000 RPM (minimum 1000 RPM)
Rotary vane (oil-lubricated & oil-less)	0.35 to 55 cu. ft.	1/40 to 5	Most common 1725 RPM (range, 880 to 3450 RPM)

Table 6

Availability of Motor-Mounted Compressors

Compressor Type	Range of Capacities (CFM Free Air at 0 PSIG)	Motor H.P. Requirements
Piston (1- and 2-stage)	1.3 to 11.0	1/6 to 2
Rocking Piston	0.6 to 3.25	1/8 to 1/2
Diaphragm	0.8 to 3.9	1/16 to 1/2
Rotary vane (oil-lubricated & oil-less)	0.35 to 55	1/15 to 3/4

Summary of Pressure Sequence

The sequence of compressed air flow in a pneumatic system is shown again in Fig 28:

- **Air Compressor** - outside air is compressed to a pressure higher than that of the atmosphere.
- **Air Line Conditioners** - devices located at various points provide moisture removal, filtration, lubrication, etc.
- **Storage** (optional)-compressed air generated by the compressor may be stored in a receiver tank for use on demand.
- **Pressure/Flow Controls** - valves control the pressure reaching the actuator, hence the force it produces. Flow control valves determine actuator speeds.
- **Work Devices** - the actual work effects are obtained by supplying air to actuators and other air-operated devices.

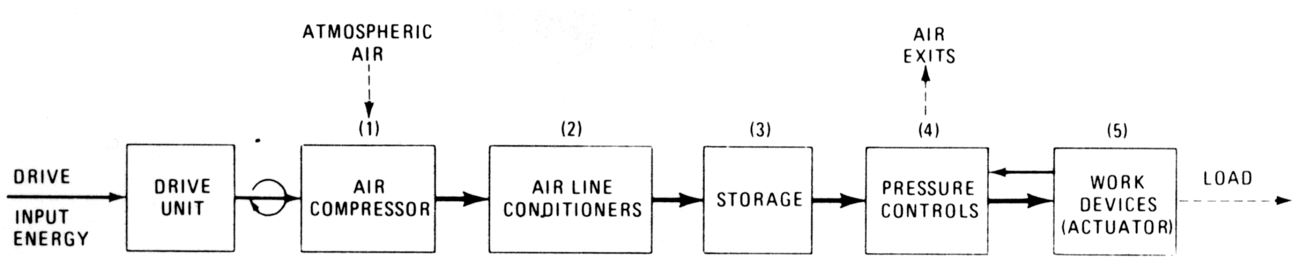
At the end of the actuator's work cycle, the "used" air is returned to the atmosphere, either directly through a control valve or through devices designed to reduce exhaust noise or remove oil vapor.

The Fig. 28 configuration shows how pneumatic power can be generated and tapped at will to provide a great variety of useful mechanical effects.

Conditioning Compressed Air

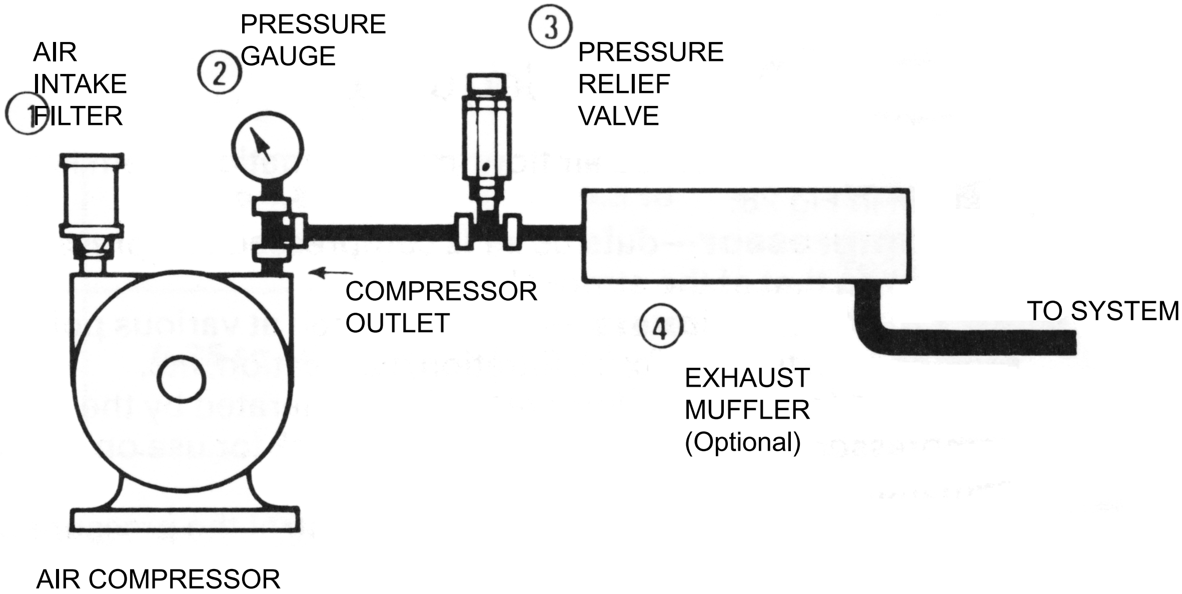
Many standard air line accessories are available to properly link the power source to the work device. In general, filtering and lubricating equipment should always be used in the air line ahead of working components.

Figure 28



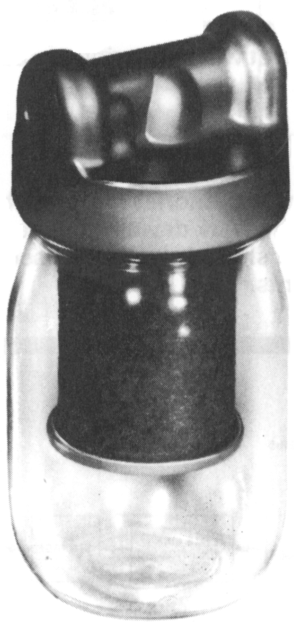
Summary of pressure sequence in a pneumatic system.

Figure 29



Typical conditioning devices used at the inlet and outlet of an air compressor.

Figure 30



Typical intake filter used to protect a pneumatic pump from contamination.

Fig. 29 shows some typical conditioning devices mounted in a compressed air system, and following are descriptions of some of the most common conditioning devices.

Intake Filter - The condition of the air entering a compressor can have a critical effect on efficiency. An intake filter (Fig. 30) should be used in all pneumatic systems to remove dust and other foreign particles from air entering the compressor. The amount of dirt, moisture, or corrosive gases in the inlet air will dictate the type of filtering or purification equipment required.

For proper service, the filter element must be cleaned or replaced periodically. Frequency depends on ambient conditions. A dirty filter element can restrict air flow through the compressor, causing overloading and consequent overheating.

Air Dryer-Some pneumatic systems cannot tolerate even a trace of moisture. While a mechanical filter removes most of the solid and liquid contaminants from the air, it is not very effective for removing water and oil vapors. These may later condense if temperatures are lower somewhere downstream. An air dryer prevents condensation by reducing the humidity of the air stream. It is normally installed after the air storage tank (receiver) and ahead of the pressure-reducing valves.

A very practical type of dryer is the dessicant unit, which uses a moisture-absorbing chemical. After entering, the air first passes through a mechanical filter, then through a dessicant bed (usually in pellet form) where virtually all the moisture is removed.

Figure 31

Combination filter and moisture separator.

Water vapor also can be removed by condensation. The air is first chilled in a unit similar to an aftercooler to condense the vapor; then, after the water is removed, the air is allowed to warm up again.

Filters, especially coalescing filters, are reasonably effective in removing mists of tiny water droplets. But they provide no protection against later condensation of water vapor.

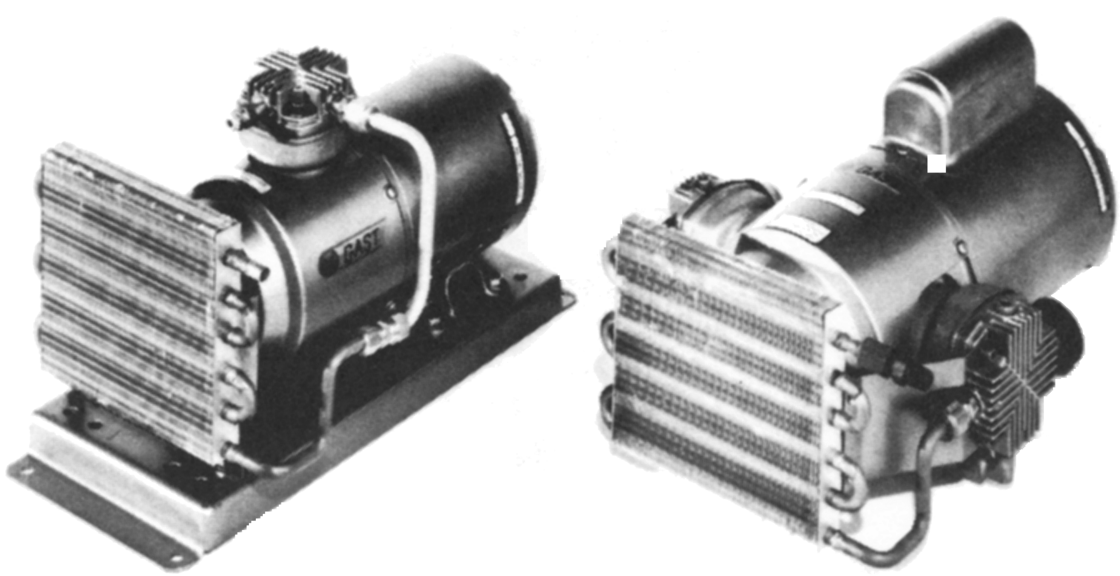
Aftercooler - The immediate purpose of an aftercooler (Fig. 32) is to cool the air emerging from a compressor. By Charles' Law, this increases the amount of air that can be stored at a given volume and pressure. Both air-cooled and water-cooled models are available, depending on the amount of cooling required.

Another problem the aftercooler solves is moisture. As the air entering the compressor is compressed, so is the accompanying water vapor. The partial pressure of water vapor increases sevenfold, for example, as the air containing it is compressed to 100 psig.

This often results in water vapor pressures exceeding the room -temperature saturation pressure. While the water may remain in gaseous form at the higher temperatures characteristic of the compressor exhaust, it is likely to condense out at some more troublesome spot as the air cools downstream.

If air remains in the receiver long enough to cool, it will drop its moisture there, where it can be readily drained off by a stopcock in the bottom of the tank. But if the air remains in the receiver only a short time, or if no receiver is used, then it will probably be necessary to use an aftercooler following the compressor. By cooling the air immediately, the aftercooler makes sure the moisture condenses out where it can be most easily dealt with.

Figure 32



Typical radiator-style aftercooler.

Figure 33

Typical pneumatic-system lubricator.

Lubricator - A variety of lubricators are used with oil-lubricated compressors to provide a fine mist of oil in the pumping chamber. The device in Fig. 33 regularly injects atomized oil into the air stream entering the compressor.

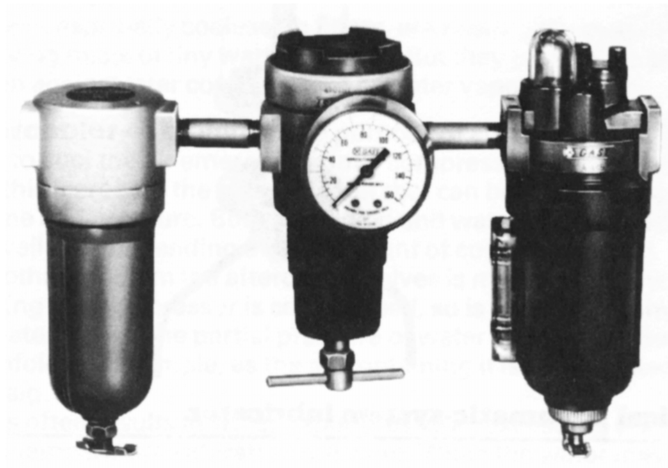
Lubricators should be placed near the equipment they lubricate. They should always be downstream of the filter and regulator. (Of course, lubricators are not required for - and should never be used on - oil-less compressors.)

FRL Unit - These are filter/regulator/lubricator combinations that function as a unit (Fig. 34). The filter is placed first in the lineup to prevent foreign matter from causing a malfunction of the regulator. The lubricator is usually last because oil mist or droplets tend to be deposited in turns or valves in the system. (The regulator is a control device described in the section on pressure control devices.)

Exhaust Muffler - This device (Fig. 35) acts as a noise absorbing unit. It also aids in trapping much of the oil and moisture that may leave the compressor with the exhaust air. The exhaust muffler is located between the pressure relief valve and the rest of the system, sometimes connected to the exhaust ports of directional control valves. These valves are discussed in a separate section.

The muffler must be of a material and design that will tolerate the working pressure. The element is often made of felt, styrofoam, paper, screen, or coarse sintered metal, with internal baffles to provide even distribution of air flow.

Figure 34



Typical commercial FRL (filter/regulator/lubricator) unit.

Figure 35

Typical exhaust muffler.

Transmission of Pressure

Pascal's Law states that pressure applied to a confined fluid (liquid or gas) is transmitted with equal intensity to every point on the containing surfaces. The forces generated by compressed air can be transmitted over considerable distances (up to 200 ft.) with small losses-up, down, over, under, and around corners. With special equipment, pressure can be transmitted for very great distances; for example, in the air brake system on a mile-long freight train. A pressure change is transmitted from one part of a system to another at the speed of sound. Transmission of the sound wave, however, requires air movement. And most practical systems are based on bulk air flow from compressor to work device. The remainder of this section describes how air behaves during pressure transmission and bulk flow.

Resistance and Pressure Drop - Various components in a pneumatic system create resistance. As air flows, energy must be expended to overcome this. The result of this expended energy is pressure drop (or loss) between adjacent points in the system. Therefore, sizing to compensate for pressure losses is an important design step.

A pressure drop occurs only while air is moving through the line. Factors affecting system resistance and pressure loss include: inside diameter of the transmission line, its internal roughness, angles or bends, transmission distance, flow volume, fitting and connection restrictions, and leaks throughout the system. Valves and other components also create pressure drops during system operation.

The pressure drop between points immediately before and after a constriction can be used to measure the air flow rate. More specifically, the flow rate is proportional to the square root of the pressure difference (within certain limits). In general, flow rate can be measured by the pressure drop across an orifice or venturi. Conversely, the pressure differential necessary to produce a given flow can be readily calculated.

Storage of Compressed Air

Air Receivers-There are two methods of supplying compressed air to a pneumatic system: It can come directly from the air compressor, or it can be supplied from a pressure storage vessel or air receiver.

An air receiver is generally defined as a storage tank containing pressurized air, used to provide a convenient on-demand source of pneumatic power. As compressed air is transmitted to an air receiver, the amount of air in the tank is progressively increased and greater pressures are developed.

Usually the air receiver is the last stop before air is transmitted to the distribution system. Its primary function is to act as a reservoir to accommodate sudden or unusually high system demands. This allows the compressor to be sized for average, rather than peak, demand. It also avoids having the compressor start and stop frequently as demand fluctuates.

An air receiver also provides some conditioning of the air. It damps pulsations from the discharge line of reciprocating compressors and provides a smoother flow of air to the system. By allowing the air to cool, it also precipitates moisture-especially if an aftercooler has not been used. Sometimes small air receivers are placed at regular intervals along the line to act as moisture collecting tanks.

Control of Compressed Air

Valves and control devices in compressed air systems fall into three general categories: those that control pressure, those that control direction of air flow, and those that control flow rate.

Pressure Control Devices

Pressure Relief Valve-This device prevents pressure from building up beyond a safe, preset limit. Located between the compressor outlet and any valve or other restriction, this device allows air at excess pressure to escape to the atmosphere. It does not close down a system. It allows just enough air to escape to return the system to designed pressure.

Safety Valve-This device is similar to the pressure relief valve, except that when it opens it does so to its full capacity. It thus provides a large, rapid reduction in pressure when required. A safety valve closes down a system by allowing all the air to escape.

Air Pressure Regulator-This device is desirable in most pneumatic systems to maintain the specific, precise pressure required by a particular application, regardless of the rise and fall of line and/or receiver pressure as the compressor cuts in and out. An air pressure regulator is often used to reduce the pressure to a value safe for certain components, or to ensure that the exact amount of thrust needed is delivered to a cylinder. The device is actually a pressure-reducing valve that operates by restricting air flow into the downstream leg of a circuit. It is not capable of relieving upstream pressure.

Directional Control Valves

These are "air gates" that open or close to direct the flow into one of several paths. Valve classification is by: the number of positions to which the valves can be actuated, moved or shifted; total number of ports; and the type of actuator that shifts the valve (pilot-operated, for example). Types of directional control valves include two-, three-, and four- way valves.

Two-way Valve - The minimum requirement for any valve is that it have at least two ports, producing a flow path through the valve. The two-way type is designed to open or block this single flow path. Flow can be in either direction. These are essentially shutoff valves that totally open or close an air line.

Three-way Valve - in addition to its two extreme positions, this valve also has a center (neutral) position. This may enable a cylinder to stop in midstroke, for example. Three-way valves usually control single-acting cylinders that extend under fluid pressure, but retract by some other means (such as load, spring, etc.).

Four-way Valve-This design is called four-way because it has four working connections (pressure inlet/cylinder blind end/ cylinder rod end/exhaust or tank). Four-way valves usually control double-acting cylinders having two working ports to extend and retract a cylinder rod. Often a throttling valve can be installed on each exhaust to permit individual speed adjustment for each direction of cylinder motion.

Rate Control Valves

The valves described above are designed to control the pressure or the routing of the forces produced by an air compressor. Rate control valves determine either the rate of air flow or whether any flow occurs at all.

Flow Control Valve-The basic function of a flow control valve is to maintain a constant flow rate, regardless of pressure level. This is necessary in many applications where the speed of a cylinder or air motor must be closely regulated.

