2

Compressed Air Production (Compressors)

OIL INJECTED ROTARY SCREW COMPRESSORS

The oil injected rotary screw compressor, with electric motor driver, has become a dominant type for a wide variety of industrial and mining applications. The engine driven portable version also enjoys great popularity for mining, construction and energy exploration applications. Natural gas engines also are used.

The stationary version is characterized by low vibration, requiring only a simple load bearing foundation and providing long life with minimal maintenance in broad ranges of capacity and pressure. They also are used widely in vacuum service.

The oil injected rotary screw compressor is a positive displacement type, which means that a given quantity of air or gas is trapped in a compression chamber and the space that it occupies is mechanically reduced, causing a corresponding rise in pressure prior to discharge.

Compression Principle

The oil injected rotary screw compressor consists of two intermeshing rotors in a stator housing having an inlet port at one end and a discharge port at the other. The ***male*** rotor has lobes formed helically along its length while the ***female*** rotor has corresponding helical grooves or flutes. The number of helical lobes and grooves may vary in otherwise similar designs.

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Air flowing in through the ***inlet port*** fills the spaces between the lobes on each rotor. Rotation then causes the air to be trapped between the lobes and the stator as the inter-lobe spaces pass beyond the inlet port. As rotation continues, a lobe on one rotor rolls into a groove on the other rotor and the point of intermeshing moves progressively along the axial length of the rotors, reducing the space occupied by the air, resulting in increased pressure. Compression continues until the inter-lobe spaces are exposed to the ***discharge port*** when the compressed air is discharged. This cycle is illustrated in Figure 2.1.

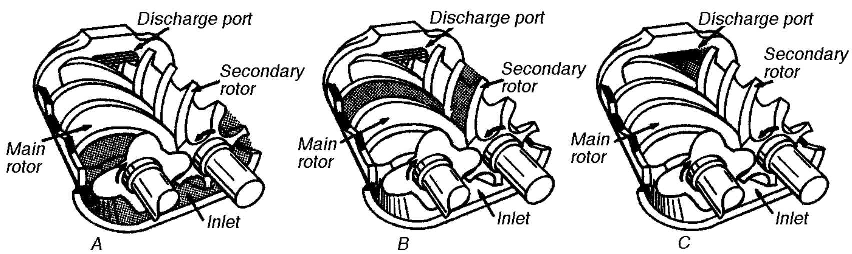

Figure 2.1 Helical Screw-Type Compressor; Compression Cycle, Single Stage

Oil is injected into the compression chamber during compression and serves three basic functions:

1) It ***lubricates*** the intermeshing rotors and associated bearings.

2) It ***takes away*** most of the ***heat*** caused by compression.

3) It ***acts as a seal*** in the clearances between the meshing rotors and between rotors and stator.

Lubrication

The generic term ***oil*** may be a hydrocarbon product but most compressors now use cleaner and longer-life synthetic lubricants, including diesters, polyglycols, polyalphaolefins, polyol esters and silicon based fluids. These newer products are suitable for a wider range of temperatures and have higher flash points. The lubricant chosen should be compatible with the compressor gaskets and seals.

A mixture of compressed air and injected oil leaves the air end and is passed to a sump/separator where most of the oil is removed from the compressed air. Directional and velocity changes are used to separate most of the liquid. The remaining aerosols in the compressed air then are separated by means of a coalescing filter, resulting in only a few parts per million of oil carry-over (usually in the range 2 - 5). A minimum pressure device, often combined with a discharge check valve, prevents excessive velocities through the separator element until a normal system pressure is achieved at start-up. Most oil injected rotary screw compressor packages use the air pressure in the oil sump/separator, after the discharge of the air end, to circulate the oil through a filter and cooler prior to re-injection to the compression chamber. Some designs may use an oil pump.

Bearings at each end of each rotor are designed to carry the radial and axial thrust loads generated. See Figure 2.2A. These bearings are lubricated directly with the same filtered oil as is injected into the compression chamber. A similar arrangement with built-in Spiral or Turn Valve for capacity control, is shown in Figure 2.2B.

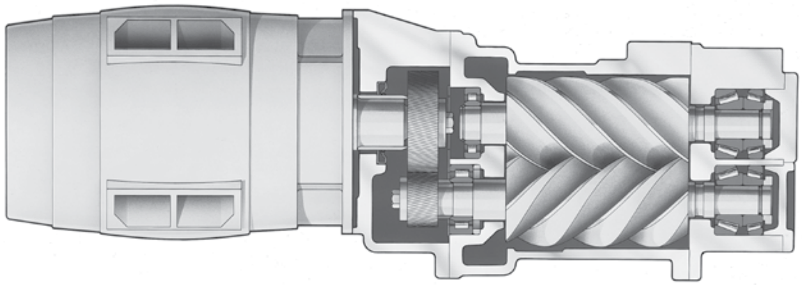

Figure 2.2A Single Stage Oil Injected Screw Compressor

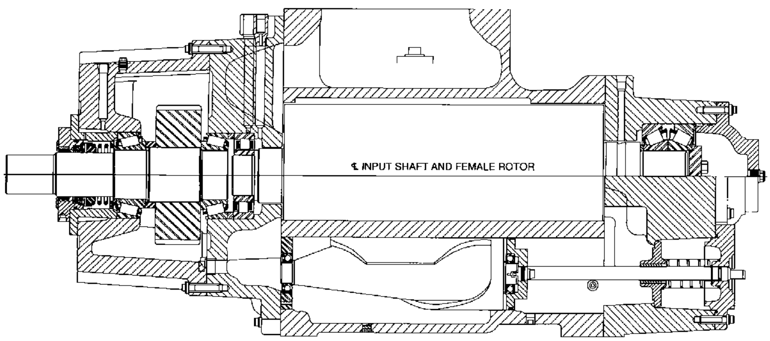

Figure 2.2B Single Stage Oil Injected Screw Compressor with Integral Variable Displacement Control Valve

Multi-stage compressors may have the individual stages mounted side by side, either in separate stators or within a common multi-bore stator housing. See Figure 2.2C. Alternatively, the stages may be mounted in tandem with the second stage driven directly from the rear of the first stage. See Figure 2.2D. Multiple stages are used either for improved efficiency at a given pressure or to achieve higher pressures.

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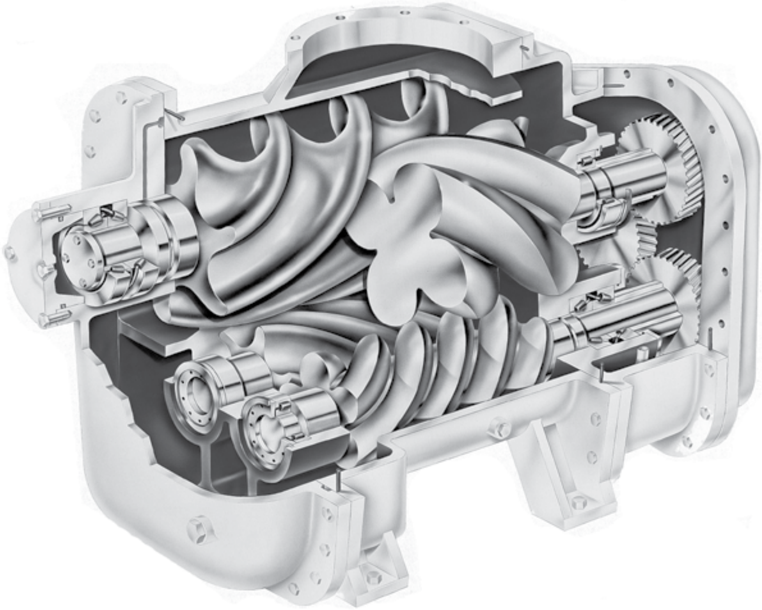

Figure 2.2C Two Stage "Over/Under" Design of Oil Injected Rotary Screw Compressor

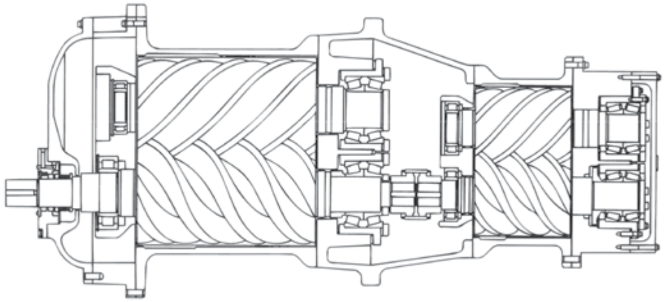

Figure 2.2D Two Stage "Tandem" Oil Injected Rotary Screw Compressor

Cooling

The temperature of the oil injected into the compression chamber generally is controlled to somewhere in the area of 140˚F, either directly or indirectly by controlling the discharge temperature. The discharge temperature must remain above the pressure dewpoint to avoid condensation of moisture that would mix with the oil. A thermostatic bypass valve allows some or all of the oil being circulated to bypass the oil cooler to maintain the desired temperature over a wide range of ambient temperatures.

Generally, a suitable temperature and viscosity of the oil are required for proper lubrication, sealing, and to avoid condensation in the oil sump. It also is necessary to avoid excessive temperatures that could result in breakdown of the oil and reduced life.

In addition to oil cooling, an air aftercooler is used to cool the discharged air and to remove excess moisture. In the majority of applications, radiator type oil and air coolers are employed and provide the opportunity of heat recovery from the compression process for facility heating (Figure 2.3). Water cooled heat exchangers, with water control valves, also are available on most rotary screw compressor packages.

Figure 2.3 Installation of oil injected rotary screw compressors with heat recovery ducting.

In multi-stage designs, oil may be removed and the air cooled between the stages in an intercooler, or the air/oil mixture may pass through a curtain of oil as it enters the next stage.

Single stage oil injected rotary screw compressor packages are available from 3 - 700 hp, or 8 - 4000 cubic feet per minute, with discharge pressures from 50 - 250 psig. Two stage versions can improve specific power by approximately 12 - 15% and some can achieve discharge pressures up to 250 psig. Oil injected rotary screw vacuum pumps also are available from 80 - 3000 inlet cfm and vacuum to 29.7 in. Hg.

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Capacity Control

Demand for compressed air seldom matches exactly the output from the compressor so some form of capacity control is essential. The type of capacity control is based on both the type and size of rotary air compressors, the application, and the number of compressors in the system. (See also Chapter 4, Compressed Air Distribution Systems). Typical capacity control systems for stationary air compressors are as follows:

Start/Stop Control is the simplest form of control, in which a pressure switch, sensing system pressure at the discharge of the compressor, sends a signal to the main motor starter to stop the compressor when a pre-set pressure is reached. When pressure falls to another pre-set pressure, the pressure switch sends a signal for the compressor to be restarted. The pressure switch will have an adjustable upper pressure setting and a fixed or adjustable differential between the upper and lower pressure settings. An air receiver is essential to prevent too frequent starting and stopping, which affects life of motor insulation due to high inrush current at each start. This type of control normally is limited to compressors in the 30 hp and under range. Its advantage is that power is used only while the compressor is running but this is offset by having to compress to a higher receiver pressure to allow air to be drawn from the receiver while the compressor is stopped.

Constant Speed or Continuous Run Control allows the compressor to continue to run, even when there is reduced or no demand for compressed air. This term may be used with **Load/Unload Control** and/or with **Inlet Valve Modulation.**

Load/Unload Control. In this type of control when the upper pressure setting is reached, the pressure switch sends a signal to close a valve at the inlet of the compressor (but maintaining a calibrated low flow), reducing the mass flow through the compressor. Simultaneously a blow-down valve, installed in a line coming from the compressor discharge but prior to a discharge check valve, is opened. When the blow-down valve is opened, the compressor air end discharge pressure is lowered gradually and the discharge check valve prevents back flow from the system or receiver.

Closing the inlet valve reduces the inlet pressure, increasing the pressure ratio across the air end. This reduces as the air end discharge pressure is reduced, resulting in reduced power requirements. In the case of oil injected rotary compressors, the rate of blow-down must be limited to prevent foaming of the oil in the sump/separator. An adequate receiver/system volume is required to allow fully unloaded operation for a sufficient period of time. It may take from 30 to 100 seconds for the sump pressure to be fully reduced, during which time the compressor bhp will reduce from 70% to about 20 - 25%. The average power consumption for a given reduced flow rate will be reduced as receiver/system volume is increased.

In oil free rotary screw compressors and oil free lobe type rotary compressors, this is the most common type of control, with both stages being unloaded simultaneously but without the blow-down time associated with oil injected types, allowing fully unloaded power almost immediately.

Inlet Valve Modulation allows compressor capacity to be adjusted to match demand. A regulating valve senses system or discharge pressure over a prescribed range (usually about 10 psi) and sends a proportional pressure to operate the inlet valve. Closing (or throttling) the inlet valve causes a pressure drop across it, reducing the inlet pressure at the compressor and, hence, the mass flow of air. Since the pressure at the compressor inlet is reduced while discharge pressure is rising slightly, the compression ratios are increased so that energy savings are somewhat limited. Inlet valve modulation normally is limited to the range from 100% to about 40% of rated capacity, at which point the discharge pressure will have reached full load pressure plus about 10 psi and it is assumed that demand is insufficient to require continued air discharge to the system. At this point the compressor will be unloaded as previously described in Load/Unload Control.

Dual Control is a term used to describe a rotary air compressor with a selector switch to enable selection of either **Modulation** (in some cases **Start/Stop**) or **Load/Unload** capacity control. This arrangement is suitable for locations where different shifts have substantially different compressed air requirements.

Automatic Dual or Auto Dual Control is a further refinement to each of the above systems. When a compressor is unloaded a timer is started. If compressed air demand does not lower system pressure to the point where the compressor is required to be re-loaded before the pre-set time has expired, the compressor is stopped. The compressor will re-start automatically when system pressure falls to the predetermined setting.

Variable Displacement (Slide, Spiral or Turn Valve) is a device built into the compressor casing to control output to match demand. Rising discharge pressure causes the valve to be repositioned progressively. This reduces the effective length of the rotors by allowing some bypass at inlet and delaying the start of compression. The inlet pressure and compression ratio remain constant so part load power requirements are substantially less than for inlet valve modulation. The normal capacity range is from 100% to 40-50%, below which inlet valve modulation may be used down to 20-40%, after which the compressor is unloaded. Curve shows Average Power v Percent Capacity with Variable Displacement Capacity Control (Slide/Spiral/Turn Valve) from 100% to 50% capacity followed by Inlet Valve Modulation to 40% capacity, then unloading. With this type of control, the inlet pressure to the air end does not change, hence, the pressure ratio remains essentially constant. The effective length of the rotors is reduced.