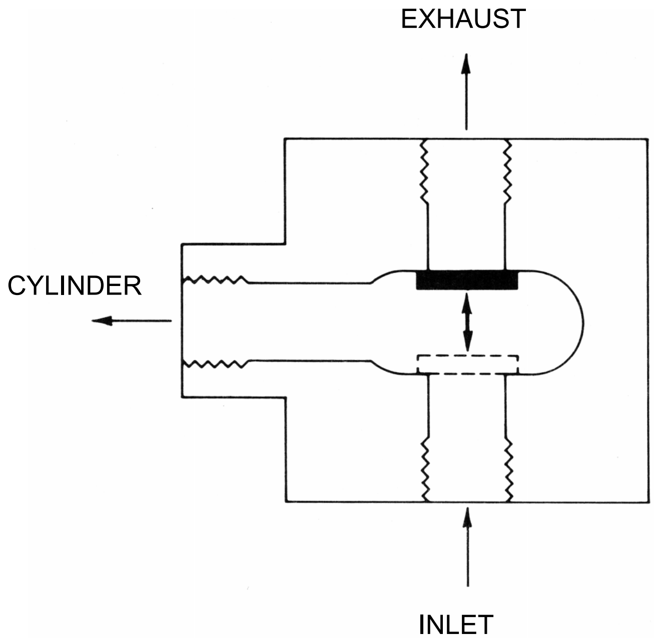
These valves govern flow by restricting air movement in one or both directions. The restriction may be fixed or variable. The more sophisticated valves also compensate for temperature fluctuations.

Check Valve - In its simplest form, a check valve is nothing more than a one-way valve. It permits free flow in one direction only and blocks flow in the reverse direction.

Quick Exhaust Valve-This device is installed at the exhaust port of a cylinder so exhaust air will immediately escape to the atmosphere, rather than having to flow back through the piping and exhaust through the control valve. A quick exhaust valve can improve system performance by increasing cylinder speed.

When air enters the valve inlet (Fig. 36), it pushes a floating rubber disc to close the exhaust port and air flows into the cylinder. When the main directional valve is shifted to shut off inlet flow and retract the cylinder, inlet pressure drops below cylinder pressure. The excess cylinder pressure pushes the disc over against the inlet, opening an exhaust path directly to the atmosphere.

Figure 36



Quick exhaust valve. Floating disc blocks either inlet or exhaust port, depending on cylinder pressure level.

Effects of Pressure: Force

Force is basically a function of pressure and area. Total force (in pounds, for example) is obtained by multiplying the pressure difference in pounds per square inch (psi) by the area of the working surface in square inches.

Pneumatic circuits are capable of exerting great forces, especially when using pneumatic cylinders. There is virtually no practical limit on the pressures that can be developed with relative ease-up to 200 psi gauge. And literally thousands of psi can be developed with special equipment to meet unusual needs.

The final objective, work, is accomplished by the forces produced by a pressure differential on opposite sides of a movable barrier such as a piston, diaphragm, or air motor vane. Work is equal to the force exerted multiplied by the distance the barrier is moved.

Work Devices - Among the multitude of work devices that can be integrated into a pneumatic system are "actuators," which convert fluid power energy into linear or rotary force and motion. An actuator is selected to provide a required torque (or force) and speed at an appropriate operating pressure.

Linear actuators, usually consisting of a piston/cylinder mechanism, convert fluid energy from the compressor into linear mechanical force and motion.

Rotary actuators produce a rotary output by rotating the output shaft through a fixed arc. The instantaneous torque produced varies directly with the fluid pressure applied to the actuator.

Air motors produce a continuously rotating output. Torque depends on both pressure and speed. (See the accompanying volume, AIR MOTORS HANDBOOK, for details.)

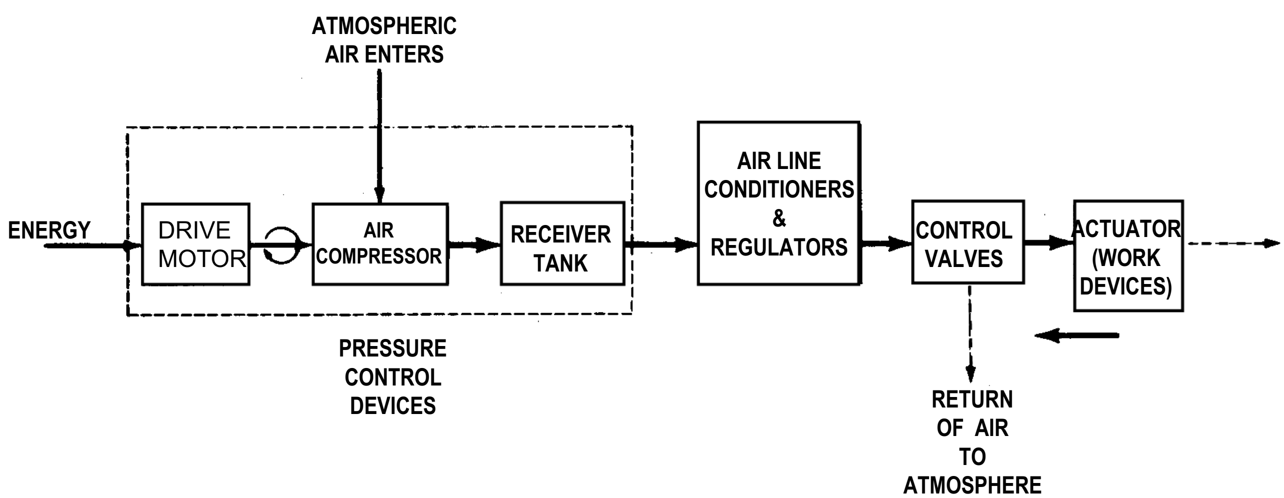
Pneumatic Power Supply Systems

So far we have considered an air compressor simply as a device that converts mechanical force and motion into pneumatic power. Now it will be helpful to consider a compressor in terms of a unified system. The dotted lines in Fig. 37 enclose the basic elements of a pneumatic power supply system.

A basic working definition of a power supply system is: the air compressor, a drive unit, and related storage and pressure control devices linked together to provide the pressure required to meet specific application requirements.

The power supply system is tailored to do a job at certain cost and performance levels. Compressed air provided by the compressor can be delivered directly to the actuator to produce work, or it can be stored in a receiver tank. In either case, the pressure must be effectively controlled. As discussed elsewhere in this volume, either on/off or load/unload cycling may be used, or excess air may be vented by a pressure relief valve.

Figure 37



Dotted line enclose the "power supply" portion of a general pneumatic system.

Receiver tanks are always required when a compressor is intended to work on an intermittent-duty basis (which is desirable because off periods permit effective cool-down of the compressor, reducing wear and providing cooler system air). Use of a receiver tank also permits a power supply system to consistently supply pressures near the compressor's maximum rating.

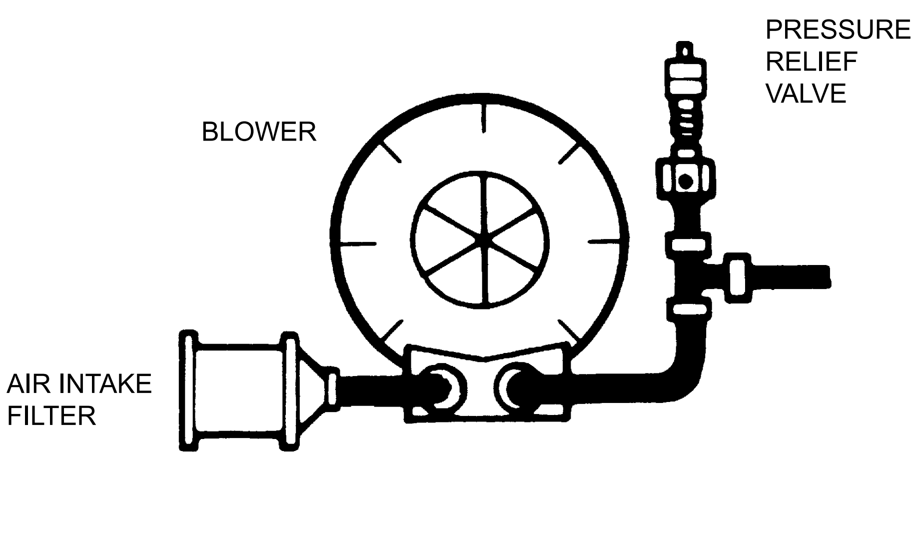
Without a receiver, demand might limit pressure buildup by bleeding off air as fast as the compressor can supply it. A safety valve is always used with a receiver to provide independent overpressure protection.

The methods used for pressure control depend greatly upon the system's design. Some representative types of pneumatic systems are described here. Although not shown in these examples, the input energy is provided by a drive motor or power takeoff in every case. If no lubricator is shown at the inlet, assume that the compressor is oil-less.

Minimum System -Some simple applications, such as aerating liquids or powders in a tank, require continuous air flow at low pressure. The "minimum" power supply system shown in Fig. 38 could handle the job. It is composed of a compressor with drive unit, an air intake filter to protect the compressor against dust and grit, and a relief valve to provide pressure protection in the event of a downstream restriction. The relief valve, normally set several pounds above the "use" pressure, must be installed in the compressor's discharge line.

Unlike the safety valve (which opens to its full capacity at a preset pressure to provide rapid pressure relief, then closes completely when the pressure drops to a safe level), the pressure relief valve provides modulated venting of excess pressure to the atmosphere. Thus, it maintains pressure at a relatively constant maximum level.

Figure 38



Basic low pressure pneumatic power supply system.

The continuous pressure-controlled relief valve action eliminates the need for elaborate on/off or load/unload control equipment. A disadvantage is that the compressor must work harder and longer if it is set to work at maximum pressure.

For smooth, pulse-free air flow in a simple aerating application such as this, either a rotary vane compressor or a regenerative blower would be practical.

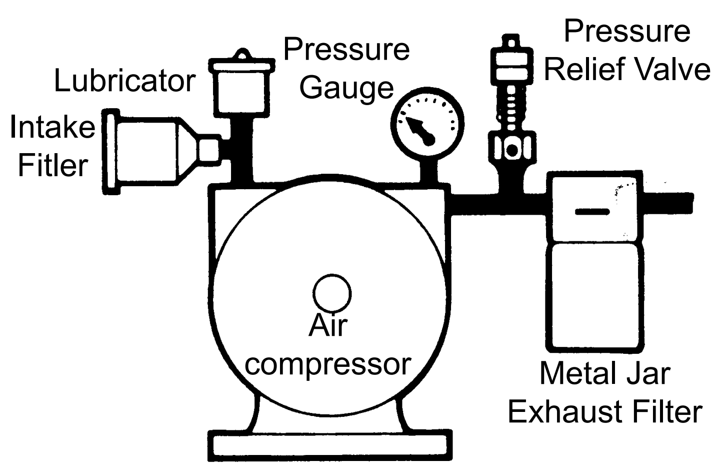
Pressure Gauge-The preceding power supply system for aeration is easily extended to actuate work devices. The pressure gauge shown in Fig. 39 is used to monitor system pressure for safety and efficiency. Other gauges may be used at various points throughout the system, but one at the compressor outlet-to give a true reading of the load under which it is working-is almost standard. Excessive pressure can harm the compressor, the system in which it is used and the materials being processed.

The figure shows how the relief valve is located in the delivery line between the compressor and the first downstream restriction, which, in this case, is an exhaust filter used to remove oil vapors. The layout shown is particularly useful for transportable pneumatic systems run by gasoline engines.

Pressure Control/Storage-Receiver tanks are normally used to store air in quantities required for intermittent-duty applications. They can also be used in continuous-operation systems to dampen pulsations, to ensure an ample air supply and to precipitate moisture from the air.

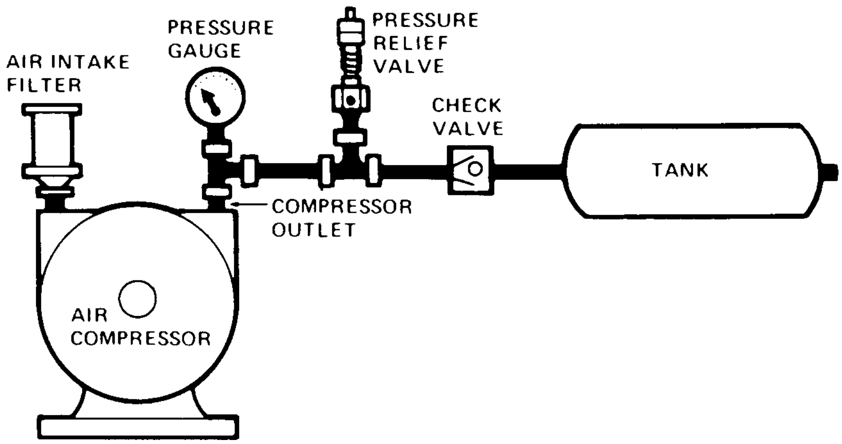
Let's assume that a rotary vane compressor is being used in Fig. 40. Since this type of compressor is valveless, a check valve must be installed ahead of the receiver tank inlet. This prevents high pressure air from blowing back from the receiver into the compressor.

Figure 39



Addition of a pressure gauge makes it possible to monitor system pressure, increasing safety and efficiency. The exhaust filter is used to remove oil vapors.

Figure 40



With a receiver tank, compressed air can be stored for use as required in intermittent-duty applications. The check valve prevents backflow when the compressor is not operating.

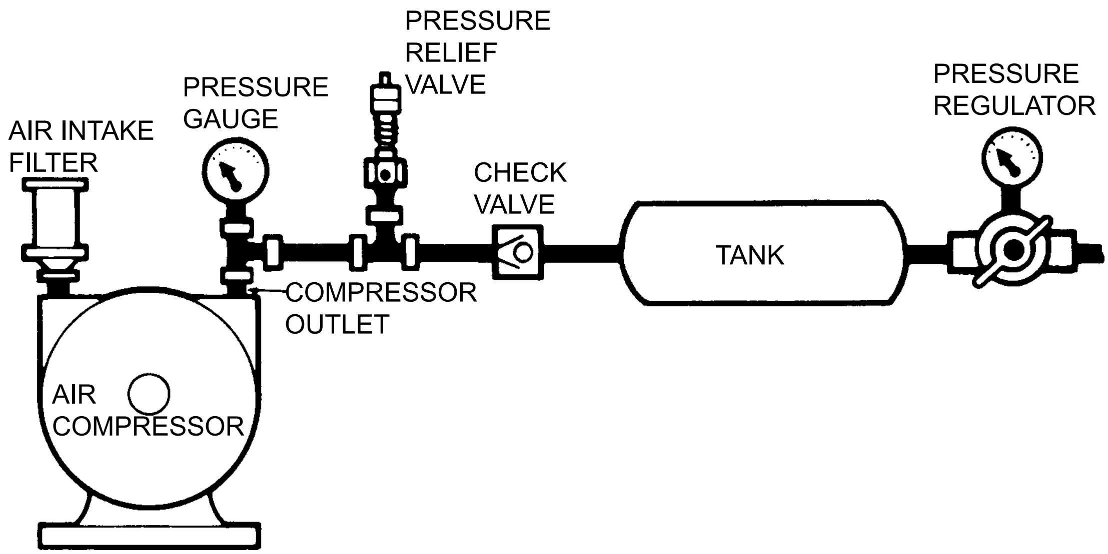
Even when using compressors with valves, such as the piston and diaphragm types, check valves are still included in the system as an extra protection against back flow.

When a receiver is used in a system designed for continuous duty, the system overpressure can be controlled by a pressure relief valve in the line from the compressor to the receiver. If redundant overpressure control is required (as it is by some safety codes for larger systems), a safety valve, set several pounds higher than normal "use" pressure, can be installed in the receiver discharge line. When a safety valve is used, it always should be set at the lowest pressure that would reflect a hazardous overpressure condition.

Downstream Pressure Control-in many continuous-duty applications, it is necessary or desirable to get the maximum use out of receiver tank capacity. This is done by maintaining the storage pressure at a higher level than the downstream system pressure. In Fig. 41 the pressure regulator is installed between the receiver tank outlet and the rest of the system. Except for this, the system is similar to the preceding power supply system.

Pressure regulators are pressure-reducing valves that work by restricting and blocking flow into the downstream leg of the circuit. They cannot relieve upstream pressure. If there are large differences between the pressure requirements of the individual system devices, additional pressure regulators can be added where needed downstream.

Figure 41



A pressure regulator installed downstream of the receiver tank allows air to be stored at a higher pressure than that at which it will be used. This allows use of a smaller tank.

A pressure regulator never should be used for upstream protection without a pressure relief valve or a safety valve at the receiver tank or near the compressor to provide the necessary overpressure protection.

Two-Compressor System-Some power supply requirements may call for a little ingenuity if a particular type of compressor of the required capacity is not available. As Fig. 42 shows, two compressors can be used to feed the same receiver tank.

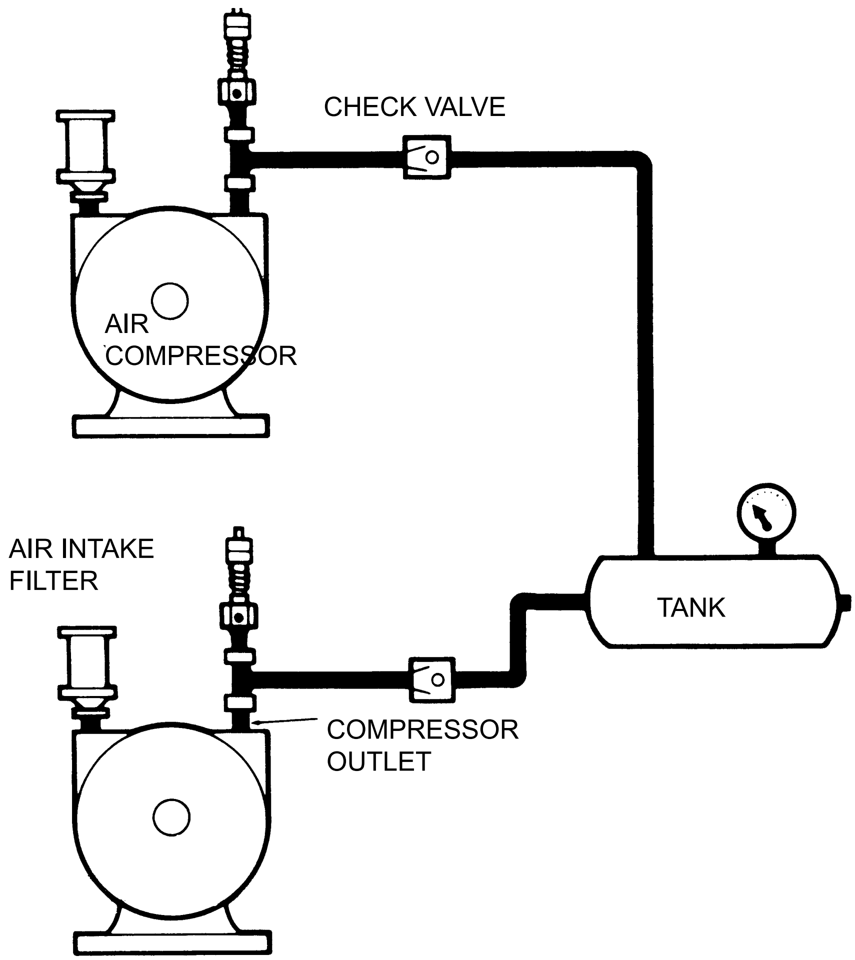
This is often done when it is necessary to provide more rapid initial charging of the receiver, or to increase flexibility in meeting large, abrupt capacity demands. Such a system also can be used to provide reliable backup operating capacity.

The two compressors may be of the same or different sizes to meet overall application needs. Usually they will be separately powered, and they may supply air to separate receiver tanks. Receiver connections should be made through check valves (as in the figure) to allow shutdown of either compressor without interrupting normal system operation.

The compressors should be equipped with separate on/off or load/unload control systems as described for the preceding systems. Different cut-in points for each compressor can be used to provide an automatic increase in capacity to meet heavier demands.

Although not shown in Fig. 42, a single safety valve installed in the receiver will provide overpressure protection. A better design is to install separate safety valves in the discharge line from each compressor to provide redundant overpressure protection.

Figure 42



A two-compressor power supply system designed to meet unusual capacity requirements.

Equipment used to generate vacuum, as noted earlier, is similar to air compressors. It's even possible to generate compressed air or vacuum with the same machine, depending on how it is installed. Vacuum pumps generally can be considered as compressors in which the discharge, rather than the intake, is at atmospheric pressure.

Recall that the essence of air compression is the increased number of molecular impacts per second. Conversely, the essence of vacuum generation is the reduction of these impacts. The vacuum in a chamber is created by physically removing air molecules and exhausting them from the system.

Removing air from the enclosed system progressively decreases air density within the confined space, thus causing the absolute pressure of the remaining gas to drop. A vacuum is created.

Because the absolute maximum pressure difference that can be produced is equal to atmospheric pressure (nominally 29.92 in. Hg at sea level), it is important to know this value at the work site.

For example, a pump with a maximum vacuum capability of 24 in. Hg cannot generate a 24-in. vacuum when the atmospheric pressure is 22 in. Hg (as in Mexico City, for instance). The proportion of the air evacuated will be the same, however. This pump therefore will pull 22 x 24/29.92 or 22 x 24/30 = 17.6 in. Hg vacuum in Mexico City.

Vacuum Pumps: Basic Operation

A vacuum pump converts the mechanical input energy of a rotating shaft into pneumatic energy by evacuating the air contained within a system. The internal pressure level thus becomes lower than that of the outside atmosphere. The amount of energy produced depends on the volume evacuated and the pressure difference produced.

Mechanical vacuum pumps use the same pumping mechanism as air compressors, except that the unit is installed so that air is drawn from a closed volume and exhausted to the atmosphere. A major difference between a vacuum pump and other types of pumps is that the pressure driving the air into the pump is below atmospheric and becomes vanishingly small at higher vacuum levels. Other differences between air compressors and vacuum pumps are:

- The maximum pressure difference produced by pump action can never be higher than 29.92 in. Hg (14.7 psi), since this represents a perfect vacuum.
- The mass of air drawn into the pump on each suction stroke, and hence the absolute pressure change, decreases as the vacuum level increases.

- At high vacuum levels, there is significantly less air passing through the pump. Therefore, virtually all the heat generated by pump operation will have to be absorbed and dissipated by the pump structure itself.

Vacuum Stages

As in compression, the vacuum-generating process can be accomplished in just one pass through a pumping chamber. Or several stages may be required to obtain the desired vacuum.

The mechanical arrangements are also similar to those for air compression. The discharge port of the first stage feeds the intake port of the second stage. This reduces the pressure, and hence the density, of air trapped in the clearance volume of the first stage. The net effect is, using a Gast diaphragm pump as an example, that the second stage boosts the vacuum capability from 24 to 29 in. Hg.

Oil-Less vs. Oil-Lubricated Vacuum Pumps

As with compressors, the application normally dictates whether an oil-less or oil-lubricated vacuum pump should be used. Either type may be used in many applications.

Oil-Less - Oil-less pumps are almost essential when production processes cannot tolerate any oil vapor carry over into the exhaust air. They also can be justified on the basis of avoiding the cost and time of regularly refilling the oil reservoirs. This is particularly important when the pumps are to be mounted in inaccessible locations.

Modern piston pumps have rings of filled Teflon, which provide hundreds of hours of duty, depending on ambient temperature and air cleanliness. Diaphragm and rocking piston pumps are designed to be oil-less.

Oil-Lubricated - The oil-lubricated types have distinct advantages if proper maintenance is provided. They can usually provide about 20 percent higher vacuums because the lubricant acts as a sealant between moving parts. And they usually last about 50 percent longer than oil-less units in normal service because of their cooler operation. They also are less subject to corrosion from condensed water vapor.

Positive Displacement Vacuum Pumps

Vacuum pumps fall into the same categories as air compressors do. That is, they are either positive displacement or nonpositive displacement machines. A positive displacement pump draws a relatively constant volume of air despite variations in the vacuum levels.

As with air compressors, the principle types of positive displacement vacuum pumps are the piston, diaphragm, rocking piston, rotary vane, lobed rotor, and rotary screw designs. The basic mechanics of each are described in Section 11. The remarks below cover aspects that apply to vacuum applications.

Reciprocating Piston Pumps -The primary advantage of the piston design is that it can generate relatively high vacuums from 27 to 28.5 in. Hg-and do so continuously under all kinds of operating conditions. The major disadvantages are somewhat limited capacities and high noise levels, accompanied by vibrations that may be transmitted to the base structure. In general, the reciprocating piston design is best suited to pulling relatively small volumes of air through a high vacuum range.

Diaphragm Pumps -The diaphragm unit creates vacuum by flexing of a diaphragm inside a closed chamber. Small diaphragm pumps are built in both one- and two-stage versions. The single stage design provides vacuums up to 24 in. Hg, while the two stage unit is rated for 29 in. Hg.

Rocking Piston Pumps -This design combines the light weight and compact size of the diaphragm unit with the vacuum capabilities of reciprocating piston units. Vacuums to 27.5 in. Hg are available with a single stage; two-stage units can provide vacuums to 29 in. Hg. Air flows, however, are limited, with the largest model available today (a twin-cylinder model) offering only 2.7 cfm.

Rotary Vane Pumps -Most rotary vane pumps have lower vacuum ratings than can be obtained with the piston design: only 20 to 28 in. Hg maximum. But there are exceptions. Some two stage oil-lubricated designs have vacuum capabilities up to 29.5 in. Hg. (Also see the section on medium-vacuum pumps.)

The rotary vane design offers significant advantages: compactness; larger flow capacities for a given size; lower cost (about 50 percent less for a given displacement and vacuum level); lower starting and running torques; and quiet, smooth, vibration free, continuous air evacuation without a receiver tank.

Rotary Screw and Lobed Rotor Pumps - Vacuum capabilities of rotary screw pumps are similar to those of piston pumps, but evacuation is nearly pulse-free. Lobed rotor vacuum pumps, like the corresponding compressors, bridge the gap between positive and nonpositive displacement units. Air flow is high but vacuum capabilities are limited to about 15 in. Hg. Capabilities can be improved with staging.

Nonpositive Displacement Vacuum Pumps

Like the corresponding compressors, nonpositive displacement vacuum pumps use changes in kinetic energy to remove air from a system. The most significant advantage of this design is its ability to provide very-high-volume flow rates-much higher than possible with any of the positive displacement designs. But because of their inherent leakage, these machines are not practical for applications requiring higher vacuum levels and low flow rates.

The principle types of nonpositive displacement vacuum pumps are the centrifugal, axial-flow, and regenerative designs. Single-stage regenerative blowers can provide vacuums up to 7 in. Hg with flows to several hundred cfm. Vacuum capabilities of the other designs are lower unless they are multistaged.

Evaluating Vacuum Pump Performance

This section covers important vacuum pump performance characteristics used in evaluating particular types and sizes. Actual pump selection, covered in a separate section, will be based on how these characteristics relate to the intended application.

The primary performance criteria cover just three characteristics:

- Vacuum level that can be produced.
- Rate of air removal.
- Power required.

Somewhat less critical are temperature effects and certain other characteristics.

In general, the best pump for a specific job is the one having the greatest pumping capacity at the required vacuum level and operating within an acceptable horsepower range.

Vacuum Level - A pump's vacuum rating is the maximum vacuum level for which it is recommended. The rating is expressed in in. Hg and is specified for either continuous or intermittent duty cycles.

Most vacuum pumps can't come near the theoretical maximum vacuum (29.92 in. Hg at sea level) because of internal leakage. For a reciprocating piston pump, for example, the upper vacuum limit may be 28 or 28.5 in. Hg, or roughly 93 to 95 percent of the maximum theoretical value.

Internal leakage and clearance volume establish the highest vacuum a pump can produce. For some pumps, this is also the vacuum rating.

In other types, however, heat dissipation is a problem. For these, the maximum vacuum rating might be based on allowable temperature rise. For example, good wear life for some rotary vane pumps requires a maximum 180^{0}F (82^{0}C) rise in casing temperature at the exhaust port. Vacuum ratings will be based on this temperature rise. They probably will be higher for intermittent than for continuous duty.

The vacuum rating listed for a pump is based on operation at 29.92 in. Hg. Operating where atmospheric pressure is lower will reduce the vacuum the pump can produce. An adjusted vacuum rating for such locations can be determined by multiplying actual atmospheric pressure by the ratio of the nominal vacuum rating to standard atmospheric pressure:

$$\text{Adjusted Vacuum Rating} = \text{Actual Atmospheric Pressure} \times \frac{\text{Nominal Vacuum Rating}}{\text{Standard Atmospheric Pressure}}$$

Air Removal Rate -Basically, vacuum pumps are rated according to their open capacity, which is the volume of air (expressed in cfm) exhausted when there is no vacuum or pressure load on the pump.

Effectiveness of the vacuum pump in removing air from the closed system is given by its volumetric efficiency, a measure of how close the pump comes to delivering its calculated volume of air. Volumetric efficiency for a positive displacement pump is given by the general equation on page 62. With vacuum pumps, this equation is applied in two different ways:

- **True (or Intake) Volumetric Efficiency** -The volume of air removed during a given time period is converted to an equivalent volume at the temperature and absolute pressure existing at the intake.
- **Atmospheric Volumetric Efficiency** -The volume of air removed by the pump is converted to an equivalent volume at standard conditions (14.7 psi and 68^0 F).

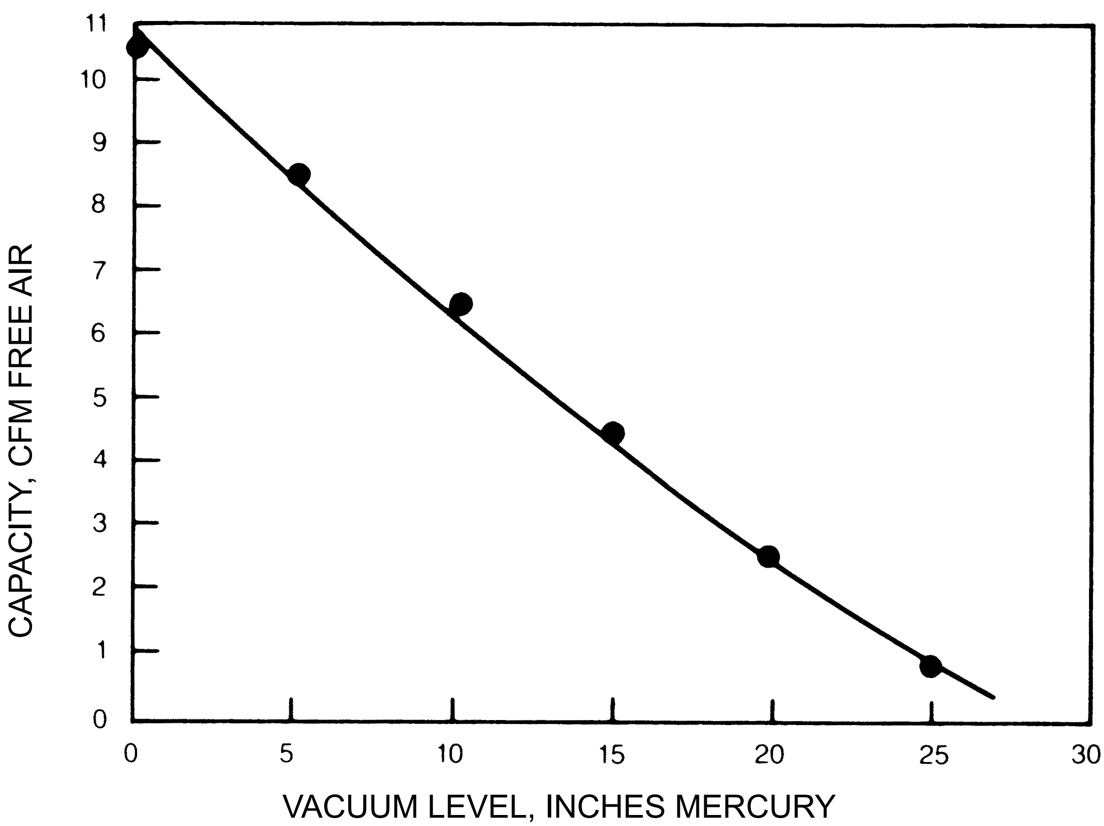
In either case, the displacement is the total volume swept by the repetitive movement of the pumping element during the same time period (usually one revolution). With various vacuum pumps having the same displacement, it is the difference in volumetric efficiencies that accounts for the difference in free air capacities. Since these differences exist, pump selection should be based on actual free air capacity rather than on displacement.

In short, the air removal rate is a measure of vacuum pump capacity. And the capacity of standard machines must be determined from the manufacturers' tables or curves showing cfm of free air delivered at rated speed for vacuum levels ranging from 0 in. Hg (open capacity) to the maximum vacuum rating. Free air capacity at different speeds for a given vacuum also may be included in the manufacturers' performance curves.

As shown in Fig. 44 and 45, the rated capacity of any pump is highest at 0 in. Hg and will drop rapidly as the vacuum level increases. This reflects a drop in both volumetric efficiency and the volume of air that can be drawn into the pumping chamber.

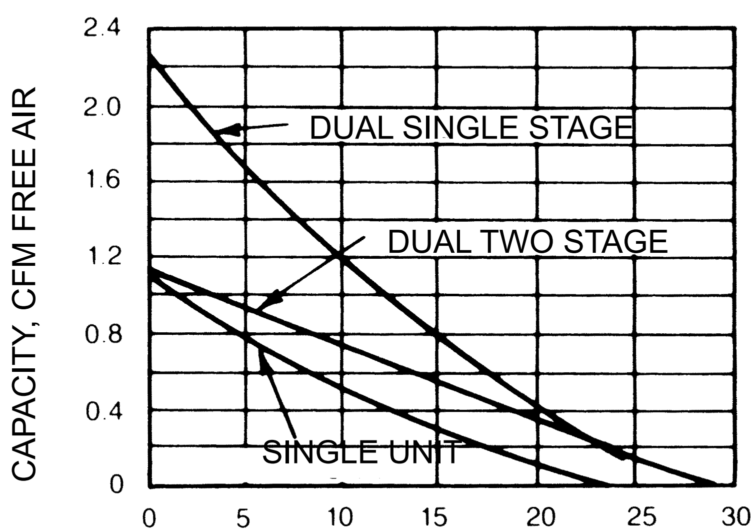
To repeat, a basic characteristic of positive displacement pumps is that capacity drops as the vacuum level increases. Fig. 44 shows this very clearly for a piston pump. The same principle holds for diaphragm pumps.

Figure 44



Capacity vs. vacuum level for a single-stage reciprocating piston pump.

Figure 45



Capacity vs. vacuum level for different types of diaphragm vacuum pumps: single cylinder, twin cylinders in parallel (dual single-stage), and twin cylinders in series (dual two-stage).

In Fig. 45, the single unit designation represents a single-chamber, single-stage pump. The dual single-stage unit has twin chambers operating in parallel. In the dual two-stage unit, the twin chambers operate in series.

Recall that the staging process produces higher vacuum levels because the first stage exhausts into a second stage already at negative pressure. This causes a reduction in absolute pressure of the air trapped in the clearance volume (the space between piston and cylinder head at the time of full compression). But as Fig. 45 shows, the basic capacity of the dual two-stage machine is reduced by 50 percent compared with the dual single-stage pump of the same size.

Fluid Power Horsepower - Different techniques have been developed to evaluate the efficiency of energy use by a vacuum pump. Most vacuum pump manufacturers catalog their test results, including brake horsepower (actual hp) and cfm vs. vacuum level. Fairly accurate evaluations of power needs can be made from such information sources.