Geometric Lift or Poppet Valves may be used to have a similar effect to Slide, Spiral or Turn Valves but with discreet steps of percent capacity rather than infinitely variable positioning. The normal range is from 100% down to 50% capacity and normally with four valves. An inlet modulation valve may be added for capacities below 50%.

Variable Speed may be achieved by variable frequency AC drive, or by switched reluctance DC drive. Each of these has its specific electrical characteristics, including inverter and other losses. Air end displacement is directly proportional to rotor speed but air end efficiency depends upon male rotor tip speed. Most variable speed drive (VSD) package designs involve full capacity operation above the optimum rotor tip speed, at reduced air end efficiency and increased input power, when compared with a constant speed compressor of the same capacity, operating at or near its optimum rotor tip speed. While energy savings can be realized at all reduced capacities, the best energy savings are realized in applications where the running hours are long, with a high proportion in the mid to low capacity range.

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

Field conversion of an existing compressor to variable speed drive must consider the electric motor, the proposed male rotor tip speed at 100% capacity and the reduction of air end efficiency at reduced speeds and capacity. (Figure 2.4).

Steam turbines and engines also can be used as variable speed drivers.

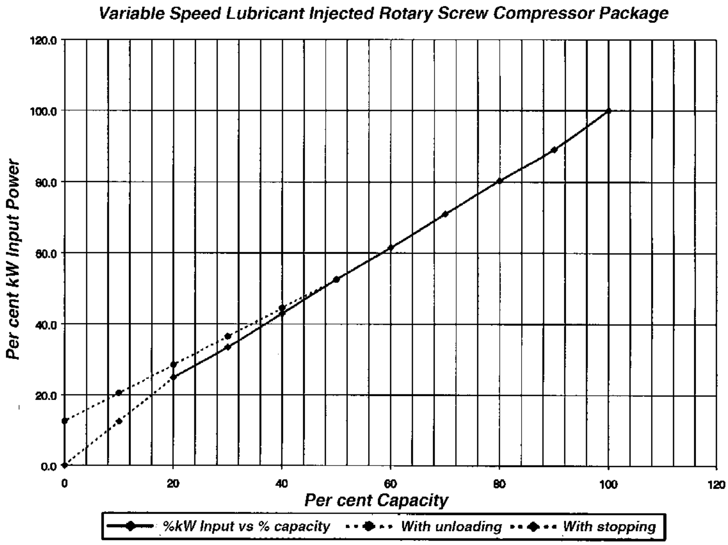

Figure 2.4 Variable Speed Curve

Multiple Compressor Sequencing is desirable in larger installations so that only a sufficient number of compressors will be in operation to meet current demand. There are two types of sequencing systems available:

1. **Cascading System:** In this system, the pressure settings in the compressors installed are overlapped so that as the pressure increases or decreases, the appropriate compressor is started or stopped, loaded or unloaded. This requires a large overall system pressure band.

2. **Rate of Change or Target Pressure System:** In this system, the individual pressure settings in each compressor are over-ridden by the master sequencer. This measures the change in pressure over time, and calculates the system flow requirement. As a result the overall system pressure band is reduced with resulting energy savings over the cascading system.

Compressors are started and stopped, loaded and unloaded, as required to maintain current system requirements. It is desirable to have only one compressor in the system at any given time to be in a reduced capacity mode of operation. This optimizes energy requirements. Sequencing can be arranged to equalize running hours of each compressor or to operate the compressors in a specified sequence, particularly where there is a mix of larger and smaller compressors. The sequence can be changed manually or automatically. Most modern compressors have microprocessor controls, which facilitate appropriate programming. Some microprocessor controls do not require a master sequencer in this type of system.

Safety Systems

In addition to pressure and temperature indicators, the stationary rotary screw compressor package incorporates a pressure relief valve for relief of excess pressure in the air/oil sump/separator vessel. An automatic, blow-down valve relieves pressure from the oil sump/separator vessel on each shutdown. Also included is a high discharge temperature shutdown and, in some cases, a high discharge pressure shutdown. Other safety/maintenance devices typically provided include high air inlet filter differential pressure, high oil filter differential pressure, high air/oil separator differential pressure, low unloaded sump pressure and motor overloads. Most compressor packages now incorporate microprocessors for controls and safety devices.

PORTABLE OIL INJECTED ROTARY SCREW COMPRESSORS

The basic design is similar to that of a stationary compressor but employing an engine driver. This also changes the type of capacity control required and the safety features. Normally, a pneumatic control valve, sensing compressor discharge pressure, is used to progressively close the inlet modulating valve as system pressure rises with decreased air demand. Simultaneously, engine speed, and hence compressor speed, is decreased.

As air demand increases, the system pressure falls, the engine speeds up and the inlet valve opens. Usually, the pressure range from fully closed inlet and idling engine to fully open inlet valve and engine at full speed is about 25 psig or, e.g., an operating pressure range of 100 to 125 psig.

Portable oil injected rotary screw compressors are available in a similar range of capacity and discharge pressure as for the stationary industrial type.

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As for stationary packages, a safety relief valve is incorporated to relieve excess pressure at the compressor discharge. An automatic blow-down valve, actuated and held closed by engine oil pressure, relieves pressure in the oil sump/separator vessel when the engine is shut down. The engine fuel supply also is cut off, stopping the engine in the event of high compressor discharge temperature. The engine itself is protected from low oil pressure, high cooling water temperature and low coolant level by being shut down if either of these malfunctions occurs.

OIL INJECTED SLIDING VANE COMPRESSORS

The oil injected sliding vane rotary compressor was introduced as an engine driven portable air compressor in 1950 and a few years later was applied as a stationary industrial air compressor with electric motor driver.

The oil injected rotary sliding vane compressor is a positive displacement type, which means that a given quantity of air or gas is trapped in a compression chamber and the space that it occupies is mechanically reduced, causing a corresponding pressure rise prior to discharge.

The basic design consists of a circular stator in which is housed a cylindrical rotor, smaller than the stator bore and supported eccentrically in it. The rotor has radial (sometimes off-set) slots in which vanes, or blades, slide. Rotation of the rotor exerts centrifugal force on the vanes, causing them to slide out to contact the bore of the stator, forming "cells" bounded by the rotor, adjacent vanes and the stator bore. Some designs have means of restraining the vanes so that a minimal clearance is maintained between the vanes and the stator bore.

An inlet port is positioned to allow air to flow into each cell exposed to the port, filling each cell by the time it reaches its maximum volume. After passing the inlet port, the size of the cell is reduced as rotation continues, as each vane is pushed back into its slot in the rotor. Compression continues until the discharge port is reached, when the compressed air is discharged. See Figure 2.5.

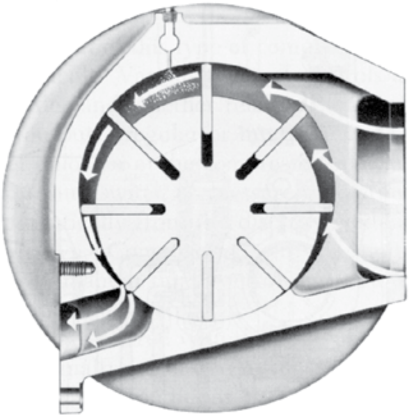

Figure 2.5 An Oil-Injected, Sliding Vane Rotary Compressor

Similar to the oil injected rotary screw compressor, oil is injected into the compression chamber to act as a lubricant, as a seal, and to remove the heat of compression. Single and two-stage versions are available with either in-line or over-under arrangement of the stages.

The oil injected sliding vane compressor normally is sold as a completely pre-engineered package in the range from 10 to 200 hp, with capacities from 40 to 800 acfm and discharge pressures from 80 to 125 psig.

Packaging, oil injection and separation, lubrication, cooling, capacity control and safety features essentially are similar to those for the oil-injected rotary screw compressor, either stationary or portable.

OIL FREE ROTARY SCREW COMPRESSORS

The oil free rotary screw compressor also is a positive displacement type of compressor. The principle of compression is similar to that of the oil injected rotary screw compressor but without oil being introduced into the compression chamber. Two distinct types are available - the ***dry type*** and the ***water injected type.***

In the ***dry type***, the intermeshing rotors are not allowed to touch and their relative positions are maintained by means of lubricated timing gears external to the compression chamber. See Figure 2.6. Since there is no injected fluid to remove the heat of compression, most designs use two stages of compression with an intercooler between the stages and an aftercooler after the second stage. The lack of a sealing fluid also requires higher rotative speeds than for the oil injected type.

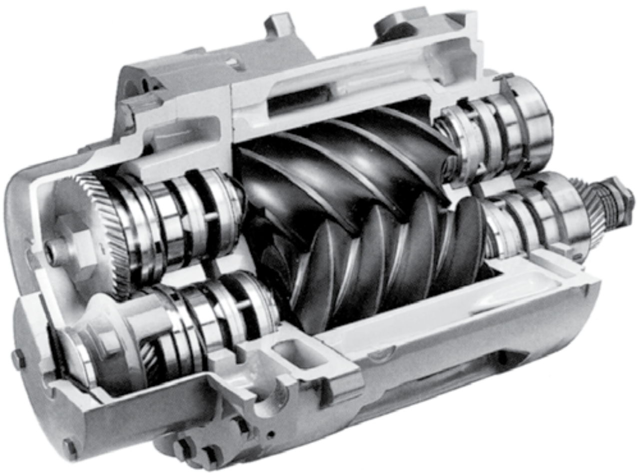

Figure 2.6 Lube Free Screw Air End

Dry type oil-free rotary screw compressors have a range from 20 - 900 hp or 80 - 4,000 cubic feet per minute. Single stage units can operate up to 50 psig while two-stage generally can achieve 150 psig.

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In the ***water injected type***, similar timing gear construction is used but water is injected into the compression chamber to act as a seal in internal clearances and to remove the heat of compression. This is shown in Figure 2.7. This allows pressures in the 100 - 150 psig range to be accomplished with only one stage. The injected water, together with condensed moisture from the atmosphere, is removed from the discharged compressed air by a conventional moisture separation device.

Similar to the oil-injected type, oil-free rotary screw compressors generally are packaged with all necessary accessories.

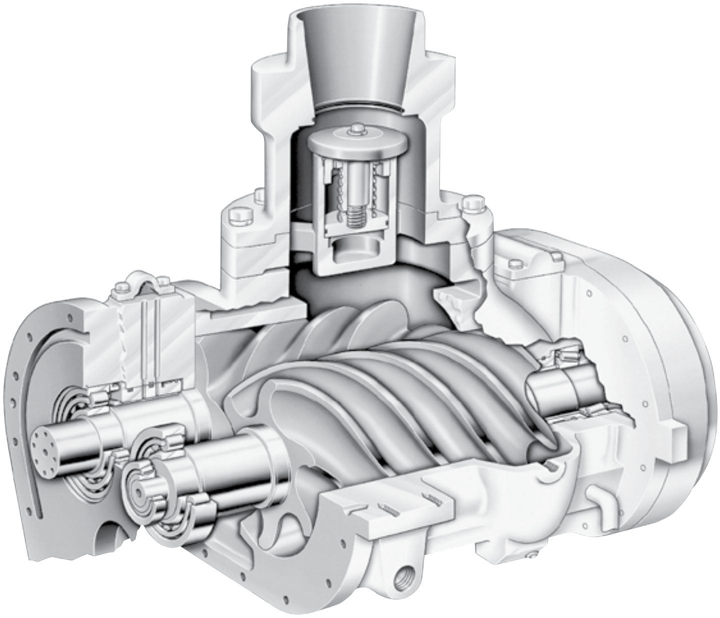

Figure 2.7 Oil Free, Water Injected Rotary Screw Compressor

Capacity Control

Capacity control for the ***dry type*** oil-free rotary screw compressor normally is the simple load/unload as sometimes employed for oil-injected rotary screw. Inlet valve modulation is avoided as this would result in a greatly increased pressure ratio and discharge temperature rise in the second stage. The inlet valve is totally closed except for a small, calibrated flow for cooling purposes. Simultaneously the unloading valve is opened, releasing any back pressure to atmosphere. Unloaded power of approximately 15 – 20% is normal.

Automatic start/stop control also may be used where demand is infrequent or sporadic. Dual control, combining automatic start/stop with load/unload is useful for long periods of high demand followed by long periods of low demand (e.g. day shift vs. night shift).

Capacity control for the water-injected oil-free rotary screw compressor normally utilizes inlet valve modulation as for the oil-injected type, the water taking away the heat of compression at increased pressure ratios combined with reduced mass flows.

Lubrication

Oil-free rotary screw compressors utilize oil for lubrication of bearings and gears, which are isolated from the compression chamber. The oil also may be used for element jacket cooling on air-cooled units due to the lack of cooling water.

Typically, an oil pump is directly driven from a shaft in the gearbox, assuring oil flow immediately at start-up and during run-down in the event of power failure. An oil filter, typically with 10 micron rating, protects bearings, gears and the oil pump from damage.

Cooling

The cooling system for the dry type oil-free rotary screw compressor normally consists of an air cooler after each stage and an oil cooler. These may be water cooled or air cooled radiator type. Some two-stage designs also employ an additional heat exchanger to cool a small portion of the compressed air for recycling to the compressor inlet during the unloaded period.

OIL-FREE ROTARY LOBE TYPE COMPRESSORS

The rotary lobe type compressor is a positive displacement, non-contact, or clearance type, design. With no mechanical contact within the compression chamber, lubrication within this chamber is eliminated. Lubrication of bearings, timing gears and speed increasing gears is all external to the compression chamber and shaft seals prevent any migration of oil to the compression chamber. This ensures oil-free compression and air delivery. A cross-sectional view of the intermeshing rotors in their stator is shown in Figure 2.8. The compression principle is illustrated in Figure 2.9. Two intermeshing rotors have lobe profiles that intermesh during rotation. Air flows into the compression chamber from the two inlet ports while the discharge ports are sealed by the rotor lobes. Rotation continues until the two discharge ports are exposed to the compression chamber, when the air is discharged. The dual inlet/outlet ports eliminate any axial thrust loads.

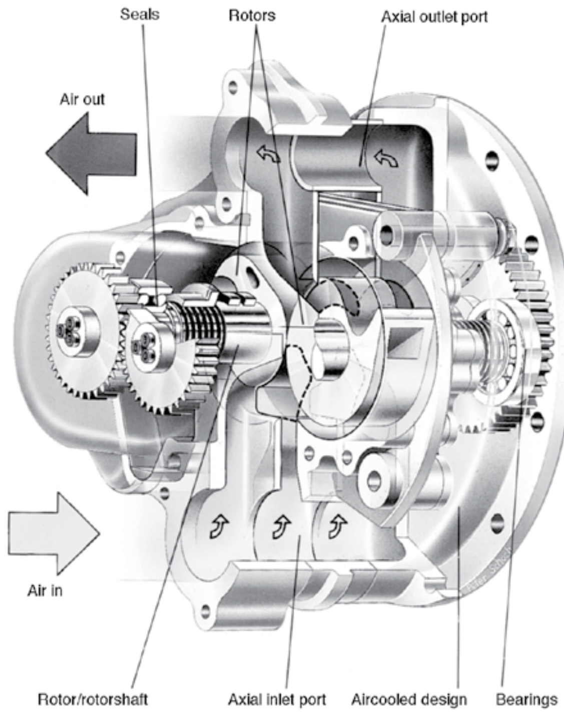

Figure 2.8 Oil-Free Rotary Lobe Compressor

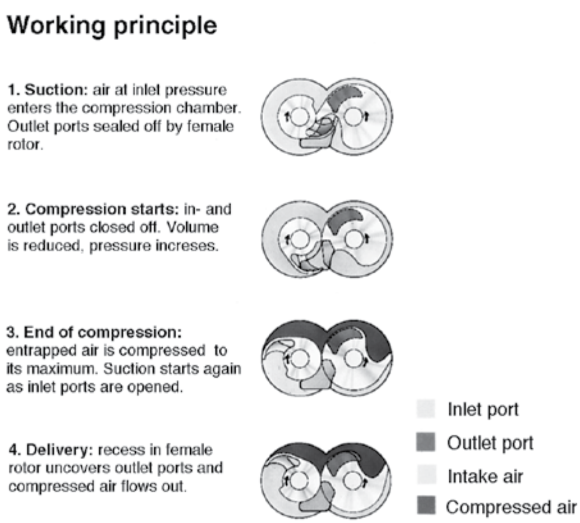

Figure 2.9 Rotary Lobe Compression Principle

As in other displacement type rotary compressors, no inlet or discharge valves are incorporated. Each stage has a fixed built-in volume (or pressure) ratio.

For most industrial air compressors operating in the 80 - 125 psig range, two stages of compression are required to handle the heat of compression. Air leaving the first stage is passed through an air intercooler, where its temperature is reduced as close as possible to atmospheric temperature and resulting condensate drained off, before it enters the second stage compression chamber where it is compressed to the desired system pressure.

A typical air end consists of a cast iron stator housing that may have passage for air or water cooling. Rotational speeds are chosen to optimize volumetric efficiency for a given profile.

Like other oil-free rotary compressors, the normal method of capacity control is the common load/unload type previously described. Both stages are unloaded simultaneously. A typical control system is shown in Figure 2.10.

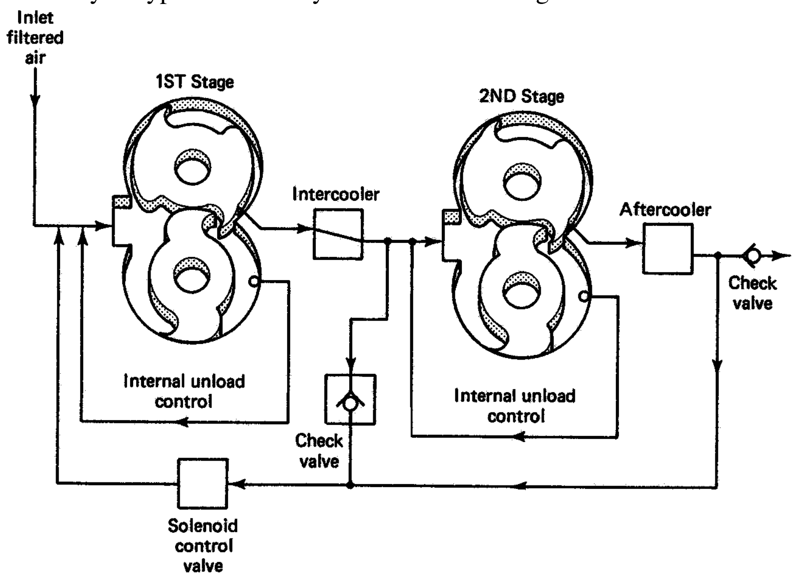

Figure 2.10 Method of Unloading the Rotary, Oil-Free, Lobe-Type Compressor

Both air cooled and water cooled packages are available from 25 - 75 hp with capacities from 85 to 315 cfm.

ROTARY SCROLL TYPE COMPRESSORS

The rotary scroll compressor has become a popular compressor as a domestic air conditioning refrigerant compressor. More recently it has been introduced to the standard air compressor market in the lower end of the horsepower range of rotary air compressors.

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The operating compression principle is accomplished by means of two intermeshing spirals or scrolls, one scroll being stationary and the other orbiting in relation to the stationary scroll. See Figure 2.11. The stationary scroll is shown in black and the orbiting scroll in white. Air entering through the suction port in the stationary scroll, fills the suction chamber consisting of the outer labyrinth of the stationary scroll and on the outside edge of the orbiting scroll, as shown in the illustration at position 1. At this position, the portion of the compression chamber at an intermediate pressure is sealed by adjacent portions of the two scrolls.

Figure 2.11 Operating Principle for a Scroll Compressor

As orbiting continues, the space occupied by the air becomes progressively reduced as shown in steps 2 through 5 and moves progressively toward the discharge port in the center of the stationary scroll.

It should be noted that the flow through the Suction Port and through the Discharge Port is continuous, providing pulsation free delivery of compressed air to the system. There is no metal to metal contact between the scrolls, eliminating the need for lubrication in the compression chamber and ensuring oil free air delivery from the scroll compressor. However, without the removal of the heat of compression, the efficiency is less than comparable oil injected air compressors.

Current models are air cooled and range from approximate 6 to 14 acfm, 2 through 5 hp, with discharge pressures up to 145 psig. This size range is expected to steadily increase. Noise levels with a sound attenuating canopy are extremely low, in the range 52 - 59 dBA at 1 meter, in accordance with the CAGI/Pneurop test code.

LIQUID RING ROTARY COMPRESSORS

The liquid ring (or liquid piston) rotary compressor also is a positive displacement type compressor. The mode of compression is similar to that of the sliding vane rotary compressor but the vanes (or blades) are fixed on the rotor. See Figure 2.12. The stator bore may be circular with the rotor eccentric to it, or elliptical with the rotor concentric to it. The former provides one compression per revolution while the latter provides two.

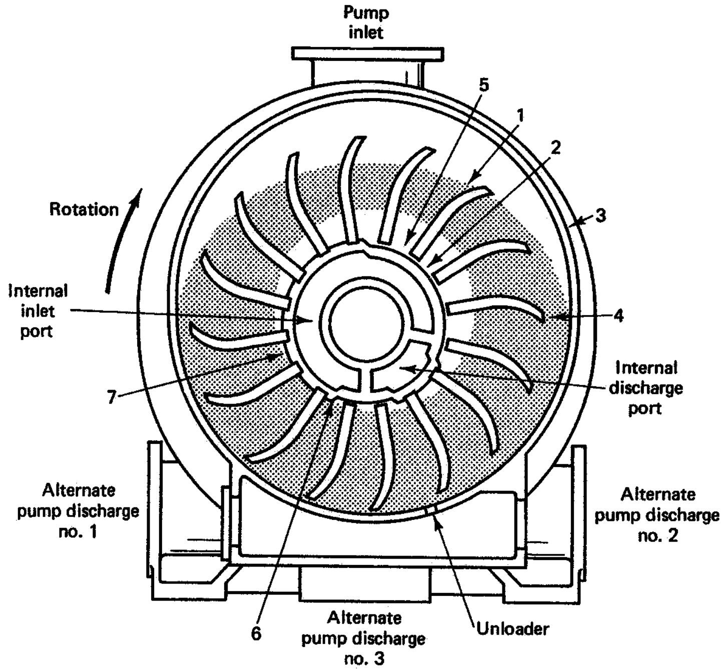

Figure 2.12 Cross Section Diagram of a Liquid Piston Rotary Compressor

A ring of liquid is swirled around the bore of the stator as the rotor turns. The depth of penetration of each vane, or blade, into the liquid, varies as rotation occurs. The space between the rotor hub and the liquid ring therefore varies. Axial inlet porting allows air to fill the space between adjacent vanes until its maximum volume. Further rotation then causes the space to be reduced and compression to occur until the discharge port is reached when the compressed air is discharged. The liquid ring also serves to remove the heat of compression and is discharged into gas-liquid separator, which removes the liquid from the gas. Because of the intimate contact of gas and liquid, the final discharge temperature can be held close to the supply temperature of the liquid, eliminating the need for an aftercooler.

Vapors are condensed in a liquid ring compressor when the liquid is cooler than the saturation temperature of the air/vapor or gas/vapor mixture in the compressor. The liquid ring, therefore, is acting as a condenser. The condensate becomes part of the sealant liquid during the compression cycle and is removed from the gas-liquid separator with the sealing fluid.

The liquid also scrubs the gas, removing solid particles from significant down to micron sizes, without damage to the unit unless the particles are abrasive.

The lubricated bearings are external to the compression chamber and isolated from it, so oil-free compression is obtained.

Capacities range from 2 to 16,000 acfm with a discharge pressure up to 35 psig in a single stage and 125 psig in a two stage version. As a compressor it is much less efficient than other rotary positive displacement types due to the energy required to swirl the liquid in the stator.

This type of design is used most commonly as a vacuum pump up to 26 in. Hg., single stage. Two stage units can achieve higher vacuum levels. This type of vacuum pump is used widely in the pulp and paper industry.

Compression of gases other than air are possible and a liquid is chosen, which is compatible with the gas being compressed. Rotor and stator materials may also require to be changed.

PORTABLE AIR COMPRESSORS

History

The history of the portable compressor follows closely the history of the construction and mining fields. These industries required compressors that could be easily moved with the work. The portable compressor, therefore, is a complete air compressor plant, sufficiently light in weight, yet strong enough to withstand the severe service encountered in construction and mining work.

Stationary air compressors were well established in industry by the turn of the century. They were heavy, large machines that required bulky, solid foundations. Often, it was necessary to use long lengths of air hose or pipe to reach the work, and this resulted in losses due to friction and leakage.

Around 1900, portable air compressors were introduced. These compressors were little more than stationary compressors on wheels and were limited in their application by the drills of that period, which weighed up to 500 lb and did less work than today's lightest drills while using a great deal more air.

In 1910, the portable compressor in most common use had one large, single-stage compression cylinder driven horizontally by a steam or oil engine. Probably the greatest single factor that stimulated portable compressor development was the advent of the lightweight air drill.

In 1933, the first two-stage, air-cooled portable compressor was manufactured. Shortly afterward, compressor manufacturers established standard sizes and rated portable compressors on actual free air delivery. In 1938 and 1939, portable compressors became more modernized, with pneumatic tires and streamlined enclosures. By 1939, the first multispeed regulation system had been introduced. This allowed the compressors and engines to idle when little or no air was demanded. In the late 1940s, regulation was improved by providing variable engine speed and controlled air intake flow throughout the air requirement range. By the late 1950s, selective loading of cylinders and simultaneous variable-speed features were added to reciprocating portable compressors.

In the 1950s, the oil-injected, sliding-vane rotary compressor was introduced. Higher-speed, overhead-valve engines made possible considerable reduction in the size and weight of the portable compressor. Oil injected into the rotating compressor acts as a coolant, lubricant, and sealant. The oil is then separated from the air, cooled, filtered, and reinjected.

In 1961, the first oil-injected, rotary-screw compressors were manufactured in this country. Since then, improvements in oil separation and cooling systems for these rotaries have resulted in lightweight units. Now, materials have also made the reciprocating units lighter and more maneuverable. Oil-free, rotary-screw portable compressors, which do not require oil in the compression chamber, also became available for primarily industrial applications.

In 1968, the first quiet or silenced portable compressors were introduced into the construction market. They were sold as hospital and residential noise attenuated machines (Fig. 2.13) in and around the Eastern megalopolis. Since 1978, all portable compressors sold in U.S. commerce that are 75 cfm and larger have been required by law to comply with the sound levels stipulated by the EPA in the Noise Control Act of 1972.

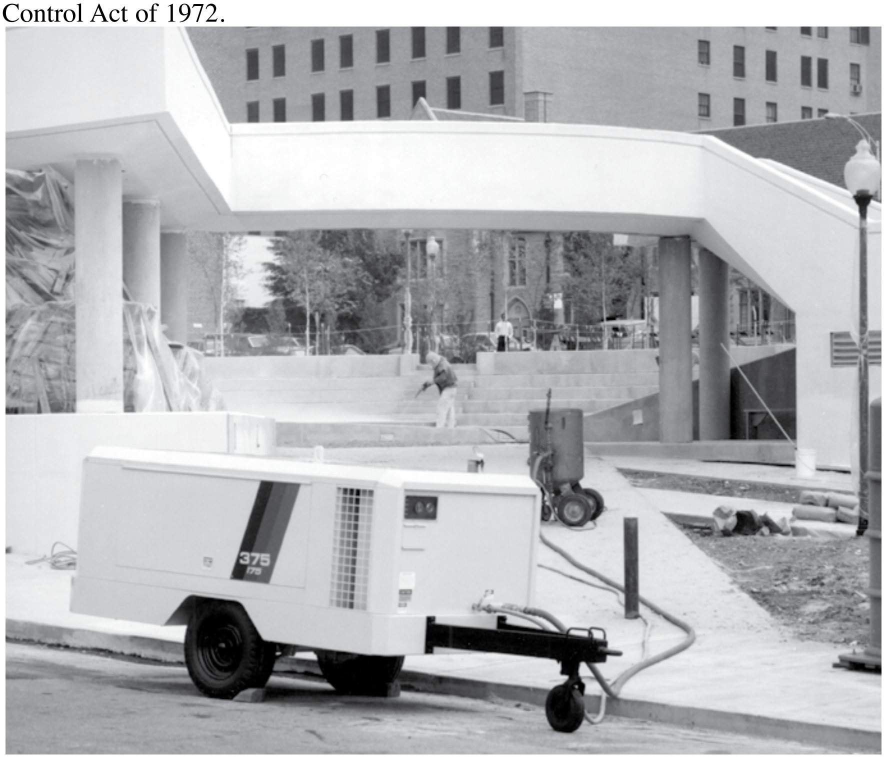

Figure 2.13 Portable compressor with a noise limiting housing suitable for use in residential areas, even near hospitals.