For example, the relative efficiency of different pumps can be obtained by calculating the cfm of free air removed per horsepower. Or input horsepower can be compared to the "fluid power horsepower" delivered, which is proportional to the product of gauge vacuum and air flow rate. All comparisons must be made at the same specific vacuum level, usually at 20 in. Hg or above.

Drive Power Requirements -The drive unit must be able to meet the pump's peak power requirement. In other words, it must be powerful enough to assure satisfactory operation under all rated operating conditions. This includes providing adequate energy to overcome friction and inertia effects at startup.

The power requirements of a vacuum pump are relatively low, compared with those of an air compressor. The primary reason is the low compression work requirement. Both the volume flow rate and the pressure difference across the machine are much lower than in a compressor.

When the pump operates near atmospheric pressure, for example, the mass flow rate (cfm free air pumped) is at its highest, but pressure differences between inlet and outlet are very small. The amount of work that must be added per pound of air is therefore very low.

At higher vacuum levels, the amount of work that must be done increases because of the larger difference between inlet and discharge pressure. The mass flow rate or cfm free air pumped drops progressively, however. The total amount of compression work thus remains very low.

Drive Speeds -in addition to actual brake horsepower required for various vacuum levels, catalogs will generally show the speed required to develop various rated capacities.

Effects of Temperature Rise

Vacuum pump performance can be significantly affected by heating of the pump itself. At higher vacuum levels, there is very little air flow through the pump. Most of the air has been exhausted. There is thus very little transfer of internal heat to this remaining air.

Much of the heat generated by friction must be absorbed and dissipated by the pump. Since some pumps generate heat faster than it can be dissipated, a gradual rise in pump temperature results, drastically reducing service life.

One solution is to give careful consideration to pump ratings. For example, a continuous-duty pump should have a high maximum vacuum rating. On the other hand, an intermittent-duty pump may be specified for high vacuum levels if the off period is adequate for effective cool-down of the pump. Complications may arise if the maximum on period greatly exceeds the off cooling period.

Whenever possible, vacuum pumps should be operated with load/unload cycling rather than on/off cycling. When a pump is unloaded, the atmospheric-pressure air being drawn through it carries the accumulated heat away rapidly. When a pump is shut off with vacuum inside, however, heat loss is much slower because it occurs only through the outside of the casing.

Vacuum Pump Selection

The previous section describes how the designer evaluates performance based on vacuum level, air flow, power requirements and temperature effects. This section covers the factors involved in applying the basic characteristics to particular operation and application needs. In short, we wish to narrow down the selection process to a single type, size, and horsepower for the vacuum pump and related system components.

Vacuum Level Factors

Basically, the selection of the appropriate type of vacuum pump is determined by comparing the application's requirements with the maximum ratings of available commercial pumps (Table 7).

But how are required working vacuum levels determined? When a mechanical force is required, the necessary working vacuum is determined in a way similar to establishing pressure requirements of air-operated devices.

Increasing the size of the device to increase its area reduces the working vacuum required. The requirements of specific vacuum devices in the line can be determined by calculations based on handbook formulas, theoretical data, catalog data, or performance curves and tests made with prototype systems.

The primary restriction on the type of vacuum pump that can be used in a given application is the actual system vacuum level determined by the designer. When this level is relatively low (about 15 in. Hg), the designer has a large variety of different types and models of pumps from which to choose. But as vacuum levels rise, the designer has fewer and fewer options, sometimes to a choice of one.

Maximum Vacuum Rating -There are practical limits on the degree of vacuum that can be economically produced to accomplish work. These limits represent the maximum vacuum capabilities of the mechanical pumps used to remove air from the system.

Depending on the type of pump involved, this limit ranges from 20 to 29.0 in. Hg. Very sophisticated and costly equipment is required to obtain higher vacuum levels.

Representative of available commercial vacuum pumps, Table 7 summarizes the Gast line. The Diaphragm and rocking piston types provide the highest vacuum ratings. Also, they are generally preferred for continuous-service applications. Capabilities of oil-lubricated rotary vane pumps, however, approach these levels.

System Control Factors

If the system's vacuum level is controlled by a relief valve, the maximum vacuum required should be selected on the basis of the highest working level of any single air device in the system. But if the system vacuum is controlled by automatic on/off or load/unload cycling of the vacuum pump, the maximum vacuum required is equal to the cutoff value of the control.

Temperature Factors

Temperature is an important consideration from two standpoints:

Environmental Temperature -For ambient temperatures above 100^{o} F (38^{o} C), select a pump rated for higher vacuum operation and provide some external cooling.

Internal Pump Temperature -Ope ration at higher vacuums increases pump temperature and can be the most severe limiting factor on pump operation. Heavy-duty pumps with cooling can operate continuously. But light-duty pumps can operate at maximum vacuums for only short periods; they must be allowed to cool between cycles.

Miscellaneous Type Selection Factors

After the basic step in matching vacuum level requirements with the maximum vacuum ratings of available pumps, the selection process proceeds by determining if any of several factors may influence the decision.

Need for Uncontaminated Air -This can apply to the intake portion of the system, where grit could enter and harm the pump mechanism. More often it applies to the exhaust portion of a system, for example, in a food processing plant where dirty air or oil vapors can contaminate products or materials. The most straightforward solution is the selection of an oil-less vacuum pump.

Maintenance-Free Operation -Nothing mechanical is absolutely "maintenance-free." But if we restrict the term to lubrication, then an oil-less vacuum pump can best satisfy this need, since periodic oiling is not required.

Pulse-Free Air Flow -Rotary vane and nonpositive displacement machines have smooth, continuous air removal characteristics without the extra cost and space requirements of a receiver tank.

Minimum Vibration/Noise -Rotary vane pumps have lower noise and vibration levels than reciprocating machines. The regenerative blower is also basically vibration-free, but the impellers may generate high-pitched noise.

Space Limitations -Again, the rotary vane design is often selected because of its relative compactness. If higher vacuums are required, the rocking piston could be suitable.

Vacuum Capacity Factors

The optimum pump size for an application is determined by comparing the rate at which air must be removed from the system with the capacities of various commercial pumps available (Table 8). In general, small capacity and large capacity pumps that have the same maximum vacuum capabilities will pull the same vacuum on a closed system. The small pump will simply require more time to reach maximum vacuum.

To directly compare vacuum pump and compressor rating data, the air removal rate is calculated in cubic feet of free air per minute (just as in pressure systems). To determine the free air that must be removed, the volume is multiplied by the vacuum level in atmospheres. The latter is obtained by dividing the gauge vacuum (in in. Hg) by standard atmospheric pressure (29.92 in. Hg). Therefore, the formula for free air is:

$$\text{Free Air} = \text{System volume} \times \frac{\text{Gauge pressure}}{29.92}$$

As with compressors, it is necessary to first calculate the free air removal for each work device over a full work cycle. This value is multiplied by the number of work cycles per minute, and the requirements for all the work devices are totaled.

Open Capacity Rating -The above total is then matched with the capacity ratings of available equipment. Generally, to accommodate possible leaks, the selected vacuum pump should have a capacity rating 10 to 25 percent above the air removal rate actually required.

The capacity of a vacuum pump is generally given by manufacturers' curves or performance tables showing cfm of free air pumped (at rated speed) against inlet conditions ranging from 0 in. Hg (open capacity) to the maximum vacuum rating. The capacity rating at the operating vacuum level is generally used to actually select the size.

Keep in mind that there is some flexibility in sizing selection. If a required type of pump is not available in the required size, then two or more smaller pumps can be teamed to provide the necessa ry capacity.

Purnpdown Rate -The above approach is difficult to apply in many vacuum applications because air removal occurs over a wide range of vacuum levels. Since volumetric efficiency changes with vacuum level, there is no one capacity rating against which the free air requirement can be compared.

One approach to this problem uses the equation:

$$t = \frac{v}{s} \ln \frac{p^*}{p}$$

Here, v is the system volume; p and p* are the initial and f inal pressures, respectively, in absolute units; t is the time available to pump the system from p to p*; and S is the pump capacity in cfm at the actual pressure in the system. If capacities are published in "free air" cfm, as they usually are, they must be converted to "actual pressure" cfm by multiplying by 29.92/(29.92-gauge pressure).

But this procedure leaves open the question of just what capacity S represents. In fact, it is the average capacity between p and p*, a value not readily available. Capacities at intervals of 5 in. Hg are commonly published, however. It is then possible to apply this equation piece-by-piece.

The open capacity is used for pumping down to 2.5 in. Hg, the capacity at 5 in. Hg gauge for p = 2.5 in. Hg gauge and p* = 7.5 in. Hg gauge (remember these numbers must be converted to absolute units), and so on. The pumpdown time thus calculated for a given pump is compared with that required by the application to determine whether the pump has adequate capacity.

Warning Note -Horsepower and displacement should not be used as sizing criteria, since they do not provide accurate measures of the quantity of air actually pumped through the machine.

Other Pump Sizing Factors

After matching the required rate of air removal with available vacuum pump open capacities, the final decision may be influenced by one or more factors.

Effects of Receiver Tank -If a receiver is used with either on/off or load/unload control of the vacuum pump, pump size generally can be smaller because (in most cases) the pump will have a longer time to evacuate a given amount of air.

Sometimes, however, a pump that's otherwise adequate will take too long to initially evacuate the receiver. The time required can be calculated by the method given under "Pumpdown Rate." If this is unacceptable, then a larger pump must be chosen. (Of course, there rarely would be much point in using a receiver if it required a larger pump than would be needed without it.)

In some intermittent-duty applications, it may be desirable to install an extra-large receiver tank to permit a longer off time-a longer cooling period. But this naturally increases the time required for initial pumpdown. If this is unacceptable, one solution is to increase pump capacity. This will reduce both the time required for initial receiver evacuation and the proportion of its duty cycle that the pump spends on.

Effects of Intermittent Duty Cycle -Very short on/off cycles can cause serious problems. When vacuum is controlled by constantly starting and stopping a pump's drive motor, the motor's thermal overload device may be tripped. This will temporarily interrupt the power and result in a pump outage.

If a vacuum/pressure switch is used to control the duty cycle, the off time interval can be extended by either increasing the receiver tank volume or by increasing the range between cut-in and cut-out switch settings. The on portion of the duty cycle can be shortened by increasing pump size.

In general, heavy-duty pumps can operate at maximum vacuum continuously. Light-duty pumps can operate for extended periods of time.

When an intermittent-duty vacuum rating is specified, the restrictions must be strictly observed. Gast intermittent vacuum ratings, for example, are based on a 10-minute on/10-minute off duty cycle. The maximum on period is established so that the pump will withstand the accompanying temperature rise. And the 10-minute off period assures enough time to cool the pump.

Drive Power Selection

A vacuum pump doesn't care how it is driven, so the decision should be based on practical and economic considerations.

After the type and size decisions have been made, based on required vacuum level and flow rate, the job of determining the correct operating speed and horsepower required to drive the vacuum pump is relatively simple. For piston vacuum pumps, a general rule is that about 1 horsepower is needed for each 20 cfm of air pumped.

A major factor in the selection process is deciding whether to have the drive unit mounted directly on the vacuum pump or installed as a separate system element. A third alternative is that the motor can be eliminated if the application provides a rotating shaft, a gasoline engine, or perhaps a special electric motor that will always be running when the vacuum pump is on.

Motor-Mounted Pumps -if a vacuum pump of the required type and size is available with its own integrally mounted electric motor drive, then there is no drive selection problem because the combination is designed to function as a unit. The pump is literally built around the motor.

For example, the rotary vane pump has its rotor installed directly on the motor shaft, and the rest of the pump is securely anchored to the motor frame. There is no need for baseplate mounting or for a power transmission component.

Motor-mounted units are much more compact and lightweight than separate-drive pumps. The end plate on the newer rotary models can be easily removed to expose the vanes for inspection or replacement.

The only drive system problems associated with a motor-mounted pump are those of supplying and controlling the required electrical power. But there is such a wide range of standard and special voltage motors available with the pumps that supply problems are generally minimal. Motor control problems are similar to those in regular heavy-duty industrial applications.

Table 9 lists some representative motor-mounted vacuum pumps. The motor horsepower is included to indicate the range of power requirements.

Separate-Drive Pumps -Separately driven pumps can be connected to the drive unit by either a belt and pulley or a coupling. When powered by belt drives, operating speeds are infinitely variable within design limitations.

Unlike motor-mounted units, a separate-drive system requires an appropriate driving device to generate the required speed and horsepower as established by the pump manufacturer. Vacuum pumps designed for use with separate-drive units are usually foot-mounted. Additional components required for separate mounting, such as baseplates and belt guards, usually can be obtained from the pump manufacturer.

Table 10 lists some representative separate-drive vacuum pumps. Speed and power requirements are included to indicate the kind of basic information the designer needs to develop an effective separate-drive system.

The rated speeds cataloged by vacuum pump manufacturers determine the rate of air removal. At lower operating speeds, both capacity and required horsepower will be proportionately reduced. In general, operation of separate-drive pumps at speeds substantially higher or lower than rated may cause problems. Always contact the pump manufacturer for guidance when the system is to be operated at other than rated speed.

Summary of Vacuum Pump Selection Factors

These basic questions should be answered before deciding which vacuum pump is best suited for a particular application:

• What degree of vacuum is required?

• What flow capacity (cfm) is required?

• What horsepower and speed requirements are needed to meet vacuum level and capacity values?

• What power is available?

• Will duty cycle be continuous or intermittent?

• What is the atmospheric pressure at the work site?

• What is the ambient temperature?

• Are there any space limitations?

Table 7

Vacuum Ratings and Applicable Vacuum Pumps

Max. Vacuum Rating (In. Hg) Continuous	Types of Vacuum Pumps
27.5 to 28.5	Piston (multistage)
25.5 to 29	Rocking piston
24 to 29	Diaphragm (single & multistage)
10 to 28	Rotary vane (oil-lubricated)
15 to 26	Rotary vane (oil-loss)
7	Regenerative peripheral blower

Table 8

Pump Capacities and Applicable Vacuum Pumps

Vacuum Pump Types	Maximum Vacuum Level (in. Hg)		Range of Capacities (CFM Free Air at 0 in. Hg.)	
	Continuous	Intermittent	Smallest	Largest
Piston	27.5-28.5	-	1.3	10.5
Rocking Piston	25.5-29.0	-	1.22	2.7
Diaphragm	23.5-29.0	-	0.49	3.6
Rotary Vane (oil-lubricated)	10-28	25-28	1.3	55
Rotary Vane (Oil-less)	15-27	15-27	0.35	55

Table 9

Availabililty of Motor-Mounted Vacuum Pumps

Vacuum Pump Type	Range of Open Capacities (cfm)	Motor H.P. Requirements
Piston (1-stage) (2-stage)	1.8 to 10.5 cfm 1.15 and 2.30 cfm	1/6 to 3/4 1/8 and 1/4
Rocking piston (1-stage) (2-stage)	1.12 to 1.6 cfm 1.25 to 2.7 cfm	1/8 to 1/4 1/4
Rotary vane (oil-lubricated & oil-less)	0.60 to 10.0 cfm	1/15 to 3/4

Availability of Separate Drive Vacuum Pumps

Vacuum Pump Type	Range of Open Capacities (cfm)	Drive Requirements	
		Horsepower	Speed
Piston	1.3 to 4.8	0.13 to 0.255	200 RPM (minimum 1000 RPM)
Rotary Vane	0.35 to 55	1/40 to 3	800 to 3450 RPM (most common, 1725 RPM)

Conditioning Air Flow in Vacuum Systems

Various basic accessories are used with vacuum pumps to properly condition the flow of air. They are described in the following pages. Fig. 46 shows how some devices are installed in a typical system.

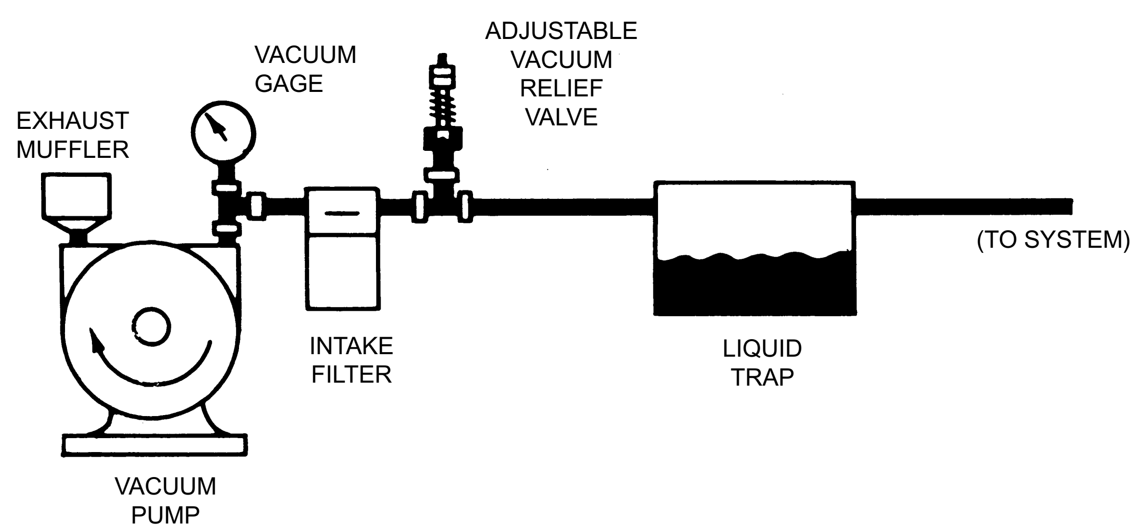
Intake filter -An intake filter should be used in virtually every vacuum system to prevent foreign particles from entering the pump. Filters are especially critical when a vacuum system must function in an environment containing a lot of dust, sand, or similar airborne solids. A range of filters, graded by the size (in microns) of the particles they can capture, is available.

A filter is normally installed at the pump's intake port. Since a clogged filter eventually becomes a harmful flow restriction, it should be removed and cleaned regularly to avoid overloading the pump.

Liquid Trap/Mechanical Filter -in addition to an intake filter at the vacuum pump, a liquid trap or mechanical filter is required when the vacuum system is used in filling operations. The objective is to prevent any of the material's being drawn through the vacuum line from getting into the pump.

As shown in Fig. 46, liquid traps are simple tank or bottle-like devices that use gravity to prevent liquids from being sucked into the vacuum system. Mechanical filters (not illustrated) are generally large bag-type devices installed in the vacuum line to capture dry powder before it can be drawn into the pump. The accumulated materials often can be recycled.

Figure 46



Exhaust Muffler -The exhaust muffler is usually a low restriction, flow-through device designed to reduce the pump's exhaust noise. There are several kinds, but the simplest and most common functions as a low-pass acoustic filter-a resonant cavity with dimensions that absorb energy from all but the lowest frequency sound waves. These are often glass or plastic jars essentially identical to intake filters but located at the vacuum pump's exhaust port.

At times the distinction between filters and mufflers almost disappears. Typical filters muffle sound significantly, while mufflers often trap oil entrained in a pump's exhaust.

Vacuum Gauge -A vacuum gauge is not strictly a system "conditioning" device, but it is described here because it is essential for monitoring system performance. The gauge should be installed at or near the intake port of the pump (Fig. 46). If it's located elsewhere in the system, the reading may be inaccurate because of dirty or clogged filters, kinked vacuum lines, closed valves, or other problems.

An accurate vacuum readout simplifies the job of locating various system malfunctions. For example, if the gauge reading at the intake port shows the targeted value, the trouble must lie between the gauge and the work device. But if the intake gauge reading is lower than the established working level, then either the pump is not working properly or there is a leak somewhere in the system.

Blocking the line just before the gauge will restore the vacuum level if a leak is the problem. Then, by moving the block further and further toward the work device, the trouble spot can be pinpointed. Additional gauges elsewhere in the system can simplify the job.

Transmission of Vacuum

Air evacuated from the work device can be routed to the vacuum pump with lines similar to those used for compressed air-metal pipe, copper tubing, and noncollapsible rubber or plastic hose. The work forces generated by a vacuum pump can be transmitted a considerable distance.

Avoiding leaks is critical. All joints, seals, valves and connecting lines must be airtight. Any restriction along the line can significantly decrease pump performance. To minimize pressure drop (vacuum losses), pipe diameters and fittings should be selected on the basis of cfm of air to be carried.

Storage of Vacuum

Vacuum Receivers -Vacuum can be stored as readily as compressed air. As an alternate to a direct vacuum pump source, vacuum can be established upon demand by use of a receiver tank. The primary function of the tank is to act as a vacuum reservoir to accommodate sudden or unusually high system demands. It also prevents possible overloading of the pump.

A receiver is a common system element when vacuum pumps operate intermittently. In vacuum applications requiring high flow for short periods-with long intervals between demand-even a small vacuum pump, running continuously, can store the necessary vacuum capacity.

A check valve is normally used on the pump side of the tank, and an appropriate operating valve is used on the downstream side.

Control of Vacuum

A vacuum pump alone cannot control the vacuum it produces. Some external means, such as a relief valve, must be provided for controlling the vacuum so that the pump will not exceed safe levels.

Without such control, production of higher vacuums would be stopped at some point only by leakage into the system. While this might seem to provide a desirable built-in safety factor, the vacuum level may exceed the maximum vacuum rating of the pump. The result could be pump failure because of extreme heat buildup.

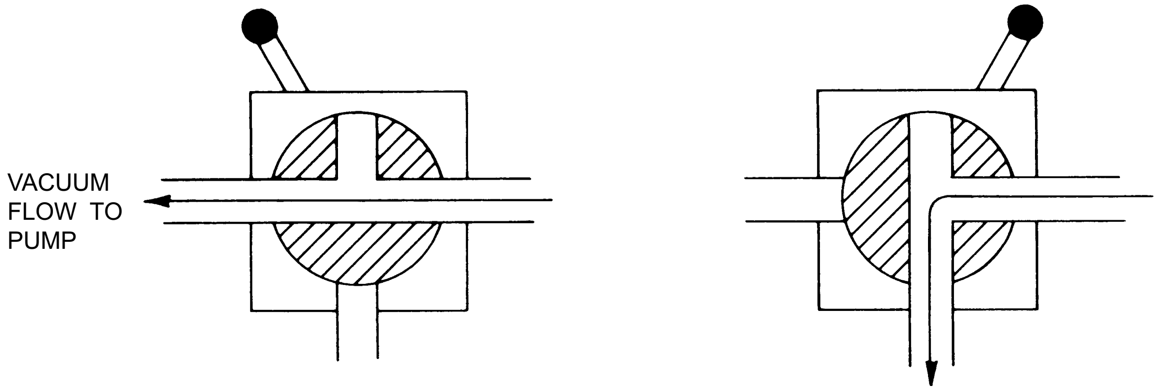
Following is a description of the major devices used to control and protect vacuum systems. (A separate section in this handbook describing Vacuum Power Supplies shows how these devices function in real systems.)

Vacuum Relief Valves -A vacuum relief valve limits the amount of vacuum pulled by the pump by allowing small amounts of air into the system whenever the valve's set point is exceeded. It does not close down the system.

The valve is adjusted so the pump pulls just the vacuum required by the application. This is desirable because operation at higher than necessary vacuums produces higher temperatures and greater workloads on the pump, thus reducing service life. It is especially important to use a vacuum relief valve with light-duty pumps, since they cannot withstand continuous operation at excessive vacuum levels.

Check Valves -The check valve allows free flow in only one direction. It is installed between the vacuum pump and the rest of the system to prevent back flow.

Figure 47



A) WORKING POSITION

B) PORT TO ATMOSPHERE

Manual control valve permits selective on/off operation of work device.

There are many types and styles of valves. For ordinary service, a cracking pressure of only 2 to 6 in. Hg is just enough to unseat the ball (or poppet) so a very small amount of flow can barely start. In some vacuum circuits, springless check valves are mounted vertically, with simply the weight of the ball or poppet holding it against its seat.

Flow Control Valves -To start and stop vacuum on demand, the atmospheric and vacuum line connections to the actuator are made through control valves, which are designed or specifically rated for vacuum service.

Fig. 47 shows how a manual three-way valve controls vacuum flow. A solenoid-operated valve could function the same way. A series of such valves could provide selective operation of multiple work devices, using a single vacuum pump.

As the A drawing shows, air flow is toward the pump, with the port to atmosphere closed. Drawing B shows release of the work device because the port to the atmosphere is open. This way, the selective operation of an individual work device will not affect the performance of any other work device in the same system.

Effects of Vacuum: Force

Vacuum systems used in industry are not designed to exert the large work forces possible with compressed air systems. Vacuum's operating range is only between zero and atmospheric pressure (approximately 14.7 psi at sea level and normal barometer); see Fig. 48. Recall that air compressors can generate many hundreds of psi.

Thus, 14.7 psi (or 29.92 in. Hg) is the maximum theoretical pressure difference that could be produced if all of the original air in a chamber were removed to produce an absolute pressure of zero psia. Although this level cannot be obtained, modern low pressure positive displacement vacuum pumps are efficient enough to remove about 95 percent of the atmospheric pressure in a chamber. That is, the equipment can provide a pressure difference (vacuum) of up to 13.8 psi.

Figure 48

Units			
PSIG	PSIA	In. Hg.	
0	14.7	0	← Atmospheric Pressure 14.7 PSI (SEA LEVEL)
-1	13.7	2.04	
-2	12.7	4.07	
-4	10.7	8.14	
-6	8.7	12.2	
-8	6.7	16.3	← Working Vacuum Level (Example)
-10	4.7	20.4	
-12	2.7	24.4	
-14	0.7	28.5	
-14.6	0.1	29.7	
-14.7	0	29.92	← Perfect Vacuum (Zero Pressure Reference)

Basic vacuum/pressure relationships in various units.

In vacuum lifting, the force of a system is a function of its maximum vacuum and the surface area to which the vacuum is applied. Each psi of vacuum on every square inch of surface area exerts a lifting pressure of one pound (by definition). To express this as an equation:

$$\text{Force (pounds)} = P_r \times a$$

where "P_r" is gauge pressure in psi and "a" is area in square inches.

In applying this equation with vacuum expressed as in. Hg, dividing by two gives an approximately correct result.

An actuator or work device is activated by switching the connection of its working volume from an atmospheric line to a vacuum line. At the beginning of the work cycle, much of the air within the actuator is drawn off, reducing the pressure inside the actuator below that of outside atmospheric pressure. At the end of the work cycle, the actuator is returned to its original pressure state by reestablishing the atmospheric connection. This eliminates the pressure difference that produced the work effect.

This principle is applied in thousands of ways in virtually every industry, ranging from simple materials transporting operations to sophisticated separation of chemical compounds in the laboratory.

For example, vacuum distillation is perhaps the most widely used laboratory vacuum process. It enables those substances that at normal pressures and higher temperatures would decompose to be distilled at room temperature.

Vacuum Power Supply Systems

The dotted lines in Fig. 49 enclose the basic elements of a vacuum power supply system. Following is a working definition of a vacuum power supply system: a system composed of a vacuum pump, a source of drive power, a receiver tank (optional), and various control and protective devices necessary to create a certain vacuum level to meet specific application needs.

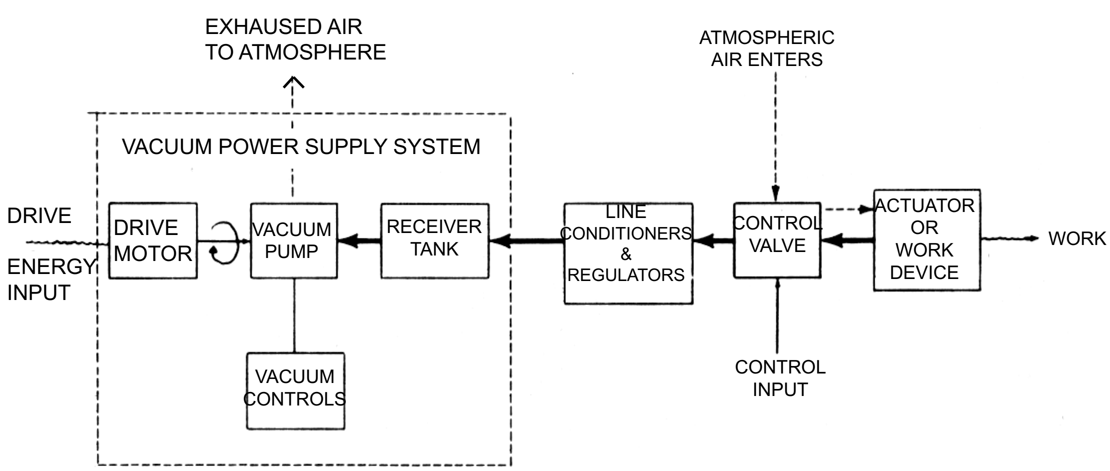
As with compressors, the mechanical energy input to the pump can be provided by an electric motor, engine, or power takeoff. The vacuum energy potential for doing work can be applied directly to an actuator via the vacuum line, or it can be accumulated for on-demand use by evacuating a receiver tank. In either case, the vacuum levels must be effectively controlled.

Following are some representative system configurations used to control vacuum levels.

Continuous Operation System -Fig. 50 shows a basic vacuum power supply system for continuous-duty service. For example, such a system might be used for dry-powder filling.

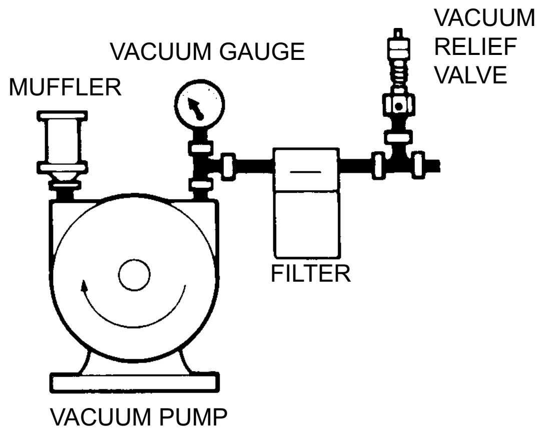
The adjustable vacuum relief valve controls the system's vacuum level by providing a modulated flow of atmospheric air into the system whenever the preset value is exceeded. The intake filter protects the pump from solid matter that might be pulled into the system. No lubricator device is shown in the figure since the pump is oil-less.

Figure 49



Dotted lines enclose the power supply section in a generalized vacuum system.

Figure 50

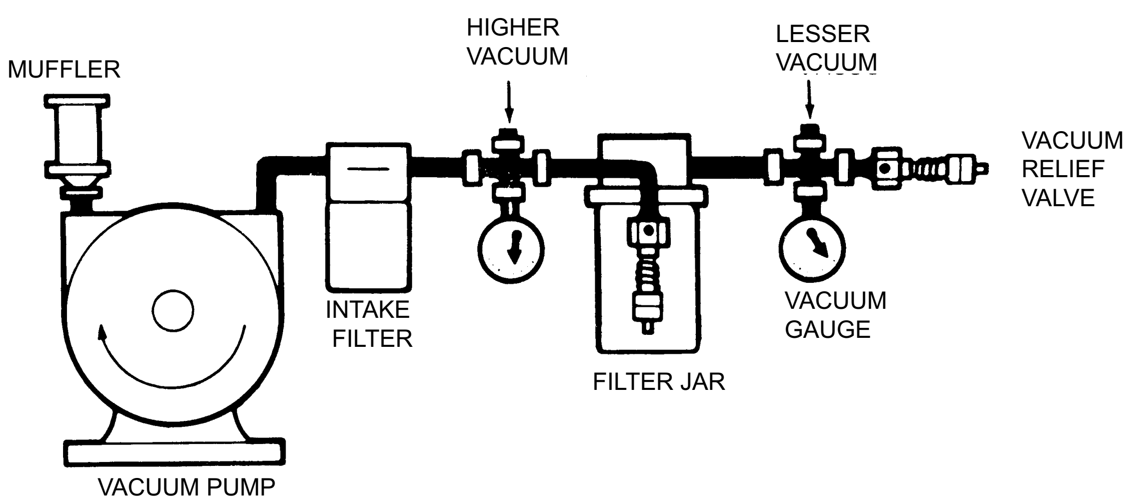


Basic vacuum power supply system for continuous duty applications.

Dual-Level Vacuum System -Fig. 51 shows how the preceding continuous-duty system can be used to provide a second, separately adjustable vacuum level downstream (note the two vacuum gauges).

Two adjustable vacuum relief valves are used. The first one controls the higher vacuum (lower pressure) level. It is mounted in a filter jar to ensure that the modulated air leakage is drawn from the downstream leg of the circuit, instead of from the atmosphere, when the preset vacuum level is exceeded. The second vacuum relief valve is mounted conventionally, drawing its modulated flow of leakage air from the atmosphere when the preset lesser vacuum level is exceeded.

Figure 51



Typical vacuum power supply system designed to maintain two separately adjustable vacuum levels.

The settings of these two valves are cumulative. That is, the setting of the two valves must not exceed the maximum vacuum rating of the pump. For example, a pump with a 20 in. Hg rating could supply such combinations of vacuum levels within the system as l8 and 2 in. Hg, 15 and 5 in. Hg, etc. Let's consider a system in which the gauges are set, respectively, for 12 and 8 in. Hg. When the vacuum pump starts, it evacuates air from the downstream leg of the system until the pressure is 12 in. Hg. At that point the first vacuum relief valve cracks, allowing the pump to draw air from the upstream leg.

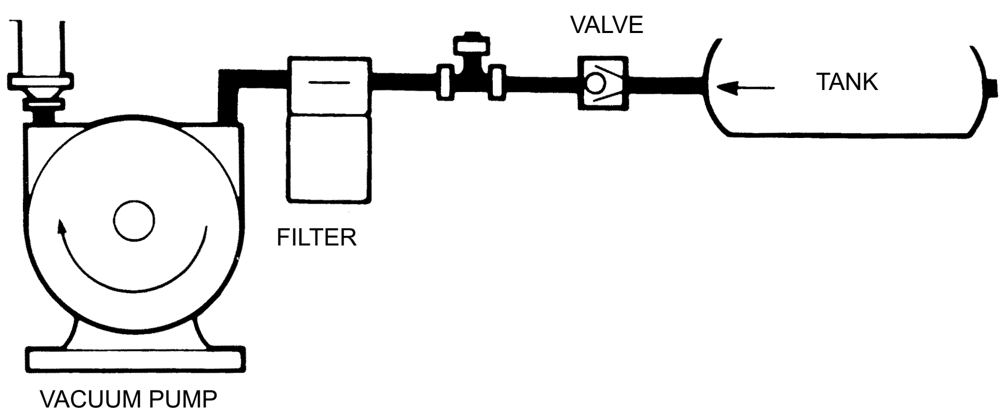
As air is evacuated from the upstream leg of the system, the vacuum relief valve maintains a constant pressure differential of 12 in. Hg between the two sections. This continues until pressure in the upstream leg falls to 8 in. Hg. At that point, the second vacuum relief valve opens, stabilizing the pressure in the upstream leg at 8 in. Hg and that in the downstream leg at 20 in. Hg.

Vacuum Storage System -Fig. 52 shows how a receiver tank can be integrated into the power supply system to accumulate vacuum for on-demand fluid power. For example, such a system could be used for vacuum forming of plastic sheets.

In general, this kind of system is used for mostly on cycle operations, where the pump runs under load continuously. The use of a receiver tank helps smooth pulses in the suction flows of piston and diaphragm pumps, which do not have pulseless delivery characteristics.