In 1982, a new type of two-stage portable air compressor was introduced. It uses a diesel-exhaust-driven turbocharger to drive centrifugal compressors as the first stage, and the same diesel mechanically drives the reciprocating piston compressor as the second stage.

Description

A portable air compressor is usually defined as a self-contained unit mounted on wheels. The unit consists of an air compressor, prime mover with silencing, cooling, control, air induction, lubrication, fuel tank, exhaust, and starting systems.

Additionally, reciprocating compressors and non-oil-injected compressors are equipped with air receivers requiring no oil separators. Oil-injected rotary units are equipped with oil-air separator systems, which also act as air receivers. Since skid-mounted units are generally obtained by removing the running gear from a portable unit, they are usually classified with the portable unit. Such a unit is seen in Fig. 2.14. Mounting designs vary, depending on weight and application of the unit. Lighter-weight units with low air capacities (below 125 cfm on 210 cmh) are usually mounted on two wheels and are predominantly gasoline engine driven, but compressors as high as 600 cfm may now be mounted on two wheels. Manufacturers now offer both gasoline- and diesel-driven units up to 250 cfm, and they are generally two-wheel units. Larger units, 250 cfm and up are normally two- or four-wheel units and diesel driven.

Figure 2.14 Three portable air compressors provide high pressure compressed air for pile driving operations on a U.S. interstate highway project. Each compressor delivers 1600 cfm of air at 150 psig.

A less common but highly useful type of portable compressor is the electric-motor-driven machine (Fig. 2.15). In external appearance, this compressor looks similar to engine-driven portable compressors, but the prime mover is an electric motor having the proper electrical and mechanical characteristics for the compressor and its application. The electric motor starter is usually an integral part of such a portable compressor package.

Large manufacturing, process industries, petrochemical, and refinery facilities find the electric-driven portable compressor useful. Furthermore, in some cases the engine-driven compressor is undesirable for reasons of safety in some of the aforementioned facilities.

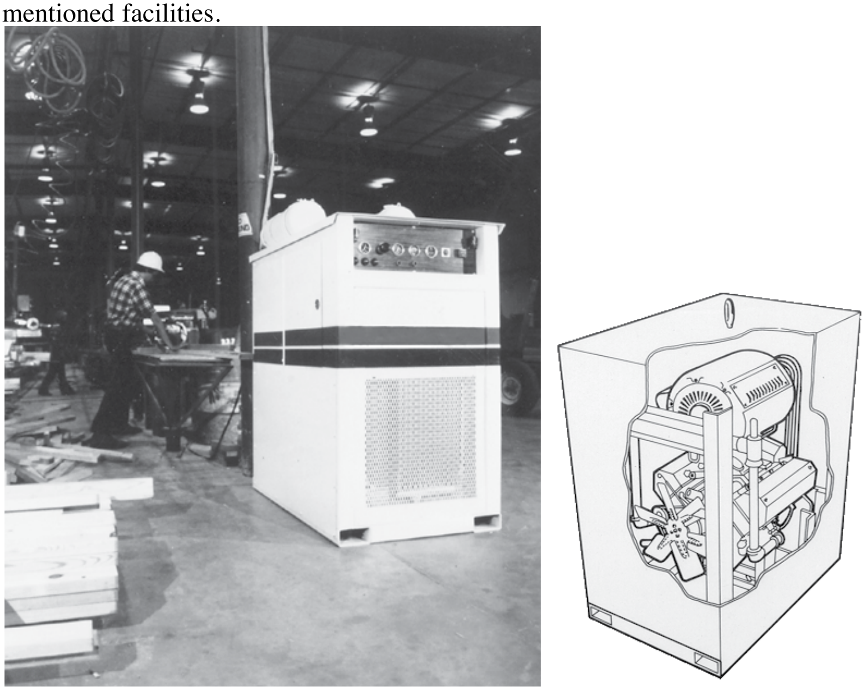

Figure 2.15 Sixty horsepower electric motor driven reciprocating compressor, located near work-stations without hazard or annoyance to workers. Units are virtually vibration free and require no foundation or bolting to the floor.

Muck-mounted units are used by utility companies because they require a highly mobile, relatively small compressor of usually 160 cfm or less for their particular type of small, short job. The same truck with its air compressor may move through congested city streets and be used on as many as three or four different jobs in one day. A truck-mounted booster unit is seen in Figure 2.16.

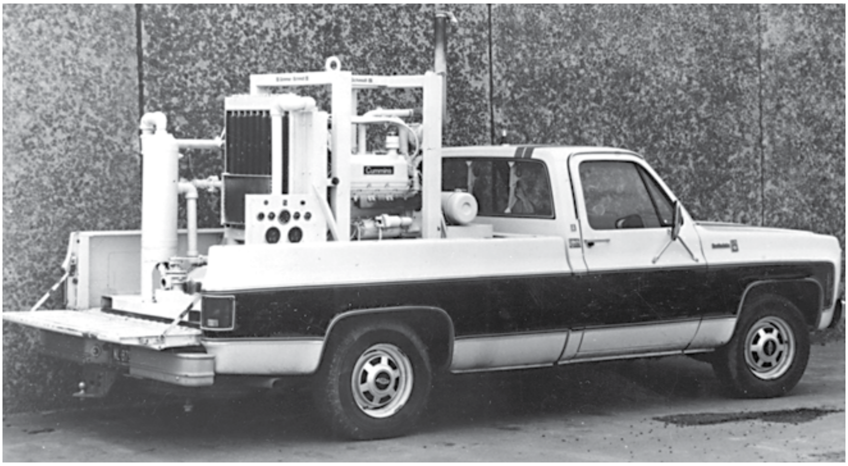

Figure 2.16 Portable booster compressors are now compact enough to fit into the bed of a pickup truck. This unit boosts 750-900 cfm from 250-350 psig to 900 psig.

One type of truck-mounted unit is the utility skid. This is a skid-mounted unit, normally not larger than 175 cfm without wheels. It is mounted across the frame behind the cab inside the utility body. A second type of truck-mounted unit is the power take-off driven type, which utilizes the truck engine for its prime mover. The extra cost for the vehicle and drive line maintenance are considerations in this configuration. Noise emissions are a major factor in these installations because of their use in populated areas.

Self-propelled compressors have the advantage that they can be moved without the need for a separate towing unit. Such compressors can be equipped with attachments to accomplish other work. Attachments include either rear- or front-mounted rock drills for highway drilling, front-end loaders, and backhoes.

Capabilities

Portable compressors are manufactured in sizes ranging from 20 to 5000 cfm, with delivery at 100 to 250 psig operating pressure. The model numbers normally designate the approximate air delivery, and size increments are such as to cover the full range available. Figures 2.17 and 2.18 are typical of the construction of a 175 and 742 cfm unit.

Figure 2.17 A silenced portable compressor in use in a residential area.

Figure 2.18 A 742 cfm portable compressor running several hand-held drills on a construction site.

Pressure

Since the most common application of the portable compressor is to operate air-powered tools (Chapter 5), the units are designed and rated for 100 psig or higher discharge pressure. The units normally have sufficient reserve for operating at high altitudes. Some manufacturers offer portable compressors and skid-mounted units with higher pressure ratings, up to 350 psig for the increasingly common higher-pressure applications, such as down-the-hole hammer, pipeline testing, oil- and gas-well servicing, sandblasting, rock drilling, and pile driving.

Air Receiver

A reciprocating portable compressor is normally equipped with an approved ASME pressure vessel. The ratings of temperature and pressure stated on the ASME data plate attached to the ASME-approved vessel must not be exceeded.

Oil-injected rotary compressors are equipped with an air-oil separator that is an approved ASME pressure vessel. The vessel contains the separator system and also serves as the compressor lubricant and compressed-air storage vessel.

Receivers are equipped with a safety valve for protection against excessive pressures and with a drain valve for removing moisture that will accumulate. The manufacturer's instruction manual should be consulted for the proper procedure for draining moisture from the receiver.

Fuel Storage Tank

Fuel tanks normally contain sufficient fuel for eight hours of operation. A fuel strainer is usually furnished and should be kept clean and in good condition. The fuel tank should be drained regularly to avoid problems from moisture condensation and accumulation of foreign particles in the tank.

Lubrication System

The lubricating system is a critical part of all designs of portable compressors. The manufacturer's recommendations should be followed in regard to lubricating oil specifications. This applies to oil-injected and dry rotary compressors, as well as reciprocating compressors.

Regulation

Since the portable compressor is most widely used to furnish air power for operating pneumatic tools and other devices designed to operate on constant pressure, the manufacturer furnishes, as standard equipment, a regulating system designed to hold the discharge pressure practically constant while the volume of air delivered to the tools varies with the demand. The manufacturer's instruction book contains detailed instructions for the care and maintenance of the particular design of regulating system, but one requirement common to all types of regulating devices is that they must not be tampered with unnecessarily nor by persons unfamiliar with their functioning and adjustment. Figure 2.19 shows a capacity curve for a typical portable control with proper adjustment.

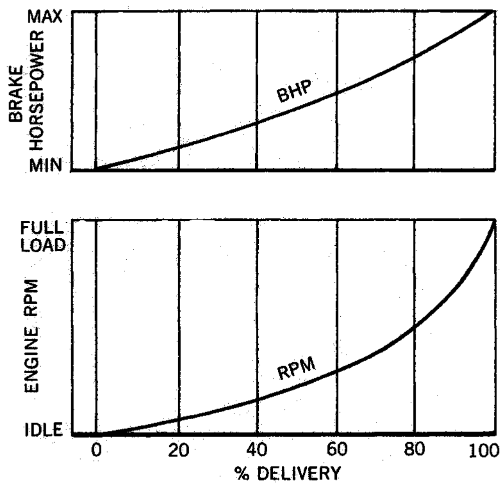

Figure 2.19 Capacity curves for a typical portable compressor capacity control.

Towing

Air compressors towed on highways are subject to state, local, and federal regulations. Department of Transportation federal regulations presently require brakes on equipment weighing 3000 lb gross weight or less if W3 is greater than 40% of the sum of W1 and W2 (Fig. 2.20).

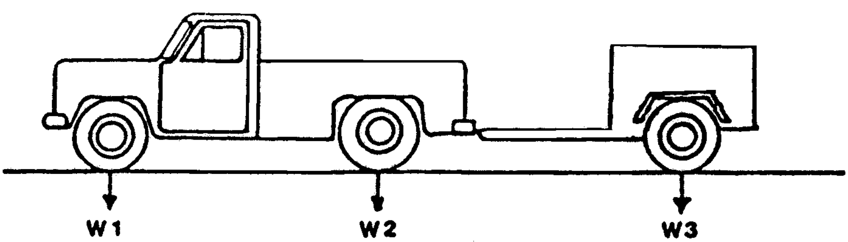

Figure 2.20 Federal regulations require brakes on towed compressors weighing 3000 pounds or less, depending upon the weight distribution W1 W2, and W3.

Applications

Portable compressors have wide and varied uses in both construction and industrial applications. Figures 2.21, 2.22, and 2.23 show some typical as well as some unusual applications. Other examples of applications are described elsewhere.

Figure 2.21 The drill on this air tool is working in rock six feet below ground level. The hole will accommodate the steel reinforced foundation for a highway sign. The compressor is a 100 cfm, 100 psig unit.

Figure 2.22 Two portable compressors being readied for the job of sandblasting a ship in dry dock to remove rust. The housing doors will be closed during the actual work.

Figure 2.23 A 265 cfm, 125 psig silenced portable air compressor on site at an oyster shell barge.

Typical Specifications

To aid prospective purchasers, the Compressed Air and Gas Institute offers the following as a set of typical specifications that may be used in the procurement of engine-driven portable compressors:

1. *General:* The portable air compressor shall consist of an air compressor and a gasoline or diesel engine rigidly connected together in permanent alignment and mounted on a common frame. The air compressor and engine shall be monoblock or direct connected through a heavy-duty industrial type clutch or coupling. The portable air compressor shall be provided with complete cooling, lubricating, regulating, starting, silencing, and fuel systems, fuel tank, air receiver (if required), and other equipment to constitute a complete self-contained unit. The unit shall be new and unused, if a model currently manufactured.
2. *Compressors:* The compressor shall deliver its rated capacity of free air per minute when compressing to a discharge pressure of 100 psig.
3. *Engine:* The engine shall be of ample power to drive the compressor at full load. It shall be equipped with suitable air filter and muffler.
4. *Cooling system:* The cooling system shall be of suitable design and sufficient capacity for satisfactory operation in an ambient temperature of 125˚F. A radiator guard shall be furnished.
5. *Regulation:* The unit shall be equipped with a regulator to vary automatically the volume of air delivered so as to meet the air demands when they are less than full capacity.

6. *Capacity ratings:* The capacity of the unit should be specified as the minimum requirement for the equipment to be operated, plus some allowance for normal decrease in efficiency. The manufacturer will then select the unit(s) nearest this capacity.
7. *Air receiver:* If one is required, the air receiver shall be made according to the ASME code and approved by the National Board, and the portable compressor shall be complete with pressure gage, safety valve, service manifold, and drain openings.
8. *Mountings:* The assembled portable air compressor shall be on a heavy frame equipped with one of the following:
   a. Two pneumatic tires with suspension springs or torsion bar suspension system
   b. Four pneumatic tires with suspension springs
   c. Skid mounts
9. *Housing:* The unit shall be enclosed by a complete housing with removable or hinged side doors that can be securely locked in place. Noise emission must meet current EPA specifications.
10. *Instrumentation:* The unit shall be equipped with the instruments required for the safe and efficient operation of the machine and should be enclosed in the above lockable housing or behind a lockable panel door.

RECIPROCATING AIR COMPRESSORS

Reciprocating compressors are positive displacement type in which a quantity of air or gas occupies a space that is mechanically reduced, resulting in a corresponding increase in pressure. A variety of such compressors is described in this chapter. Also included are vacuum pumps, which may be regarded as compressors having sub-atmospheric inlet pressure.

SINGLE-ACTING RECIPROCATING AIR COMPRESSORS

This type of compressor is characterized by its automotive type piston driven through a connecting rod from the crankshaft. Compression takes place on the top side of the piston on each revolution of the crankshaft, Figure 2.24. A design variation in small single stage oil-less compressors is a combined piston and connecting rod that tilts or rocks in the cylinder during its travel within the cylinder.

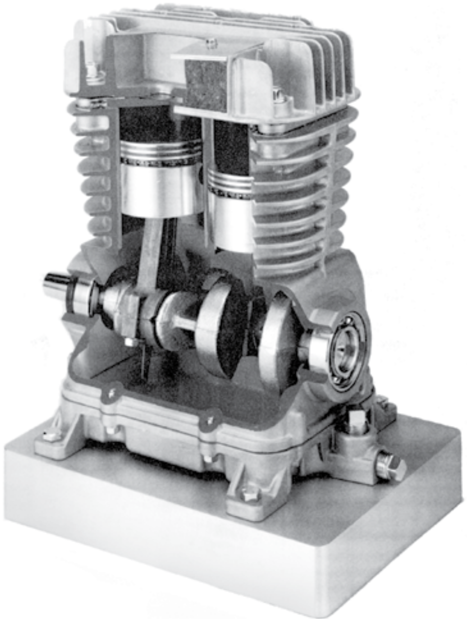

Figure 2.24 Two Cylinder Single-Acting Reciprocating Compressor

Single acting reciprocating air compressors may be air cooled, Figure 2.25 or liquid cooled, Figure 2.26, although the vast majority are air cooled. These may be single-stage, usually rated at discharge pressures from 25 to 125 psig, two-stage, usually rated at discharge pressures from 125 psig to 175 psig, or multi-stage for pressures above 175 psig.

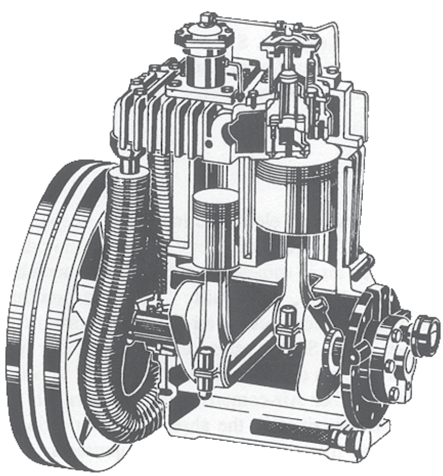

Figure 2.25 Air Cooled, Single-Acting, Two-Stage Reciprocating Compressor

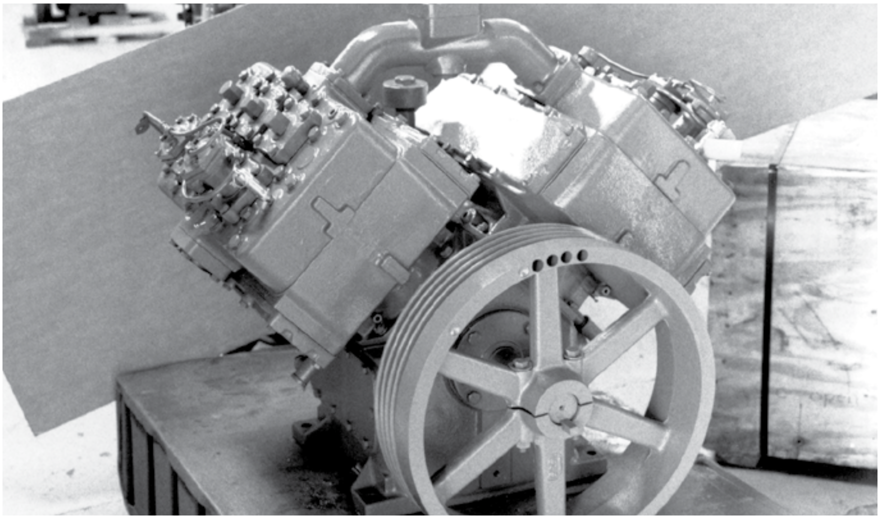

Figure 2.26 A Single-Acting Liquid-Cooled Compressor

The most common air compressor in the fractional and single digit hp sizes is the air cooled reciprocating air compressor. In larger sizes, single-acting reciprocating compressors are available up to 150 hp, but above 25 hp are much less common.

Two-stage and multi-stage designs include interstage cooling to reduce discharge air temperatures for improved efficiency and durability. Coolers may be air cooled, Figure 2.27 or liquid cooled, Figure 2.28.

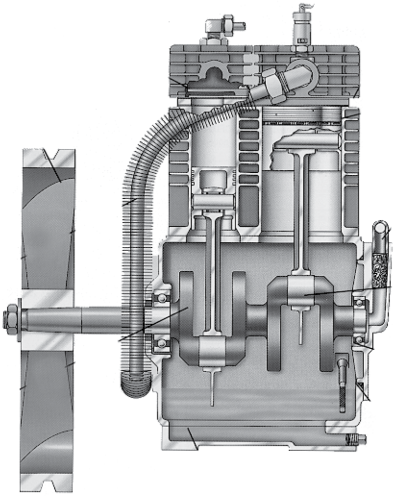

Figure 2.27 Air-Cooled, Single-Acting Reciprocating Compressor

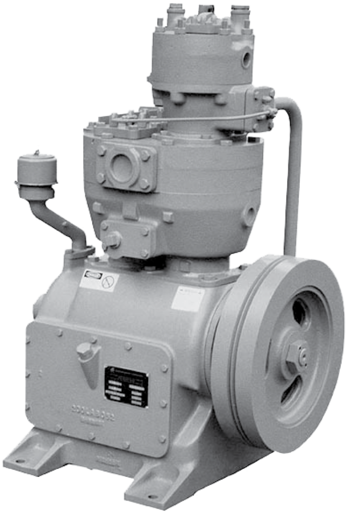

Figure 2.28 Water-Cooled, Single-Acting Reciprocating Compressor

Pistons

Pistons used in single-acting compressors are of the automotive or full skirt design, the underside of the piston being exposed to the crankcase. Lubricated versions have a combination of compression and oil control piston rings which:

1. Seal the compression chamber.
2. Control the oil to the compression chamber.
3. Act (in some designs) as support for piston movement on the cylinder walls.

In lubricated units, compression rings generally are made of cast iron and oil control rings of either cast iron or steel, Figure 2.29.

Figure 2.29 Single-Acting Compressor Cylinders and Crankshafts

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Oil-free, or non-lube, designs do not allow oil in the compression chamber and use pistons of self-lubricating materials or use heat resistant, non-metallic guides and piston rings that are self-lubricating. These are shown in Figure 2.30. Some designs incorporate a distance piece or crosshead to isolate the crankcase from the compression chamber.

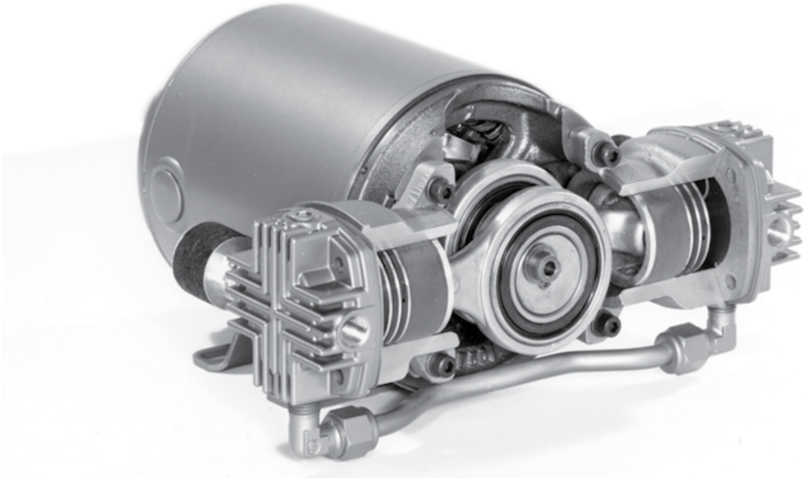


Figure 2.30 Balance-Opposed, Oil-Free, Single-Acting Reciprocating Compressor

Oil-less designs have piston arrangements similar to oil-free versions but do not have oil in the crankcase. Generally these have grease pre-packed crankshaft and connecting rod bearings, Figure 2.31.

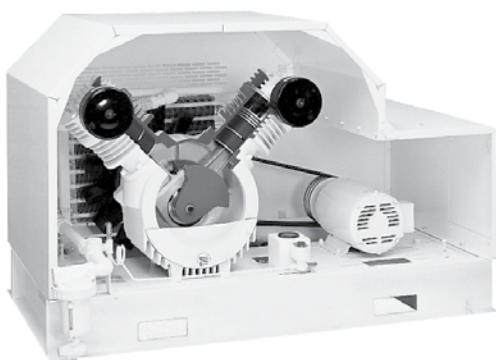


Figure 2.31 Oil-Free, Single-Acting Reciprocating Compressor

Cylinders

A variety of cylinder arrangements is used. These include:
1. A single vertical cylinder.
2. In-line or side by side vertical cylinders.
3. Horizontal, balance opposed cylinders.
4. V or Y configuration.
5. W configuration.

These are illustrated in Figure 2.32.

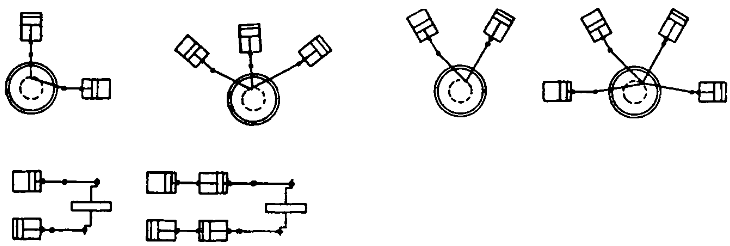
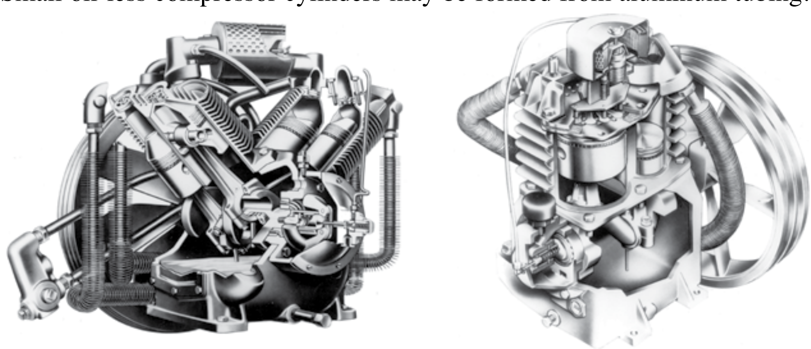


Figure 2.32 Various cylinder arrangements used in displacement compressors. Some are suitable for single-acting compressors, while others are double-acting, and require a cross-head and guide.

The number of cylinders is dependent on the capacity of air required and the number of stages. Cylinders may be separate castings, cast en-bloc, or a combination of the two, as shown in Figures 2.33A and B. Cylinders and heads for air cooled designs normally have external finning for better heat dissipation. Materials may be cast iron or die cast aluminum, with or without an iron or steel bore liner. Small oil-less compressor cylinders may be formed from aluminum tubing.



Figures 2.33A and 2.33B Cylinders may be individual castings (A), cast en-bloc, or a combination of the two (B).

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Valves

In general, both inlet and discharge valves for single-acting compressors are of the automatic pressure type that open and close on a small differential pressure. To secure rapid action, the valve elements are made light in weight and are proportioned for low lift. As a valve opens, increasing spring pressure minimizes the impact forces on the valve elements and reduces noise levels. In some designs, cushioning pockets, which form as the valves approach the full open position, further minimize impact. There are three types of compressor valves. These are:

1. The reed type, Figure 2.34, has only one moving part, which flexes between its closed and open positions. It requires no lubrication, has the advantage of low clearance volume with high flow area and is easily cleaned or replaced. Reed valves may be designed in various configurations for a tandem, two-stage cylinder arrangement as shown in Figure 2.35.
2. The disc type, Figure 2.36, includes a flat disc, ring or plate, which seats on the edges of a slightly smaller opening in the valve seat. The disc is backed by a valve spring. Springs used for this type of valve are of various configurations, such as a coil spring or a spring shaped like the disc except for its wave deformity, which allows it to act as a wave spring. Some disc type valves have multiple springs. In some applications, several valves per cylinder are used.
3. The strip or channel type, Figure 2.37, which consists of a valve seat having a number of slots or ports and a corresponding number of valve strips or channels that cover and close off the slots. A bowed leaf spring for each channel, or the bowed valve strip itself, returns the valve strip or channel to its seat after the air has passed through. A stop plate limits the lift of the valve.

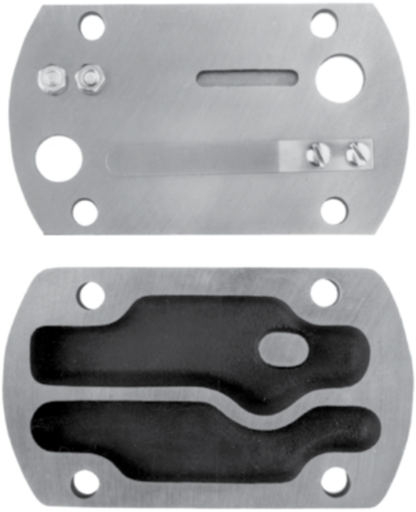

Figure 2.34 A reed valve showing the reed (upper view) and the air or gas passages, (lower view).

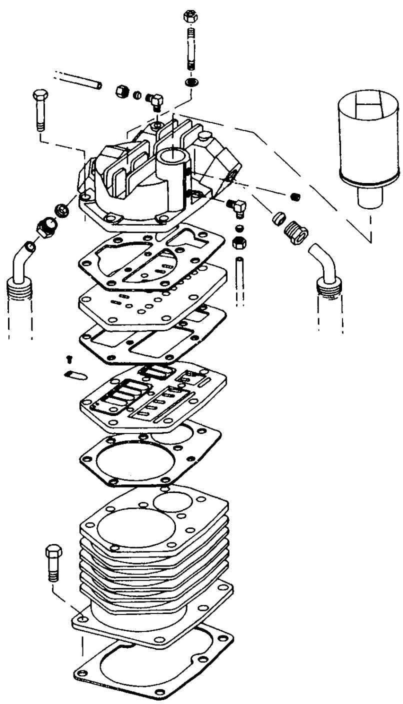

Figure 2.35 Cylinder, valve and head assembly, showing the reed valve for a tandem, two-stage cylinder arrangement.

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Figure 2.36 Disc valve, also known as a ring valve.

Figure 2.37 A channel-type valve. The bowed springs serve to return the valves to their seats.

Cooling

Single-acting air compressors have different arrangements for removing the heat of compression. Air cooled versions have external fanning for heat dissipation on the cylinder, cylinder head and, in some cases, the external heat exchanger. Air is drawn or blown across the fans and the compressor crankcase by a fan that may be the spokes of the drive pulley/flywheel. This is illustrated in Figure 2.38.