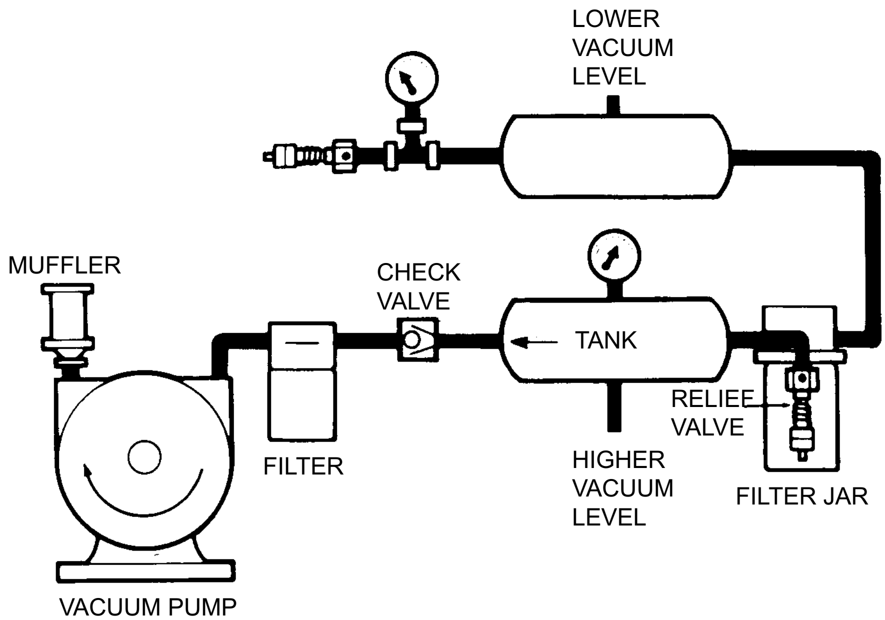
A receiver tank also enables very rapid operation of the actuator, because any air contained in it rapidly expands to fill the evacuated receiver. This action provides a large drop in the absolute pressure within the receiver tank.

Figure 52



Basic power supply system including vacuum storage.

Figure 53



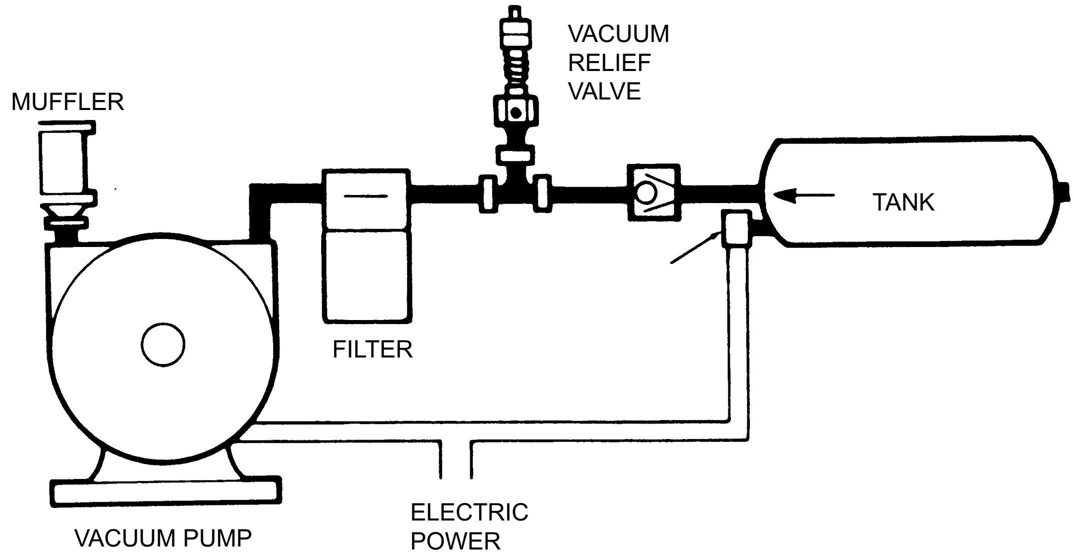
Vacuum power supply system with two receiver tanks to provide separate vacuum levels.

Since rotary vane pumps are valveless, a check valve is necessary when that type of pump is used with a receiver tank. This valve prevents leakage of atmospheric air into the tank when the pump is not operating. But no matter what type of pump is used, the only control device required is an adjustable vacuum relief valve to maintain the required vacuum level.

Dual Level Vacuum Storage System -Fig. 53 shows how two receiver tanks can be hooked up to provide different system vacuum levels on demand. Except for the vacuum storage capability, this power supply is similar to the dual-level system. The only additional component is a check valve installed at the outlet of the first receiver tank.

On/Off Cycling System -If the vacuum pump is driven by an electric motor, the receiver tank vacuum level can be controlled by intermittent on/off cycling. Fig. 54 shows the connection of a vacuum switch at the receiver tank to the vacuum pump drive motor. The system is used mainly for extended off cycles.

In operation, the vacuum switch allows the drive motor to run until it senses that the vacuum level has reached a preset upper cut-out level. At that point, the contacts open and stop the drive motor, shutting down the pump. Then the check valve closes. The switch contacts remain open until the receiver tank's vacuum level drops to a preset lower cut-in level. Then the contacts close to actuate the drive motor, and the pump is automatically turned on again.



Vacuum level in this sytem is controlled by a vacuum switch, which automatically actuates the drive motor.

The usual range between the cut-in and cut-out points is 5 to 15 in. Hg. A vacuum relief valve is always used with this type of system to provide independent protection against excessive vacuum.

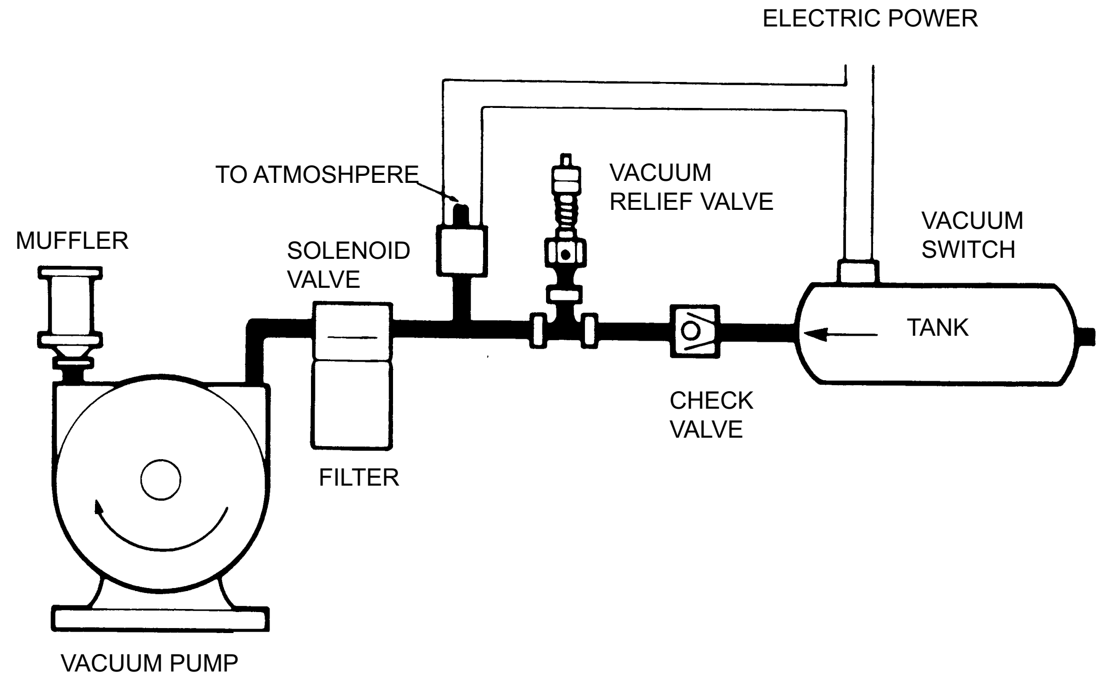
Load/Unload Cycling System -The vacuum switch used in the preceding system can also automatically maintain a desired receiver tank vacuum level in systems driven by gasoline engines or power takeoffs.

As Fig. 55 shows, the switch is connected to a solenoid valve mounted in the line. Normally, the solenoid valve remains energized (vent closed). But when the vacuum switch at the receiver tank senses that the vacuum has reached its upper (unload) limit, its contacts open and de-energize the solenoid, opening its valve. This permits atmospheric air to be drawn in and circulated through the pump chamber.

The solenoid remains de-energized until the pressure switch senses that the vacuum level in the receiver has dropped back to its lower load limit. A signal is transmitted to the solenoid to stop the venting action. But because of the check valve in the line, the opening of the solenoid valve does not immediately affect the vacuum level in the receiver. The vacuum there will be reduced only via normal system operation.

A distinct advantage of load/unload operation is that the air drawn through the chamber in the unloaded condition permits more effective pump cooling than would on/off cycling.

Figure 55



In this system, the vacuum level is controlled by a solenoid switch, which automatically admits atmospheric air.

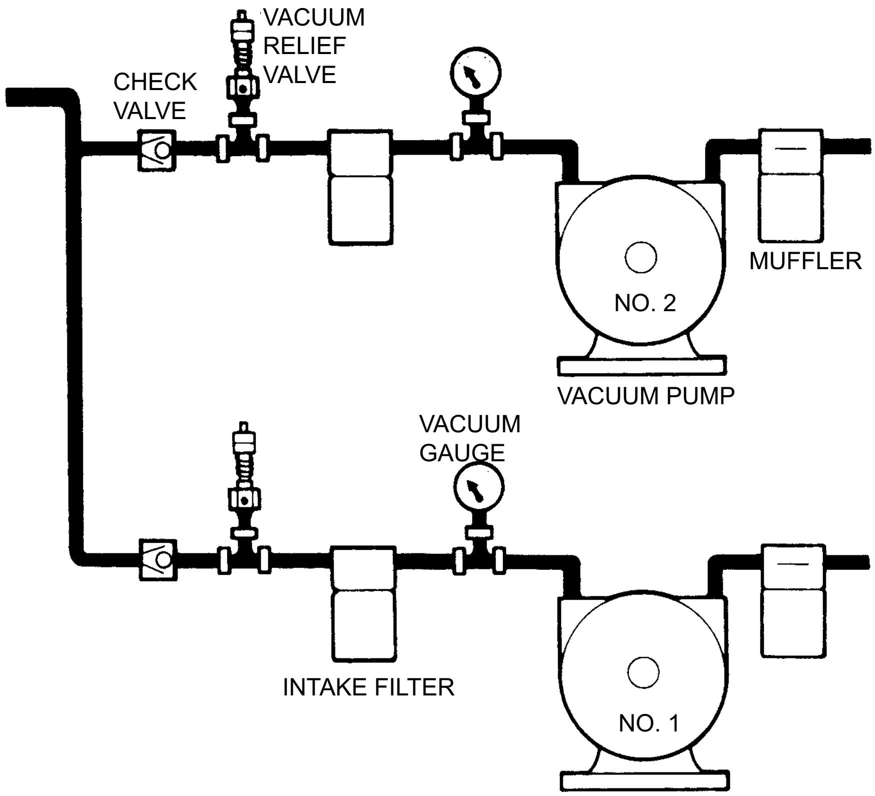
This reduces wear and permits operation at higher than normal vacuum levels.

As shown in Fig. 55, a vacuum relief valve always must be used in load/unload systems to provide independent high vacuum protection.

Two-Pump Vacuum System - Fig. 56 shows how two vacuum power supply systems are sometimes combined to meet larger capacity requirements, or perhaps to allow faster initial pump-down of the receiver tank volume. In general, such an equipment configuration can provide greater flexibility in meeting varying system demands. Often, a redundant operating capability is required in applications where loss of the vacuum pumping function could be harmful or costly.

The two vacuum pumps are normally powered and controlled separately. Although not shown in Fig. 56, the two power supplies should be equipped with separate on/off or load/unload control systems. In some applications, different upper and lower vacuum switch settings are desired to actuate automatic cut-in of the second pump. For example, it may be used only to meet a heavier load demand.

Separate check valves connect the pumps to the rest of the system or to receiver tanks. Along with separate vacuum relief valves, total redundancy is provided so that either machine can be shut down without interrupting the rest of the system.



Two-pump vacuum power supply system can meet special needs.

Combined Compressor/ Vacuum Pump Systems

There are many applications where both pressure and vacuum are required to perform work in the same cycle. One major category is paper-handling equipment such as computers, printing presses, sorting machines, folders, and the like.

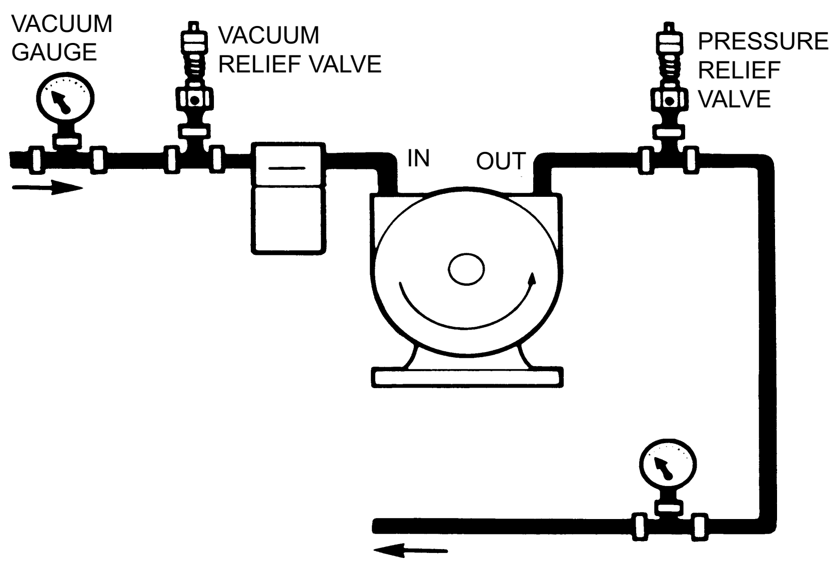
While two separate machines could be teamed up to handle the job, the use of either a dual-chamber or single-chamber compressor/vacuum machine can reduce costs. Some models are designed with double inlet ports, permitting simultaneous pressure/vacuum operation.

Compressor/Vacuum Power Supply System

In general, the vacuum level must be relatively low (about 10 in. Hg maximum) for the pump to receive enough air to compress to positive pressures on the outlet side. A rule of thumb is to limit vacuums to 10 in. Hg and pressures to 3 psi. The limitation usually results from the unit's temperatures.

Fig. 57 shows a power supply designed to provide both pressure and vacuum. The system has some limitations.

Figure 57



Combination compressor/vacuum pump power supply system. Vacuum is routed from the application (top of drawing) and pressure is routed to the application (bottom).

Since the total of the pressure and vacuum values used must not exceed the maximum vacuum rating of the pump, the positive pressure must be relatively low. (Remember to use consistent units. A guide based on unit conversion values is that the vacuum level used must be decreased by 2 in. Hg for every psig of pressure used.) As Fig. 57 shows, both vacuum and pressure levels are controlled by separate relief valves. These also protect the machine against possible damage from excessive vacuum or pressure.

Compressor/Vacuum Pump Selection

Dual-Chamber Unit

The dual-chamber combination utilizes separate compressor and vacuum sections (Fig. 58).

Maximum Pressure/Vacuum Ratings -in the initial phase of equipment selection, the job of determining specific pressure and vacuum requirements should be treated as separate problems.

Pressure -Determine the maximum pressure rating, free air capacity, and possible oil-less exhaust requirements for the compressor section of the machine.

Vacuum -Determine the maximum vacuum rating, open capacity, and possible oil-less exhaust requirements for the vacuum pump section

Figure 58

Typical dual-chamber compressor/vacuum pump.

Table 11 lists some representative dual-chamber rotary vane machines. The oil-lubricated unit has generally higher maximum pressure ratings than the oil-less one.

Drive Power Requirements -The machines listed in Table 11 are separate-drive units, permitting maximum flexibility in accommodating a range of horsepower and speed requirements. Power requirements for the units listed range from 1 to 2 hp, speed requirements from 800 to 1200 rpm.

Single-Chamber Unit

Instead of separate pressure/vacuum sections, a single-chamber machine can meet both requirements, either simultaneously or alternately. The pressure requirement of the air-actuated work device is provided by piping the flow from the discharge port. The vacuum requirements are met by providing a connection to the pump inlet port. Fig. 59 shows a representative machine.

For simultaneous pressure and vacuum requirements, these dual capabilities are basically limited to applications having relatively low levels of pressure and vacuum. For alternate pressure and vacuum requirements, though, the machine can accommodate pressure and vacuum levels up to its maximum rating.

The rotary vane design, with high duty characteristics, is often selected because of its effective exterior cooling.

Figure 59

Typical dual-chamber compressor/vacuum pump.

Maximum Pressure/Vacuum Ratings -Single-chamber machine selection is normally based on relative pressure/vacuum requirements. That is, if the required pressure is high relative to vacuum, then a basic compressor is chosen. If the vacuum requirement is higher than the pressure requirement, then a basic vacuum pump is chosen. The combined duties must not exceed the maximum pressure or vacuum rating. Capacity selection Is based on the largest of the capacities required by the system - either the rate of air consumption or the rate of air removal.

Drive Power Requirements -If the restrictions regarding system pressure/vacuum levels are observed, there should be no problem concerning power or duty cycle requirements. These parameters will be the same as when applying the machine to separate pressure or vacuum systems.

Table 11

Available combined compressor/vacuum systems

Type	Compressor Section		Vacuum Pump Section		
	Rated Capacity (cfm)	Max. Pressure Rating (psig)	Rated Capacity (cfm)	Max. Vacuum Rating (in. Hg)	
				Intermit.	Cont.
Rotary Vane (oil-lubricated)	7 to 14	15	7 to 9	20	15-20
Rotary vane (oil-less)	7.5 to 9.5	10	16 to 19	20	20
Diaphragm	0.4 to 2	60	0.5 to 1.8	26	26
Rocking Piston	.7 to 3.0	100	1.2 to 1.6	27.5	27.5
Piston	1.25 to 1.6	100	1.3 to 1.8	27.5	27.5

Pressure Application

Problem: A pressure source is used to agitate a plating tank of corrosive chemicals. Depth of liquid in the tank is 6 ft. A pipe with forty 1/32-in. diameter holes placed at bottom of tank will discharge air and promote agitation of the fluid. The pump must be oil-less so that no oil contamination will be introduced.

Solution:

1 .Required pressure is determined by depth of fluid in the tank; 1 in. H^20 = 0.0361 psi, so 6 ft H^20 = 6 x 12 X 0.0361 = 2.6 psi. To overcome losses, plan on 3 to 5 psi.
2. Using 5 psi, we find from a compressed air handbook that a 1/32-in. diameter hole will pass 0.25 cfm.
3. Total air usage at 5 psi will be 40 x 0.25 = 10 cfm.
4. From Table 13, a Gast Model 1550 oil-less compressor operating at 1440 rpm will meet the requirement (10 cfm at 5 psi).

Pressure Application

Problem: An environmental air monitoring agency requires a-quiet, portable compressor to pump a mixture of 90 percent air and 10 percent hydrogen sulfide from its container at atmospheric pressure to a 1/32-in. diameter jet at 25 psi minimum. What model should be selected?

Solution:

1 .A handbook shows that a 1/32-in. orifice will pass 0.56 cfm at 25 psi and 0.7 cfm at 35 psi.
2. Since hydrogen sulfide is toxic, a sealed compressor is desirable.
3. The diaphragm pump is a commonly available sealed type.
4. Checking the flow data in Table 14 for Model DOA-101, we find by interpolation that it produces 0.62 cfm minimum at 25 psi and 0.47 cfm at 35 psi.
5. Therefore, Model DOA-101 will meet all flow and pressure requirements.
6. Hydrogen sulfide gas does not affect the diaphragm material.
7. Model DOA-101 is quiet (55 dBA at 25 psi) and is the final choice.

Vacuum Application

Problem: Slabs of marble weighing 250 lb. are to be loaded and unloaded from a flat bed truck at the rate of one per minute. Vacuum will be used to protect the surface of the slabs. Since it will be mounted on the truck, the vacuum lifter is to be powered by a gasoline engine. A manufacturer of vacuum lifting devices is contacted.

Solution:

1. A basic design value for vacuum lifting is 20 in. Hg, which is equivalent to 20 x 0.4913 = 9.83 lb. of lifting force per square inch of suction cup area.
2. Two vacuum cups will be used as a safety feature so that the load can still be carried should one cup become damaged or inoperative.
3. If one cup leaks, then the pump selected must be large enough to compensate for the air leaking into the vacuum holding system.
4. For 9.83 lb. of lifting force per square inch, a vacuum cup must have an area of 250 9.83 = 25.43 sq. in. to hold the load.
5. The diameter of a circle with an area of 25.43 sq. in. is 5.7 in. Since the manu facturer has a standard 6-in. vacuum cup, he decides to use it.
6. The orifice in the center of the vacuum cup is 1/8 in. in diameter permitting an air flow of 4.7 cfm when providing a differential of 20 in. Hg.
7. The pump chosen must be large enough to compensate for a potential leak of this amount and still draw a vacuum of 20 in. Hg on the remaining cup. Table 15 indicates that a separate drive Gast Model 2065 (5.0 cfm at 20 in. Hg) will do the job and can be driven by belt from the gasoline engine.
8. Assuming a tubing system 3/4 in. in diameter and 25 ft. long, a volume of 0.08 cu. ft. must be evacuated in 10 seconds, the time allotted for evacuation in the one-slab-per-minute specification. Model 2065 does the job.

Vacuum Application

Problem: Design a vacuum palletizer for loading and unloading pallets of small boxes 12 by 12 in. that can weigh as much as 80 lb. each. The pallet is 48 in. square, and when the lifter comes down, there can be any combination of boxes on a particular layer. They will always be positioned according to a set pattern.

Cycle time is once every 4 minutes.

Solution:

1. Using 10 in. Hg as the basic design requirement, we get 10 x 0.4913 = 4.91 lb. of lifting force per square inch of suction cup.
2. A vacuum cup with an area of 80/4.91 = 16.3 sq. in. is required to lift each box.
3. A 4.56 in. diameter sucker cup would do the job with no safety factor.
4. For a 50 percent safety factor, 5.6 in. diameter cups would be needed. A standard diameter, such as 6 in., should be selected.
5. The orifice in the sucker cups is 1/16 in., permitting an air flow of 0.74 cfm at 10 in. Hg (for each of the 16 cups).
6. Because we are going to lift from one to 16 cartons, we must select a pump that will accommodate the flow from 15 open cups. Hence, it must pump at least 15 x 0.74= 11.1 cfm at 10 in. Hg.

7. The pump is mounted on the carrier or lifting head so the system contains less than 1 cu. ft.
8. From Table 15, we see that a Gast Model 2565 pump can be used (flow 13.5 cfm at 10 in. Hg).

Combined Vacuum-Pressure Application

Problem: A vacuum forming machine requires both a pressure and vacuum source. Pressure is used to raise the 100-lb. heater plate 12 in. in 10 seconds, using an air cylinder with 3 sq. in. area. Vacuum of 15 in. Hg is used to draw heated plastic sheet down into the mold, which has a volume of 1/2 cu. ft. The mold vacuum level of 15 in. Hg is to be attained within 2 minutes. The pump selected must be compact with no V-belt drives.

Solution:

1. A 50 psi piston type compressor can be operated with one cylinder at pressure and the other at vacuum providing that the sum of the operating pressure in psi and 1/2 the vacuum duty in in. Hg does not exceed 50.
2. Pressure required in the 3 sq. in. cylinder to raise the 100 lb. heater plate is 100/3 = 33 psi. (Use 40 psi, since pressure exceeding 33 psi will be needed to raise the plate.)
3. To move the 3 sq. in. cylinder 12 in. requires a volume of 3 x 12 or 36 cu. in. of air at 40 psi.
4. Converting 36 cu. in. air at 40 psi to free air, use P1V1 = P2V2. (P1 and P2 expressed as absolute pressure, or gauge pressure + 14.7 psi).

$$(40 + 14.7)(36) = (14.7)(V2)$$
$$V_2 = (54.7/14.7)(36) = 134 \text{ cu. in.}$$
$$134 \text{ cu. in.}/1728 = 0.078 \text{ cu. ft.}$$
$$0.078 \text{ cu. ft. in 10 sec.} = 0.078 \times 60/10 = 0.46 \text{ cfm}$$

5. Therefore, for the pressure portion of the problem, we need a source to provide 0.5 cfm at 40 psi.
6. The vacuum portion requires a source that will pump 1/4 to 1/2 cfm at 15 in. Hg. (A pump with a capacity of 1/2 cfm at 15 in. Hg will lower a 1/2 cu. ft. volume from atmospheric pressure to 15 in. Hg in one minute).
7. A pressure of 40 psi requires a piston-type pump. From Table 16 we see that the two-cylinder Gast Model 2LBB would produce from one cylinder 1/2 x 1.50 = 0.75 cfm at 40 psi, exceeding the minimum requirement of 0.5 cfm.
8. The vacuum cylinder will produce about 1/2 the open flow of one cylinder. For Model 2LBB, this is 1/2 of 2.4 cfm, or 1.2 cfm cylinder. Again, this exceeds the requirements of 1/4 to 1/2 cfm at 15 in. Hg.
9. Checking that the effect of combined duties is less than 50, we have 40 psi + 15 in. Hg/2 = 47.5.
10. Final choice is the 2LBB-10-M200X, a motor-mounted, compact piston type unit.

APPENDIX

Nomenclature

V_{vc} = volume of expanded air, cu ft
V = volume of free air, cu ft
P = pressure (psi)
P_a = absolute pressure (psia)
P_1 = inlet (or original) absolute pressure
P_2 = discharge (or final) absolute pressure
V_1 = inlet (or original) volume
V_2 = discharge (or final) volume
T_1 = inlet (or original) absolute temperature
T_2 = discharge (or final) absolute temperature
m = mass
T = absolute temperature (^{o}Rankine or ^{o}Kelvin)
R = 0.08207 lit-atm per mole-^{o}K

Gas Law Units

Pressures Used in Gas Laws

Positive Gauge Pressure is the pressure above atmospheric pressure (measured in psig).

Negative Gauge Pressure (vacuum) is the difference between atmospheric pressure and the pressure remaining in the evacuated system (measured in inches Hg or negative psig).

Absolute Pressure is the pressure above a perfect vacuum condition measured in psia. When using gas laws, pressure must be absolute pressure values.

Metric Units -in metric systems pressures are given in "bars" equal to 14.50 psi. The unit of force is the newton, and the unit of area is the square meter. One bar is 10,000 newtons per square meter.

Temperatures Used in Gas Laws

Absolute Temperature is the temperature above absolute zero, the point where all thermal activity ceases. Such a perfect gas would exert no pressure if kept at a constant volume. In U.S. units, absolute temperature is given in degrees Rankine (^{o}R), and absolute zero is equivalent to - 460^o F.

NOTE: In metric units, absolute zero is - 273^{o}C, and absolute temperatures are given in degrees Kelvin (^{o}K).

Conversions to and from degrees Rankine and degrees Kelvin are made as follows:

^{o}R = ^{o}F + 460
^{o}F = ^{o}R - 460

^{o}K = ^{o}C + 273
^{o}C = ^{o}K - 273

Volume Measurements for Gas Laws

For the General Gas Law, the volume unit must correspond to the value of R used. For example, when the value 53.3 is used, volume must be in cubic feet.

For other laws, the only requirement is that all volumes be given in the same units. For example, receiver tank volumes are sometimes given in gallons. To convert gallons into cubic feet, simply multiply by 0.1337.

Gas Laws

As discussed in section 1, the relationships of pressure, volume, and temperature of a quantity of air are interrelated. The first three laws cover conditions where the quantity or mass of air is constant. The fourth law (General Law) provides for computation involving change in the mass of air.

Boyle's Law

$$P_1V_1 = P_2V_2$$

This basic law covers the relationship between changes in pressure and volume when temperature remains constant.

Variations:

(a) $V_2 = \dfrac{P_1V_1}{P_2}$

(b) $P_2 = \dfrac{P_1V_1}{V_2}$

(c) Free air calculation:

$$V_{FA} = V \times \frac{(P + 14.7 \text{ psi})}{14.7 \text{ psi}}$$

Charles' Law

$$\frac{P_1}{P_2} = \frac{T_1}{T_2} \quad : \quad \frac{V_1}{V_2} = \frac{T_1}{T_2}$$

The basic forms above cover changes in pressure and volume caused by temperature changes. A pressure change is calculated for a system where the volume is constant. A volume change is calculated where the pressure remains constant.

Variations:

(a) $P_2 = \dfrac{P_1 \times T_2}{T_1}$

(b) $V_2 = \dfrac{V_1 \times T_2}{T_1}$

(c) $T_2 = \dfrac{P_2 \times T_1}{P_1}$

(d) $T_2 = \dfrac{V_2 \times T_1}{V_1}$

Combined Gas Law

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

The above basic form combines Charles' and Boyle's Laws to cover variation of all variables.

Variations:

(a) $$P2 = \frac{P_1V_1}{V_2} \times \frac{T_2}{T_1}$$

(b) $$V2 = \frac{P_1V_1}{P_2} \times \frac{T_2}{T_1}$$

(c) $$T2 = \frac{P_2V_2}{P_1V_1} \times T_1$$

General Gas Law or Equation of State of an Ideal Gas

$$mR = \frac{PV}{T}$$

The above basic form includes the effect of mass (in pounds). The right-hand portion of the equation is the same as the Combined Gas Law. R is a constant that varies with the gas being considered and the units used. For air in units of the U.S. system (T in degrees Rankine, V in cubic feet, m in pounds mass, and P in pounds per square foot absolute-to obtain psfa, multiply psia by 144), R has the value 53.3.

Variation:

$$PV = nRT$$

In this variation, n represents the amount of gas in moles. A mole is 6.02 X 1021 molecules; its weight in grams equals the molecular weight of the gas. This makes R independent of the gas involved. In metric units (T in degrees Kelvin, V in liters, and P in atmospheres), R has the value 0.08207.

Terms of either form of the General Gas Law may be manipulated mathematically as required to find the value of any single variable.

Flow Measurements

The volume of air delivered by a compressor or removed by a vacuum pump is given in cubic feet per minute (cfm). This may be either cfm at actual temperature and pressure or standard cubic feet of air per minute (scfm)-that is, cfm at atmospheric pressure and a standard temperature of 68^{o}F (20^{o}C). (For utmost accuracy, scfm also requires correction to a standard humidity of 36 percent.)

The term "free air" is often used interchangeably with "standard air." Strictly speaking, "free air" refers to air at ambient conditions - the conditions at a compressor's intake or a vacuum pump's discharge port. These conditions vary somewhat from day to day.

In the metric system, flow may be given in terms of cubic meters per second, cubic decimeters (liters) per second, or cubic centimeters per second. (One cubic meter per second =35.31 cu ft /sec; one cubic meter per hour= 0.588 cfm).

Force and Mass

In the U.S. system, the unit of force is the pound (lb.). The unit of mass is the slug. In the metric system, the unit of force is the newton (n), which equals 0.225 lb. The unit of mass is the kilogram (kg), equal to 0.06849 slug. At the earth's surface, a kilogram weighs about 2.2 lb.

Power

The units of power are horsepower (hp) and watts (w) or kilowatts (kw). One U.S. horsepower equals 0.746 kw and 1.014 metric horsepower.

Air Pressure Tables

Air pressure losses (in U.S. and metric units) resulting from friction, pipe bends, and in components are given in Tables A-1 through A-6. Tables A-7 and A-8 give air flow through orifices of various sizes.

Loss of Air Pressure (PSI) Because of Friction (per 100 feet of pipe, 100 PSI initial pressure) (U.S. Units)

CFM Free Air (14.7 PSI)	Equivalent CFM Compressed Air (100 + 14.7 (PSI)	Nominal Pipe Diameter, Inside (Inches) Loss of pressure (PSI)					
		1/2	3/4	1	1 1/4	1 1/2	2
10	1.28	1.38	0.09	0.03	0.007		
20	2.56	1.42	0.34	0.10	0.026	0.012	
30	3.84	3.13	0.74	0.23	0.056	0.026	
40	5.13	5.55	1.28	0.38	0.096	0.044	0.013
50	6.41	8.65	2.00	0.60	0.146	0.067	0.020
60	7.69		2.84	0.84	0.210	0.095	0.027
70	8.97		3.85	1.12	0.280	0.130	0.036
80	10.25		5.01	1.44	0.360	0.160	0.046
90	11.53		6.40	1.85	0.450	0.200	0.058
100	12.82		7.80	2.21	0.550	0.250	0.069
125	16.02		12.40	3.41	0.850	0.380	0.107
150	19.22		18.10	4.91	1.200	0.540	0.150
175	22.43			6.80	1.640	0.730	0.200
200	25.63			8.79	2.120	0.950	0.260
250	32.04				3.300	1.480	0.400
300	38.45				4.710	2.100	0.570
350	44.86				6.450	2.860	0.770
400	51.26	`			8.300	3.700	0.990
450	57.67					4.650	1.270
500	64.08					5.790	1.560
600	76.90					8.45	2.230
700	89.71						3.000
800	102.50						4.000
900	115.30						5.050
1,000	128.20						6.200

Table A–2

Loss of Air Pressure (Kg/cm^2) Because of Friction (Per 30 Meters of Pipe and 7 Kg/cm^2 Initial Pressure) (Metric Units)

Liters Free Air Per Min	Equivalent Liters Per Minute Compressed Air	Nominal Pipe Diameter, Inside (Millimeters) Loss of pressure (kg/cm^2) 12.7 mm	19.1 mm	25.4 mm	31.8 mm	38.1 mm	50.8 mm
300	37	0.027	0.006	0.002	0.0005		
600	75	0.100	0.024	0.007	0.0018	0.0008	
850	110	0.220	0.053	0.016	0.0039	0.0018	
1,150	150	0.390	0.090	0.027	0.0067	0.0031	0.0009
1,400	180	0.608	0.140	0.042	0.010	0.0047	0.0014
1,700	218		0.200	0.059	0.015	0.0067	0.0019
2,000	255		0.270	0.079	0.020	0.0091	0.0025
2,300	290		0.350	0.100	0.025	0.0110	0.0032
2,550	330		0.450	0.130	0.032	0.0140	0.0041
3,000	364		0.550	0.155	0.038	0.0160	0.0049
3,500	455		0.870	0.240	0.060	0.0270	0.0075
4,250	540		1.270	0.346	0.065	0.0380	0.0105
5,000	640			0.480	0.115	0.0510	0.0140
5,700	730			0.615	0.150	0.0670	0.0180
7,000	900				0.232	0.1040	0.0280
8,500	1,100				0.332	0.1480	0.0400
10,000	1,300				0.455	0.2000	0.0540
11,000	1,500				0.585	0.2600	0.0700
13,000	1,700					0.3260	0.0890
14,000	1,855					0.4100	0.1090
17,000	2.250					0.5950	0.1560
20,000	2.620						0.2100
23,000	3,000						0.2800
25,000	3,400						0.3550
30,000	3,760						0.4350

Table A–3

Loss of Air Pressure Due to pipe Bends (per 100 Feet of Straight Pipe) (U.S. Units)

Angle of pipe bend	Nominal Pipe Diameter, Inside (Inches) Loss of pressure (PSI) 1/2	3/4	1	1 1/4	1 1/2	2
90^0	1.60	2.00	2.50	3.40	4.00	5.10
45^0	0.73	0.92	1.18	1.55	1.85	2.35

Table A–4

Loss of Air Pressure Due to pipe Bends (per 30 Meters of Straight Pipe) (Metric Units)

	Nominal Pipe Diameter, Inside (Millimeters) Loss of pressure (kg/cm^2)					
Angle of pipe bend	12.7 mm	19.1 mm	25.4 mm	31.8 mm	38.1 mm	50.8 mm
90^0	0.48	0.60	0.75	1.02	1.20	1.53
45^0	0.22	0.28	0.35	0.47	0.56	0.71