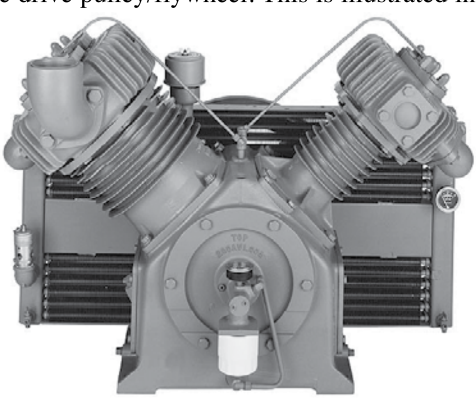

Figure 2.38 Air-Cooled Single-Acting Reciprocating Compressor

Liquid cooled compressors have jacketed cylinders, heads and heat exchangers, through which liquid coolant is circulated to dissipate the heat of compression. See Figure 2.39. Water, or an ethylene glycol mixture to prevent freezing, may be employed.

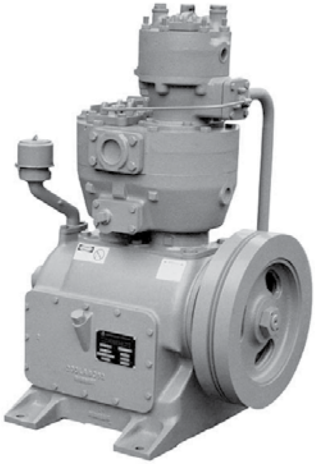

Figure 2.39 Water-Cooled Single-Acting Reciprocating Compressor

Bearings

The main crankshaft bearings usually are anti-friction ball or tapered roller bearings. Some designs may employ sleeve type main bearings. Crank pin and piston pin bearings normally are of the journal or sleeve type. Some connecting rod designs include precision bore bearings at the crank pin and needle roller bearings at the piston pin, or a plain piston pin. Figure 2.40 shows an exploded view of a typical design.

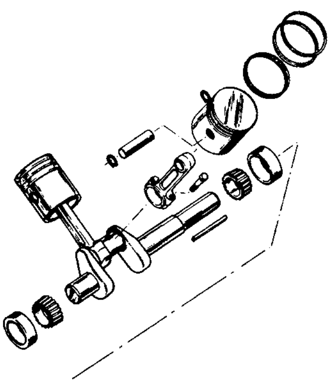

Figure 2.40 Blow-up of the working parts of a compressor showing tapered roller main bearings and insert-type connecting rod bearings.

Lubrication

Single-acting air compressors may utilize a splash or a full pressure lubrication system. A controlled splash lubrication system is shown in Figure 2.41. A dipper on the connecting rod dips into the oil reservoir in the crankcase each revolution and produces a splash of oil which lubricates the connecting rod, piston pin, main bearings and cylinders.

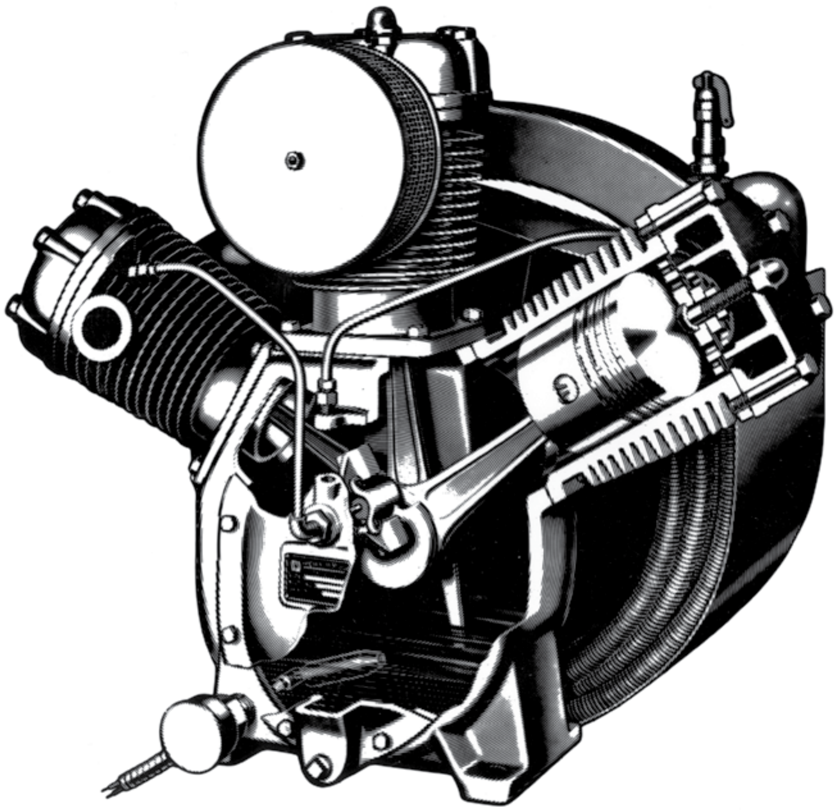

Figure 2.41 Splash-feed lubrication showing oil dipper and oil level float gage.

A variation, as shown in Figure 2.42, is the use of rings, running in a groove in the crankcase and dipping into the oil reservoir, to provide positive lubrication to critical wear areas.

A full pressure lubrication system is shown in Figure 2.43. In this design, a positive displacement oil pump draws oil from the reservoir in the crankcase and delivers it under pressure through rifle drilled passages in the crankshaft and connecting rod to the crank pin and the piston pin. This pressure system provides spray or drop lubrication to the anti-friction main bearings and cylinders.

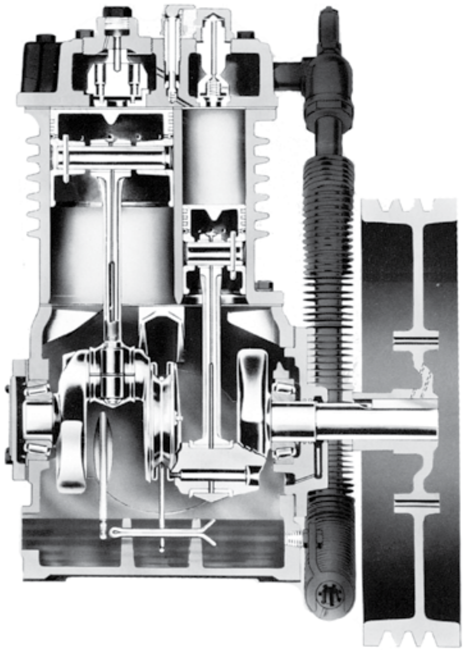

Figure 2.42 Splash-feed lubrication using oil rings.

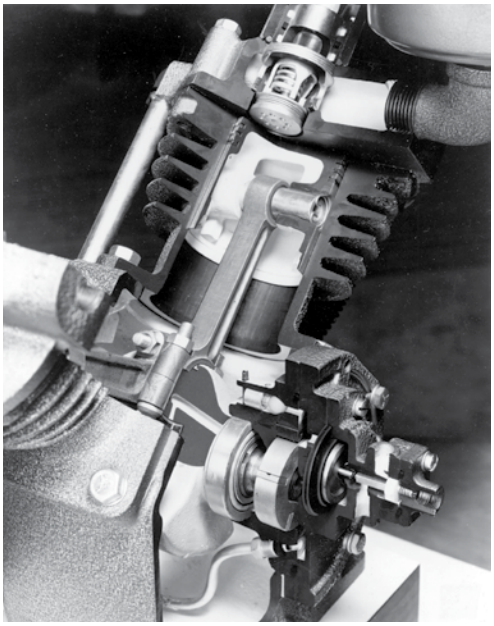

Figure 2.43 A full pressure lubricating system in a two-stage, air cooled, reciprocating air compressor.

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Exposure of the piston and cylinder walls to the oil results in some oil carry-over into the air stream delivered from the compressor and carbon deposits on valves and pistons. The specified oil should be used and changed regularly as specified by the compressor manufacturer.

As previously stated, oil-less designs use self-lubricating piston and guide materials and pre-lubricated and sealed bearings with no liquid oil in the crankcase.

Balance

Rotative and reciprocating motion produce forces that must be counterbalanced for smooth operation. The weight of the reciprocating parts and counterbalanced crankshafts are used to achieve optimum running balance. Minimal vibration not only minimizes maintenance but is essential for compressors mounted on an air receiver or storage tank. These may be referred to as tank mounted compressors. This type of air receiver must be capable of accepting not only the static pressures to which it is subjected but also the dynamic loading caused by unbalanced reciprocating forces from the air compressor and its driver mounted on it.

Drives

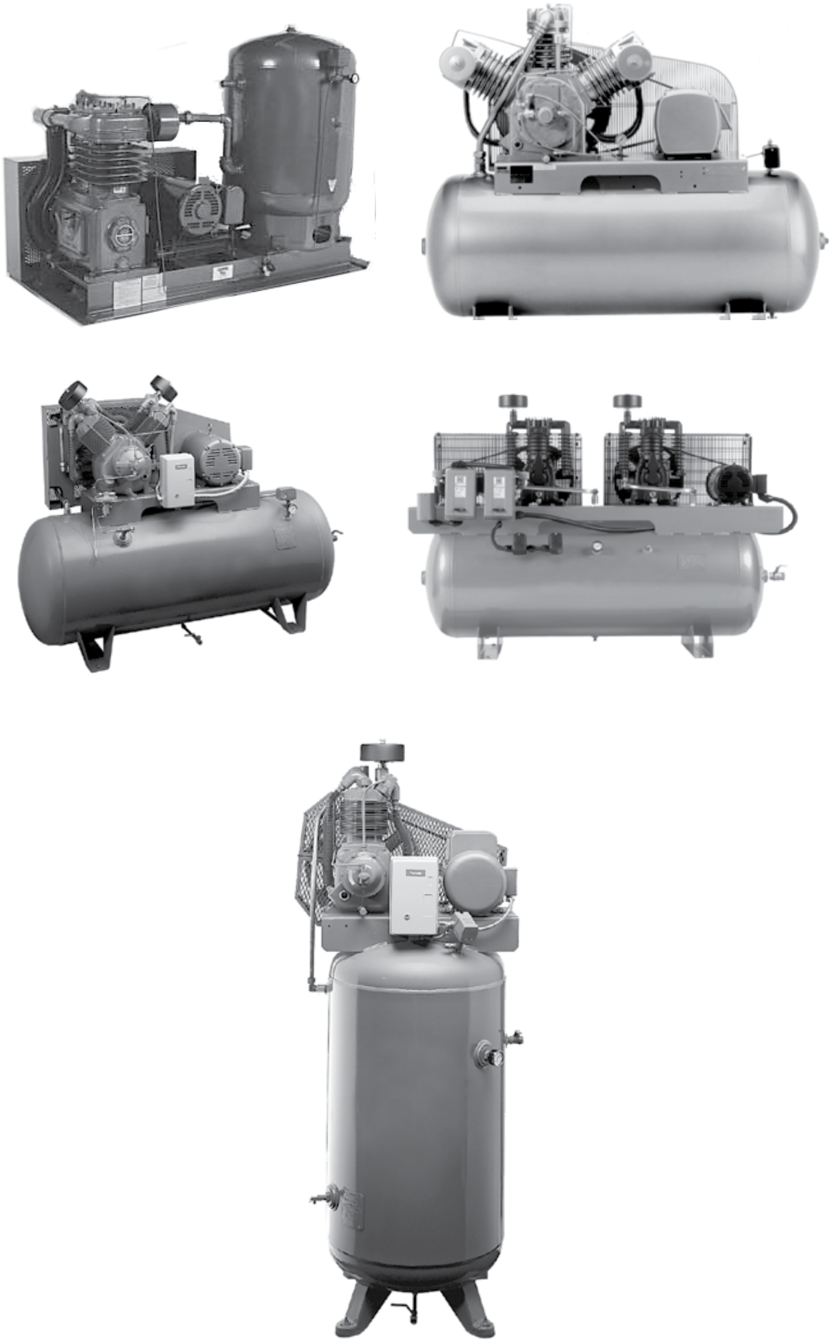
The most common drive arrangement is belt drive from an electric motor, internal combustion engine or engine power take-off. The compressor sheave also acts as a flywheel to limit torque pulsations and its spokes often are used for cooling air circulation. Belt drive allows a great degree of flexibility in obtaining the desired speed of rotation.

Flange mounted or direct coupled motor drives provide compactness and minimum drive maintenance. Belts and coupling must be properly shielded for safety and to meet OSHA requirements in industrial plants. Fractional horsepower compressors normally are built as integral assemblies with the electric motor driver.

Unit Type and Packaged Compressors

A packaged compressor may be defined as an air compressor with its driver and associated components self contained and ready for installation and operation. These may include the compressor, driver, starter, intake filter, cooling system with aftercooler and all necessary mechanical and electrical controls. An air receiver and interconnecting piping also may be included. These require no foundation and simplify installation. Figures 2.44 A, B, C, D and E illustrate these compressors. These compressors may use vertical or horizontal air receivers. Smaller models are available for portable use with wheel mounting or very small hand carried tanks.

Unit type compressors are very suitable for installation close to a point of use where a dedicated air compressor is desirable, as discussed in the chapter on Compressed Air Distribution Systems.



Figures 2.44A, B, C, D and E Various Unit-Type Compressors

Capacity Control

Start/Stop Control is the simplest form of control, in which a pressure switch, sensing system pressure, sends a signal to the main motor starter to stop the compressor when a pre-set pressure is reached. When pressure falls to another pre-set pressure, the pressure switch sends a signal for the compressor to be restarted. The pressure switch may have an adjustable upper pressure setting and a fixed or adjustable differential between the upper and lower pressure settings. Adjustments should be made only by qualified personnel and in accordance with the manufacturer's specifications.

An air receiver is essential to prevent too frequent starting and stopping, which affects the life of motor insulation due to high inrush current at each start. This type of control normally is limited to compressors in the 30 hp and under range. Its advantage is that power is used only while the compressor is running but this is off-set by having to compress to a higher receiver pressure to allow air to be drawn from the receiver while the compressor is stopped. This type of control is best suited to light or intermittent duty cycles. Air cooled reciprocating air compressors typically are rated for duty cycles ranging from 50/50 (50% on and 50% off) to 75/25 (75% on and 25% off).