Table A–5

Flow of Air Through Orifice, CFM, With Discharge of Orifice at Atmospheric Pressure of 14.7 lb. / in.^{2} Absolute and 70^{0}F (U.S. Units)

SUPPLY PSI (GAUGE)	Orifice size (Inches)									
	1/32	1/16	3/32	1/8	5/32	3/16	7/32	1/4	9/32	5/16
65	1.15	4.49	10.10	17.90	27.90	40.30	55.20	71.80	89.90	111.70
70	1.21	4.77	10.80	19.10	29.70	42.80	58.80	76.40	95.70	118.80
75	1.30	5.06	11.40	20.20	31.50	45.40	62.30	81.00	105.50	126.00
80	1.37	5.35	12.10	21.10	33.30	48.00	65.80	85.60	107.40	133.10
85	1.44	5.64	12.70	22.50	35.10	50.60	69.40	90.30	113.20	140.30
90	1.52	5.92	13.40	23.70	36.90	53.20	72.90	94.80	119.00	147.50
95	1.59	6.21	14.00	24.80	38.70	55.70	76.50	99.40	124.90	154.60
100	1.66	6.50	14.70	26.00	40.50	58.30	80.00	104.60	130.70	161.80
125	2.03	7.94	17.90	31.70	49.50	71.40	97.78	127.10	159.80	197.50
150	2.40	9.28	21.20	37.50	58.40	84.40	115.40	150.40	189.00	233.30

Table A–6

Flow of Air Through Orifice, Liters Per Minute, With Discharge of Orifice Pressure of 14.7 lb. / in.^{2} Absolute and 70^{0}F (U.S. Units)

SUPPLY PSI (GAUGE)	1/32	1/16	3/32	1/8	5/32	3/16	7/32	1/4	9/32	5/16
4.57	32.4	127.2	283.2	507.0	798.0	1,146.0	1,560.0	2,028.0	2,541.0	3,156.0
4.92	34.2	135.6	306.0	540.0	840.0	1,230.0	1,662,0	2,160.0	2,736.0	3,360.0
5.27	36.6	143.4	322.8	571.2	891.0	1,284.0	1,761.0	2,292.0	2,976.0	3,570.0
5.62	39.0	151.8	342.0	597.0	942.0	1,356.0	1,860.0	2,418.0	3,036.0	3,768.0
5.98	40.8	159.6	360.0	636.0	993.0	1,431.0	1,962.0	2,556.0	3,204.0	3,978.0
6.33	43.2	167.4	379.8	678.0	1,044.0	1,518.0	2,061.0	2,676.0	3,372.0	4,170.0
6.68	45.0	176.4	396.0	702.0	1,095.0	1,573.2	2,163.0	2,808.0	3,528.0	4,374.0
7.03	46.8	183.6	416.4	741.0	1,146.0	1,650.0	2,142.0	2,958.0	3,690.0	4,575.0
8.79	57.6	225.0	507.0	897.0	1,401.0	2,016.0	2,760.0	3,600.0	4,518.0	5,580.0
10.55	67.8	262.8	600.00	1,062.0	1,650.0	2,385.0	3,264.0	4,251.0	5,34.0	6,600.0

Conversion Factors

Length

	in.	ft.	yd.	mile	mm.	cm.	m.	km.
1 in.	1	0.0833	0.0278	-	25.40	2.540	0.0254	-
1 ft.	12	1	0.333	-	304.8	30.48	0.3048	-
1 yd.	36	3	1	-	914.4	91.44	0.9144	-
1 mile	-	5280	1760	1	-	-	1609.3	1.609
1 mm	0.0394	0.0033	-	-	1	0.100	0.001	-
1 cm.	0.3937	0.0328	0.0109	-	10	1	0.01	-
1 m.	39.37	3.281	1.094	-	1000	100	1	0.001
1 km.	-	3281	1904	0.6214	-	-	1000	1

Area

	sq. in.	sq. ft.	acre	sq. mile	sq. cm.	sq. m
1 sq. in	1	0.0069	-	-	6.452	-
1 sq. ft	144	1	-	-	929.0	0.0929
1 acre	-	43,560	1	0.0016	-	4047
1 sq. mile	-	-	640	1	-	-
1 sq. cm.	0.1550	-	-	-	1	0.0001
1 sq. m.	1550	10.75	-	-	10,000	1

Volume

	cu. in.	cu. ft.	cu. yd.	cu. cm.	cu. meter	liter	US gal.	Imp. gal
1 cu. In.	1	-	-	16.387	-	0.0164	-	-
1 cu. ft.	1728	1	0.0370	28,317	0.0283	28.32	7.481	6.229
1 cu. yd	46,656	27	1	-	0.7646	764.5	202.0	168.2
1 cu. cm.	0.0610	-	-	1	-	0.0010	-	-
1 cu. m.	61,023	35.31	1.308	1,000,000	1	999.97	264.2	220.0
1 liter	61,025	0.0353	-	1000.028	0.0010	1	0.2642	0.2200
1 US gal.	231	0.1337	-	3785.4	-	3.785	1	0.8327
1 Imperial Gallon	277.4	0.1605	-	4546.1	-	4.546	1.201	1

Weight

	grain	oz.	lb.	ton	g.	kg.	metric ton
1 grain	1	-	-	-	0.0648	-	-
1 oz.	437.5	1	0.0625	-	28.35	0.0283	-
1 lb.	7000	16	1	0.0005	453.6	0.4536	-
1 ton	-	32,000	2000	1	-	907.2	0.9072
1 g.	15.43	0.0353	-	-	1	0.001	-
1 kg.	-	35.27	2.205	-	1000	1	0.001
1 metric ton	-	35,274	2205	1.1023	-	1000	1

Pressure

	lb. / sq. in	lb. / sq. ft	int. atm.	kg / cm^2	mm Hg at 32^0F	in. Hg at 32^0 F
1 lb/sq. in.	1	144	-	0.0703	51.713	2.035
1 lb/sq. ft.	0.00694	1	-	-	0.3591	0.0141
1 international atmosphere	14.696	2116.2	1	0.0333	760	29.92
1 kg / sq. cm.	14.223	2048.1	0.9678	1	735.56	28.95
1 mm. Hg.*	0.0193	2.785	-	-	1	0.039
1 in. Hg.	0.4912	70.73	0.0334	0.0345	25.400	1

*Also known as 1 torr.

Power

	hp	watt	kw	Btu / min.	Btu / hr.	ft-lb / sec.	ft - lb / min.	g. cal / sec.	metric hp
1 hp	1	745.7	0.7475	42.41	2544.5	550	33,000	178.2	1.014
1 watt	-	1	0.001	0.0569	3.413	0.7376	44.25	0.2390	0.00136
1 kw	1.3410	1000	1	56.88	3412.8	737.6	44.254	239.0	1.360
1 Btu/min.	-	-	-	1	60	12.97	778.2	4.203	0.0239
1 ,metric hp	0.9863	735.5	0.7355	41.83	2509.8	542.5	32.550	175.7	1

GLOSSARY

Absolute Pressure
In pressure or vacuum systems, absolute pressure is the pressure above a perfect vacuum condition (zero pressure). In a pressure system, it is equal to the positive gauge pressure plus atmospheric pressure. In a vacuum system, it is equal to the negative gauge pressure subtracted from atmospheric pressure. U.S. units for absolute pressure are pounds per square inch, absolute (psia).

Actuator, Linear
Converts fluid energy into linear mechanical force and motion. Usually consists of a movable element, such as piston and piston rod operating within a close-fitting cylindrical bore.

Actuator, Rotary
Converts fluid energy input to mechanical output. Rotates an output shaft through a fixed arc to produce oscillating power.

Adiabatic
A change, such as expansion or compression, without loss or gain of heat. Any sufficiently fast process is approximately adiabatic.

Air Compressor
Device that causes a gas to flow against a pressure; converts mechanical force and motion into pneumatic fluid power. An air pump.

Air Motor
A device that converts the flow of pressurized air into continuous rotary motion or torque.

Atmosphere
Unit of pressure that will support a column of mercury 29.92 inches high at O^{o}C, sea-level, and latitude 45^{o}. Actual day-to-day atmospheric pressure fluctuates about this value.

Atmospheric Pressure
Pressure exerted by the atmosphere in all directions, equal at sea level to about 14.7 psi.

Back Pressure
Resistance to flow in a system.

Barometer
Device for measuring atmospheric pressure at a specific location.

Barometric Pressure
The reading, in inches of mercury (in. Hg), showing atmospheric pressure at a given location.

Brake Horsepower
The actual or useful horsepower of an engine, usually determined from the force exerted on a dynamometer connected to the drive shaft.

Boyle's Law
The absolute pressure of a fixed mass of gas varies inversely as the volume, provided the temperature remains constant. $P_1V_1 = P_2V_2$

Charles' Law
The volume of a given mass of gas is directly proportional to its absolute temperature, provided the pressure on the gas is held constant. $V_1/T_1 = V_2/T_2$- (Also, $P_1/T_1 = P_2/T_2$ at constant volume.)

Check Valve
In simplest form, a two-way directional valve. It permits free flow in one direction and blocks flow in the reverse direction. Can function as either a directional or pressure control device.

Clearance Volume
The space between a piston and cylinder head at full compression.

Compressed Air
Air that has been reduced in volume and exerts a gauge pressure.

Dessicant Dryer
An absorption material that removes moisture from air.

Differential Pressure
Difference in pressure between two points in a system or component.

Differential Pressure Switch
Switch with a low pressure and high pressure adjustment; fluid pressure actuates an electric switch to perform functions.

Displacement
The total volume swept by the repetitive motion of the pumping element. Displacement per revolution is determined by size of the pumping chamber(s). Displacement per minute also depends on compressor speed. Displacement is meaningful only in positive displacement compressors.

Efficiency, Volumetric
Ratio of actual capacity to theoretical displacement multiplied by 100 percent.

Filter, Air Intake
Device whose primary function is the retention by a porous medium of insoluble contaminants from a fluid. Installed at intake port of compressor or vacuum pump.

Fluid Mechanics
The grouping of hydrodynamics (mechanics of liquids) and pneumatics (mechanics of gases). A fluid is any gas or liquid, the shape of which yields to pressure.

Fluid Power
Energy transmitted and controlled through use of a pressurized fluid within an enclosed circuit.

Fluid Power Horsepower
Power that is proportional to the product of gauge pressure (or vacuum) and air flow rate.

Flow Rate
The quantity of fluid passing a point per unit of time. In pneumatics, this is commonly represented by cfm (cubic feet per minute).

Force
That which can impose a change of velocity on a material body. It is a vector quantity indicating magnitude and direction. Force is theoretically proportional to the mass of the body acted on and the acceleration produced.

Free Air
Air under the atmospheric conditions (including temperature) at any specific location.

FRL
Filter/Regulator/Lubricator. Used in pneumatic systems, this combination filters particles and separates moisture from incoming air; regulates the supply pressure; and lubricates the air by adding oil vapor.

Gauge Pressure (Positive)
The pressure differential above atmospheric pressure (see pressure gauge).

Gauge Pressure (Negative)
The difference between pressure remaining in an evacuated system and atmospheric pressure (see vacuum gauge). Also known as -gauge vacuum" or "vacuum level." In effect, it is the pressure drop produced by evacuating the system. Measured in inches of mercury (in. Hg). Caution: It is a potentially misleading term which must be carefully defined when used; negative pressure (absolute) doesn't exist.

General Gas Law
Obtained by combining Boyle's Law and Charles' Law, and is used to compute change in volume, pressure, and temperature of a gas. Essentially, it states that the product of pressure and volume, divided by the temperature, remains constant.

Head
Energy per pound caused by pressure, velocity, or elevation. The height of a column or body of fluid above a given point expressed in linear units. Often used to indicate gauge pressure. Pressure is equal to the height multiplied by the density of the fluid.

Hydrostatic Pressure
The pressure exerted equally in all directions at points within a confined fluid (liquid or gas). It is the only stress possible in a fluid at rest.

Isothermal
Compression or expansion of a gas taking place at constant temperature. In practice, this is a slow process because of the time required to remove heat generated by compression or to replace heat absorbed by expansion.

Kinetic Energy
The energy a body possesses by virtue of its motion. Kinetic energy is added to a fluid either by rotating it at high speed or by providing an impulse in a direction of flow.

Lubricator
Pneumatic component that lubricates by injecting atomized oil into the air stream.

Manometer
A differential pressure gauge in which pressure is indicated by height of a liquid column of known density. Pressure is equal to the difference in vertical height between two connected columns multiplied by the density of the manometer liquid.

Maximum Pressure Rating
Highest pressure level recommended for a compressor.

Maximum Vacuum Rating
Highest vacuum level recommended for a vacuum pump.

Muffler (Exhaust)
A low-restriction flow-through device that reduces air line noise. It also traps moisture (and oil, in lubricated systems).

Nonpositive Displacement
(Of compressor or vacuum pump). One that uses kinetic energy to create pressure gradients (slopes) for moving air.

Open Capacity
The volume of air exhausted per minute when there is no vacuum or pressure load on the pump, expressed in cfm.

PSIA
Pounds per square inch absolute-pressure measured from a state with a total absence of air (see absolute pressure).

PSIG
Pounds per square inch gauge-pressure above or below (vacuum) atmospheric pressure.

Pascal's Law
A pressure applied to a confined fluid at rest is transmitted with equal intensity throughout the fluid at right angles to containing surfaces.

Pneumatic Fluid Power
The energy transmitted and controlled through use of a pressurized fluid within an enclosed circuit.

Positive Displacement
(Of a compressor or vacuum pump.) One that moves a specific volume of air for each cycle of operation.

Power
The rate of which work is being done. Simply expressed by the
formula: Power= Work/Time

Power Takeoff
Any rotating shaft for driving other machines.

Pressure
Force per unit area acting on a surface, usually expressed inpounds per square inch (psi) or in megaPascals (MPa).

Pressure Differential
Difference in pressure between two points in a system or component.

Pressure Drop
Any reduction in pressure from normal value.

Pressure Gauge
A device that displays the pressure level in a system. Most gauges use atmospheric pressure as a reference level and measure the difference between the actual pressure and atmospheric pressure; the readout is called "gauge pressure." (A gauge that reads below atmospheric is called a vacuum gauge).

Pressure Relief Valve
A valve that provides modulated venting of excess pressure, instead of building up abnormal pressures in the system.

Pressure Regulator
A valve that provides modulated venting of excess pressure, instead of building up abnormal pressures in the system.

Pressure Switch
An electroical switch operated by fluid pressure.

Quick Exhaust Valve
A valve that releases air directly to the atmosphere, bypassing the directional valve. This reduces backpressure resistance.

Rated Capacity (Pressure)
The cfm of free air delivered by a compressor at rated speed. Usually given for pressures ranging from 0 psig to the maximum pressure rating.

Rated Capacity (Vacuum)
The cfm of free air exhausted by a vacuum pump at rated speed. Usually given for vacuums ranging from 0 in. Hg to the maximum vacuum rating.

Rated Pressure
The qualified operating pressure recommended for a component or system by the manufacturer.

Receiver Tank
Container in which gas is stored under pressure or vacuum as a source of pneumatic fluid power. Accommodates sudden or unusually high system demands. Prevents frequent on/off cycling of an air compressor or vacuum pump and absorbs pulsations.

Regulator
Device to control flow of gases, thus controlling the magnitude of the force and torque produced by the actuator.

Safety Valve
A valve that opens to its full capacity to provide a rapid and large reduction in pressure a when predetermined value is exceeded.

Solenoid Valve
A valve operated by an electromagnetic drive.

Standard Air
Air at a temperature of 68^0F, a pressure of 14.70 psia, and a relative humidity of 36 percent.

Surge Point
In centrifugal blowers, an unstable condition occurring at about 50 percent of rated flow when backpressure temporarily exceeds the pressure ratio developed by the compressor.

Vacuum
A space containing air or other gas at less than atmospheric pressure; usually expressed in inches of mercury (in. Hg).

Vacuum Gauge
Device for determining the pressure level in a partial vacuum.

Vacuum Pump
A device that pulls air out of a closed container or system.

Vacuum Relief Valve
A valve that controls system vacuum level. It operates by providing a modulated flow of atmospheric air into the system.

Valve
Device that controls fluid flow direction, pressure, or flow rate.

Volumetric Efficiency
The ratio of a pump's actual delivery to its computed fluid delivery multiplied by 100 percent.

A Worldwide Supplier Of Quality Air Products

Air Compressors - Vacuum Pumps - Vacuum Generators
Air Motors - Geared Air Motors - Regenerative Blowers - Accessories

Corporate Headquarters
Gast Manufacturing, Inc
A Unit of IDEX Corporation
PO BOX 97
2300 M-139
Benton Harbor, MI 49023-0097
Ph: 616/926-6171
Fax: 616/927-0860
http://www.gastmfg-com/
marketing@gastmfg.com

Customer Assistance/Sales
Gast Manufacturing, Inc
2550 Meadowbrook Rd
Benton Harbor, MI 49022
Ph: 616/926-6171
Fax: 616/925-8288

Gast Manufacturing, Inc
Eastern Sales Office
505 Washington Ave
Carlstadt, NJ 07072
Ph: 201/933-8484
Fax: 201/933-5545

Gast Manufacturing, Inc
Midwest Sales Office
755 North Edgewood
Wood Dale, IL 60191
Ph: 630/860-7477
Fax: 630/860-1748

European Sales & Service Headquarters
Gast Manufacturing Co Ltd
Beech House
Knaves Beech Business Centre
Loudwater, High Wycombe
Bucks, England HP10 9SD
Teli Int + 44 (0) 1628 532600
Fax: Int + 44 (0) 1628 532470

Asian Sales Office
Gast Hong Kong
Unit 12, 21/F, Block B
New Trade Plaza
6 , On Ping Street, Shatin
N.T. Hong Kong
Ph :(852) 2690 1008
Fax: (852) 2690 1012