Constant Speed Control allows the compressor to continue to run, even when there is reduced or no demand for compressed air. This term may be used with Load/Unload Control. In this type of control when the upper pressure setting is reached, a pilot device sends a signal to actuate an inlet valve unloader, Figure 2.45. A common method holds the inlet valve(s) open so that air is drawn into and pushed back out of the cylinder without any compression taking place. This also requires an adequate air receiver since air delivery is either 100% or zero but may operate within a narrower pressure differential than stop/start control. Load/unload capacity control should be used where the duty cycle is heavy and continuous.

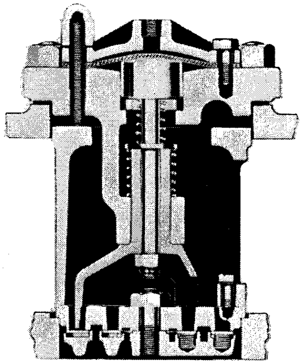


Figure 2.45 An intake valve unloader in which fingers hold the valve open until compressed air pressure is reduced.

Load/unload control also may be obtained using valves in the compressor discharge lines. When actuated by a pressure signal, the valve is held open so that air is released to atmosphere instead of being delivered to the receiver/tank. The air passes through the compressor discharge valves and possibly through the discharge line, before being released to atmosphere.

Internal combustion engine driven compressors normally use load/unload control to avoid the need to start and stop the engine. The same signal that operates the valve unloaders, can be used to operate an engine throttle control, so that the engine runs at idle speed with reduced power and noise during the unload cycle. The amount of power used by the compressors during the unload cycle is dependent on how well the design minimizes flow resistance into and out of the compression cylinder(s) and the magnitude of the mechanical friction.

When load conditions are changeable, special control systems are available that select start/stop or constant speed control to match the prevailing air demand cycle.

Operating Conditions

Stationary single-acting and unit type air compressors are adaptable to a wide range of conditions of temperature, altitude and humidity. High altitude or high humidity conditions may require a specially rated driver and cooling arrangements. Where oils are used, oils should be selected for the application and temperatures involved and in accordance with the manufacturer's recommendations.

Some applications of single-acting reciprocating air compressors are discussed in the chapter on Applications.

DOUBLE-ACTING RECIPROCATING AIR COMPRESSORS

This type of reciprocating air compressor uses both sides of the piston to compress the air. The piston is driven by a piston rod extending through a packing gland from a crosshead, which, in turn, is driven through a connecting rod from the main crankshaft. This is illustrated in Figure 2.46. The crosshead and its guide ensure that the piston and its rod operate in a straight line. Air is compressed as the piston moves in each direction of one complete stroke or one revolution of the crankshaft.

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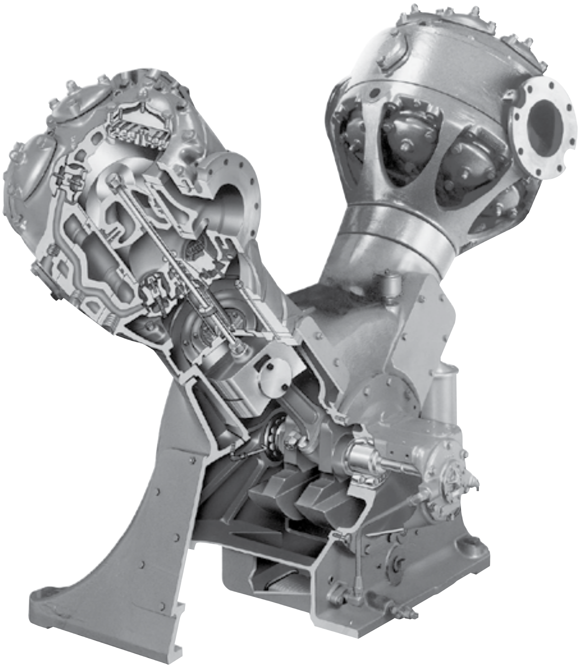

Figure 2.46 Double-acting compressor cylinder showing cooling water jackets around the cylinder and in the crank end cylinder head. Stud bolts shown are used to connect the cylinder to the distance piece between the cylinder and the crankcase.

The basic double-acting compressor has a single cylinder, single throw crankshaft, crosshead and connecting rod. The arrangement may be horizontal, as shown in Figure 2.47, or vertical, as shown in Figure 2.48.

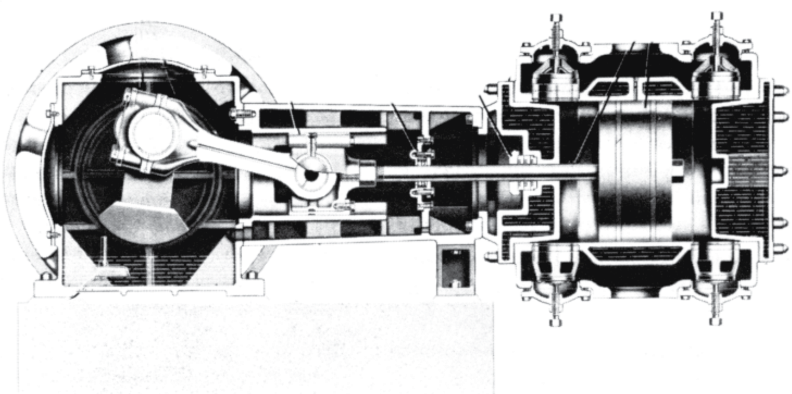

Figure 2.47 Single Cylinder, Single Throw, Horizontal, Double-Acting Compressor

Figure 2.48 A Single Cylinder, Oil Free, Double-Acting Compressor with Vertical, Single-Throw Frame

Multiple cylinder, double-acting air compressors, may have cylinders operating in parallel for increase flow rate, or in series for increased overall compression ratios. The common crankshaft may have single or multiple throws to drive the connecting rods and, hence, the pistons. Compressors have been built with as many as 10 crank throws on a single crankshaft. Figure 2.49 shows a two-stage air compressor with the first and second stages arranged at a right angle to each other, with a single throw crankshaft. Figure 2.50 shows a four cylinder arrangement having two first-stage cylinders operating in parallel and two second-stage cylinders also operating in parallel, also with a single throw crankshaft. Multi-stage compressors use water cooled heat exchangers after each stage to cool the air before entering the next stage, improving overall compression efficiency.

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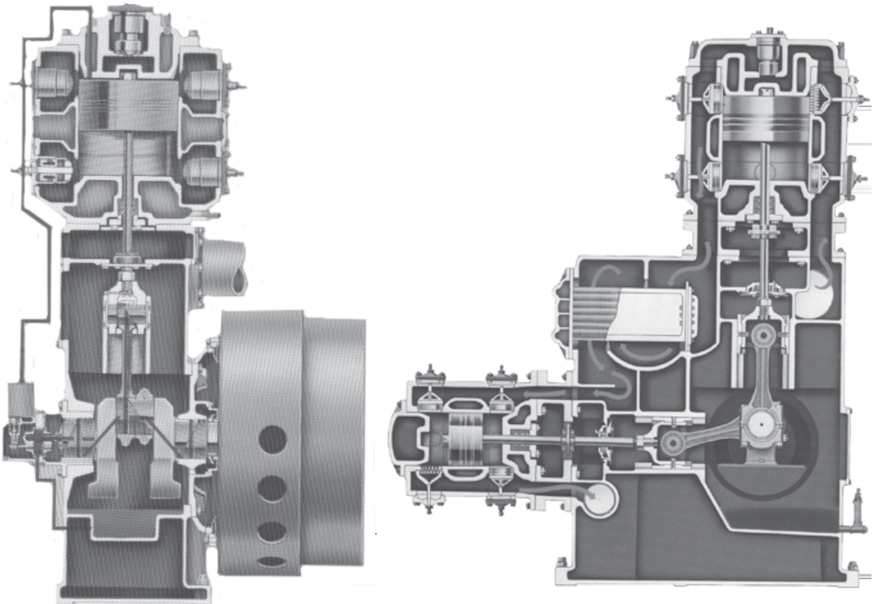

Figure 2.49 Two-Stage, Single Throw, Double-Acting Compressor with Vertical First Stage and Horizontal Second Stage Cylinder

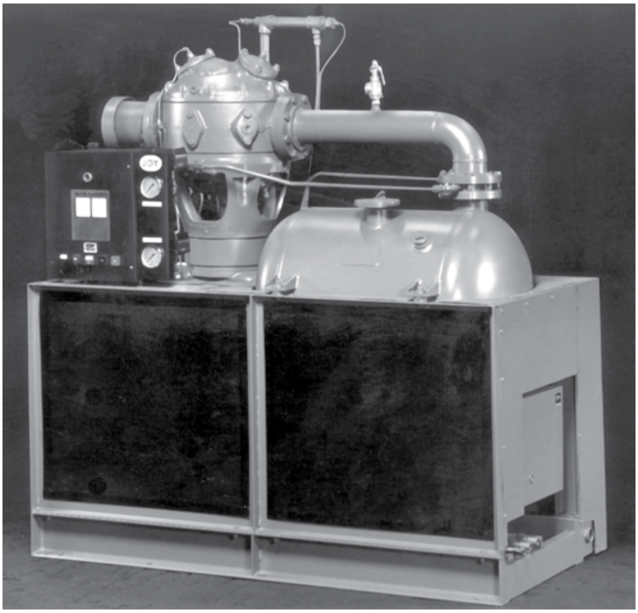

Figure 2.50 Packaged Vertical Single-Cylinder Compressor Complete with Air Receiver

Range of Sizes

Double-acting reciprocating air compressors range from approximately 10 hp to 1,000 hp, although for standard plant air applications, they have given way to rotary and centrifugal type air compressors. Discharge pressures up to several thousand psig are possible. Single-stage air compressors are common for 100 psig service but efficiency is improved with two stages and intercooling.

Types and Configurations

This type of compressor is a heavy duty, continuous service compressor. Cooling water jackets normally are incorporated in the cylinders and cylinder heads to remove some of the heat of compression, maintain thermal stability and improve lubrication, reducing carbonization of valve parts. Water cooling jackets around valves and piston rod packing are essential due to localized heating. Valves may be located in the cylinders, as shown in Figure 2.51, or in the cylinder heads, as shown in Figure 2.52.

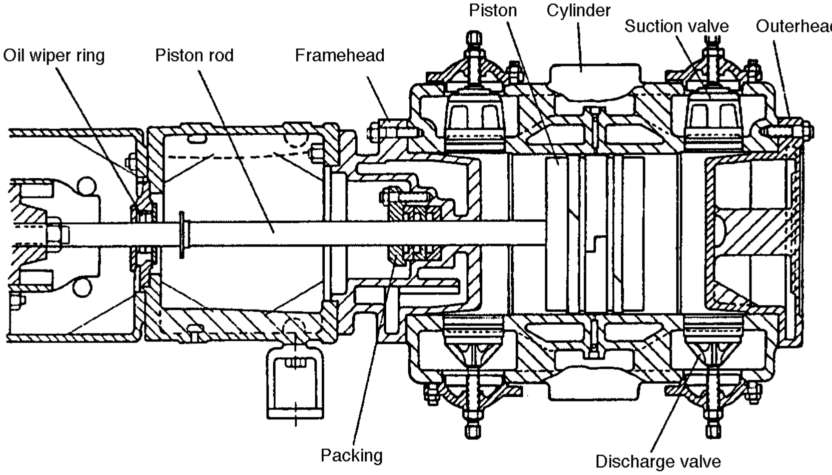

Figure 2.51 Horizontal Cylinder Arrangement with Suction and Discharge Valves around the Cylinder

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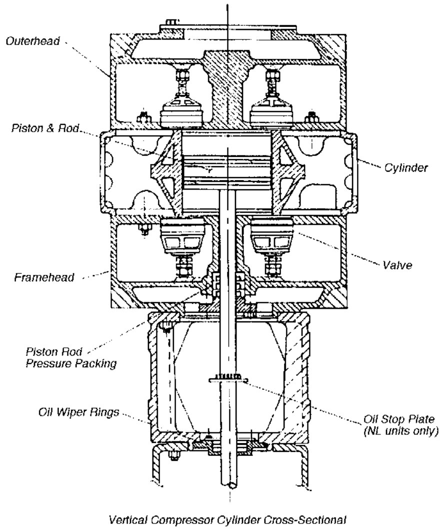

Figure 2.52 Cylinder arrangement with suction and discharge valves located in the cylinder heads.

A distance piece between the crankcase and the cylinder may incorporate the piston rod packing that prevents leakage of compressed air from the cylinder, along the rod and scraper rings that prevent migration of lubricant from the crankcase to the cylinder. An extended distance piece is used for oil-free, or non-lube compressors, to prevent any portion of the piston rod that enters the oil-free cylinder from also entering the lubricated crankcase.

Lubrication

Lubrication of the crankcase may be from a splash system, a forced feed system, or a combination of both. Cylinders are lubricated by means of a mechanical, force feed lubricator with one or more feeds to each cylinder. The type, size and application of the compressor will determine the method of lubrication and the type of lubricant. Modern synthetic lubricants now are common for cylinder lubrication.

Oil-Free, or Non-lube Compressors

These terms normally are used for air compressors that do not have any lubricant fed to the cylinder(s). Piston rings and rod packing usually are of PTFE-based materials, carbon, or other synthetic materials, which can operate without added lubrication.

In the majority of oil-free compressors, the piston rides in the cylinder bore on the synthetic or carbon wearing (or rider) ring or shoe, see Figure 2.53.

Alternatively, but less common, a tailrod and external crosshead may be added outboard of the cylinder head. With this arrangement, the weight of the piston is carried by the piston rod, supported by the main and external crossheads.

Where oil-free air delivery is not critical, a compressor with a standard length distance piece may be used, allowing a portion of the piston rod to enter both the cylinder and the crankcase. A small quantity of oil then migrates from the crankcase to the cylinder.

Where oil-free air is essential, the extended distance piece must be used, A baffle plate also is attached to the piston rod as a barrier to prevent the migration of oil along the piston rod, as shown in Figure 2.53.

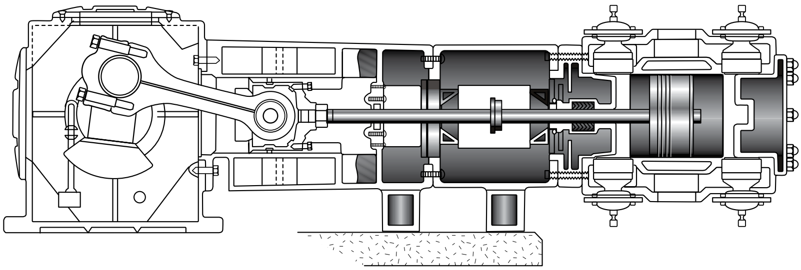

Figure 2.53 A non-lubricated compressor with piston riding on carbon wearing rings.

Capacity Controls for Double Acting Reciprocating Air Compressors

Reciprocating air compressors are positive displacement type, essentially having constant speed and capacity with variable pressure. The capacity can be varied to meet required demand by means of several types of capacity control.

These control systems are based upon maintaining the discharge pressure within prescribed limits. A pressure sensing element, or pilot, allows control air to operate the unloading mechanism. These are of three basic types:

1) Mechanically holding open the suction valves, allowing air drawn into the cylinder to escape without being compressed. See Figure 2.54.
2) The use of clearance pockets, which allows a predetermined portion of the compressed air to be diverted to the clearance pocket(s), then re-expand into the cylinder as the piston returns on its suction stroke. This reduces the amount compressed air delivered to the system and the amount of atmospheric air entering the cylinder.
3) Closing off the inlet air to the first stage cylinder of a two-stage compressor, then venting the second stage cylinder to atmosphere after a near vacuum has been reached within the compressor.

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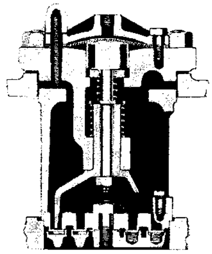

Figure 2.54 Suction valve unloader in which fingers hold the valve open until compressed air pressure is reduced.

Type 1) normally is used to provide compressor capacities of 0% or 50%, by holding open the suction valves on one end or on both ends of the cylinder. 100% capacity is obtained by allowing the suction valves to operate normally. This generally is called three step capacity control.

A combination of types 1) and 2) can provide 0%, 25%, 50%, 75% and 100% capacity and generally is known as five step capacity control. These are illustrated in Figure 2.55.

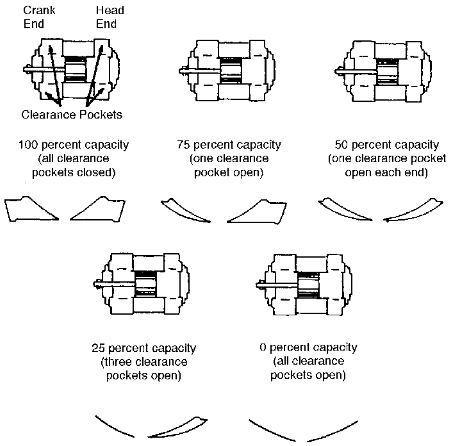

Figure 2.55 Capacity control steps by means of suction valve unloaders and clearance pockets.

Prime Movers for Double-Acting Reciprocating Air Compressors

Motive power for this type of compressor may be provided by one of the following principal drivers:

1. Electric motor.
2. Oil or gas engine.
3. Steam engine or turbine.

The most common driver is the electric motor, which is energy efficient and reliable. This may be induction type, synchronous type, wound rotor type or DC type, the first two types being the most common.

The compressor drive motor can vary by the type of connection between the motor and the compressor:

1. Motor with belt drive.
2. Flange mounted motor.
3. Direct connected motor.
4. Motor and flexible coupling.
5. Motor and speed reducing gearbox.

Belt drives generally are limited to about 150 hp with an 1800 rpm drive motor. The selection of other types depends on the specific characteristics of the compressor, including speed and torque. Arrangements generally are made for the motor to be started with the compressor unloaded. Flywheel mass may be necessary to keep current fluctuations within allowable standards. A typical compressor torque-effort diagram is shown in Figure 2.56. In all cases, NEMA Standards must be observed.

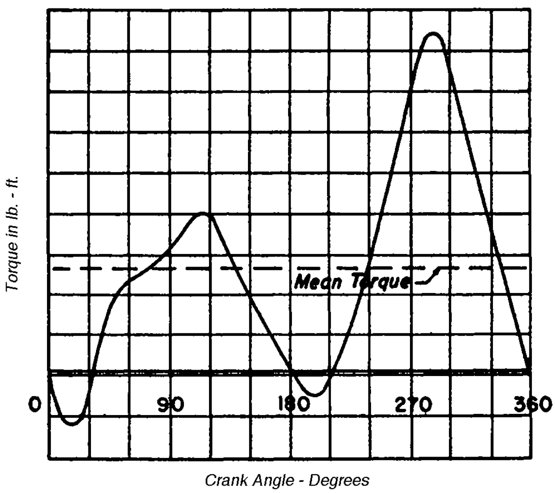

Figure 2.56 A Typical Torque-effort Diagram

Varying installation requirements will determine some of the required design characteristics of the drive motor. Typical enclosures include: Open Drip-Proof; Totally Enclosed and Explosion Proof (dependent upon type of risk in specific location). Non-sparking belt drive arrangements may also be necessary. Electrical controls also must meet the requirements of the location. Ambient temperature and altitude also can be major considerations.

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Synchronous motors, requiring direct current field excitation, can provide a power factor of 1.0 and can have a leading power factor to offset lagging power factor of other equipment in the plant, reducing energy costs.

Full voltage starting normally is used for this type of compressor. Reduced voltage starting, where essential, must take into account the required starting torque.

Steam Turbine Driven Compressors

Steam engines have given way to steam turbines, where steam is used as an available and economical source of power. Generally, a speed reducing gear is required between a relatively high speed steam turbine and a relatively low speed double-acting reciprocating compressor. Suitable coupling and flywheel arrangements also are required. Steam turbine speed governors may be manual or automatic in response to changes compressed air system pressure and/or flow.

Performance Guarantees

Capacities are based upon prevailing ambient conditions, while power normally is stated in bhp at full load capacity and pressure. The capacity and bhp normally are guaranteed within 3 percent but not cumulative. This means that the specific power in bhp/cfm also is guaranteed within 3%, so that factors such as compression efficiency and/or mechanical efficiency are not significant for comparison of different compressors. The full load bhp and motor efficiency will allow specific power in kW/100 cfm for comparison with other types of packaged compressors. Table 2.1 is given as a typical example of compressor performance at full load, 75% and 50% capacity. ASME PTC-9 or ISO 1217 are common standards for performance tests. Members of the Compressed Air and Gas Institute now use a standardized Performance Data Sheet to show compressor performance based upon a standardized test.

Table 2.1 Data on Direct-connected, Motor-driven, Two-stage Compressor

Size of compressor cylinders, in		20 1/2 and 12 1/2 x 8 1/2	
Piston displacement, cfm (1st stage)		1662	
Rpm		514	
Discharge pressure, psig		100	
Altitude, ft above sea level		0	
	Full Capacity	3/4 Capacity	1/2 Capacity
Actual capacity, ft^3 free air per min.	1395	1046	697.5
Bhp at compressor shaft	260	200	143
Bhp per 100 cfm actual capacity	18.6	19.25	20.5
Motor efficiency, percent	93.2	92.3	90.2
Electrical hp input per 100 cfm actual capacity	19.95	20.85	22.77

INSTALLATION AND CARE OF DOUBLE-ACTING RECIPROCATING AIR COMPRESSORS

Manufacturers normally supply a manual for installation and operation of their compressors. It is recommended that in addition, ASME B19.1, Safety Standards for Air Compressor Systems, also be consulted.

Location

Proper location and installation will materially reduce maintenance and operating costs. The compressor should be located in a clean, well-lighted area of sufficient size to permit cleaning, ready inspection and any necessary dismantling, such as removal of pistons with rods, flywheels, belt sheaves, crankshafts, or intercooler tube bundles. The installation drawings furnished by the manufacturer show space required for major dismantling. The location should be such as to keep piping runs short, with a minimum of elbows to minimize pressure losses. Locate the compressor accessories, such as the aftercooler and air receiver, to permit short, straight runs of piping to minimize vibration caused by pressure pulsations from the compressor discharge.

In plants such as foundries and woodworking plants, where dusty conditions prevail, the compressor(s) should be located in a separate machinery room or dust-free room with provision for drawing clean air from outside the building to the compressor inlet. An air inlet filter should be installed and, for dusty atmospheres, select the heavy duty type. It should be remembered that this filter is to protect the compressor. Additional filtration downstream of the compressor may also be necessary to protect equipment at points of use.

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Foundation

A properly designed and constructed compressor foundation performs two functions:

1. It maintains compressor alignment and at proper elevation;
2. It minimizes vibration and prevents its transmission to any building structures external to the foundation.

The foundation must satisfy these requirements:

1. The foundation base area must be properly distributed so that the pressure it imposes on the supporting soil will not at any point exceed the safe bearing capacity for the particular soil encountered. The foundation will sink if this requirement is not satisfied (Figure 2.57A).
2. The base area should be so disposed that there is not too great a difference in unit loading on different equal areas of the supporting soil. If the unit loading varies too greatly under different parts of the foundation, the block probably will tilt (Figure 2.57B).
3. The foundation must have such proportions that the net result of the total vertical load and horizontal unbalanced reciprocating forces always falls within the base area of the foundation (Figure 2.57C). If the resultant falls outside, the foundation and compressor probably will topple or stand at an angle.
4. There must be enough weight in the foundation to prevent it from sliding on the supporting soil because of unbalanced forces (Figure 2.57D).

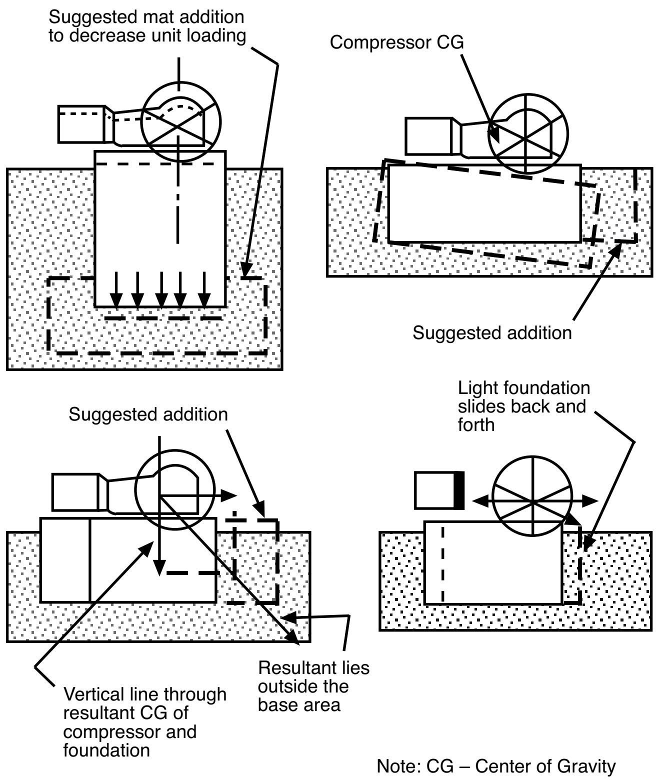

Figure 2.57A, B, C, D The foundation must distribute the weight evenly over an area sufficiently large that the machine does not settle.

To determine the required thickness of a concrete foundation, it is necessary to know:

1. Compressor weight and its center of gravity.
2. Magnitude, frequency, location, and direction of any unbalanced forces.
3. Quality of the subsoil.

Foundation Plans

Manufacturer's foundation plans, submitted with the compressor, are designed for good, hard, firm ground, such as well cemented sand and gravel or hard clay, always dry. The recommended foundation provides for enough thickness to support the compressor's weight plus disturbing forces, and for installation of sufficiently long foundation bolts to secure good anchorage. Carefully worded notations, appended to the plans, stress the importance of soil quality and urge investigation by a foundation expert if there is any question of the soil's ability to carry the load.

Subsoil Characteristics

To prevent objectionable vibrations and to eliminate very expensive correction after the compressor has been erected, it is highly important that the soil characteristics (static loading limits and elasticity) be known and understood. Soil quality may vary at different seasons. Soil may be wet in spring and early fall. These conditions must be considered carefully, because foundations have been known to move when the soil is wet, the movement completely disappearing after the soil has thoroughly dried. Soft clay, alluvial soils, loose sand and gravel, silt, and filled ground are poor supports for foundations of large reciprocating compressors. When such soils are found, the bearing area should be increased by placing the foundation on a mat, as shown in Figure 2.57A. With poor or wet soil, piling may be necessary to provide the necessary vertical support. It is usually advisable to also install batter piles (piles driven at an angle at the foundation ends) to absorb the horizontal unbalanced forces. Where possible, neighboring installations on similar soil should be observed to help determine the necessary precautions.

In many instances where substantial concrete depth is available in existing basement floors or where the footing is solid rock, the only requirement is to apply a surface covering of properly bonded concrete to support the compressor. There is little reason to excavate a perfectly sound structure to provide for a concrete block, such as may be indicated by the compressor manufacturer's foundation plan.

Regardless of past experience in a given area, it is recommended on any sizable installation that soil tests be made before proceeding with the foundation. Corrections on any foundation are extremely costly. The character of the subsoil should be ascertained by borings on the site of the foundation. Such borings should be at least four in number, one near each corner of the proposed foundation. It is highly recommended that the borings be made and judged by a competent foundation engineer, who should recommend suitable construction. Table 2.2 covers various soils encountered and gives the safe bearing capacity for static loads in tons per square foot. For dynamic loading, the allowable or design loads for a compressor foundation should not exceed one-quarter to one-sixth of the values given.

Table 2.2 Safe Static Load in Tons per Square Foot (For allowable dynamic loading use 1/4 - 1/6 of loadings shown.)

Type of Soil	Tons per sq ft
Solid ledge of hard rock, such as granite	25 to 100*
Sound shale and other medium rock, requiring blasting for removal	10 to 15
Hardpan, cemented sand and gravel, difficult to remove by picking	8 to 10
Soft rock in disintegrated ledge, or in natural ledge difficult to remove by picking	5 to 10
Compact sand and gravel, requiring picking for removal	5 to 6
Hard clay, requiring picking for removal	4 to 5
Gravel, coarse sand, in thick, natural beds	4 to 5
Loose medium or coarse sand; fine, dry sand	3 to 4
Medium clay, stiff but capable of being spaded	2 to 4
Fine, wet sand, confined2 to 3	
Soft clay	1

*It is ordinary not advisable to impose on any type of soil a unit load greater than 15 tons because the safe crushing strength of concrete (in the foundation) is usually taken at about 15 tons per sq. ft.

Subsoil Elasticity

For many years, design considerations for compressor foundations have dealt only with the static load problem. Recent experience has illustrated clearly that not enough emphasis has been placed on the elastic characteristics of sub-soils. In addition to the static load imposed on the subsoil by all structures, reciprocating machines also exert a dynamic loading on the foundation. To cope with the dynamic loading, installation and care of stationary reciprocating compressors it is necessary to consider the elastic characteristics of the ground on which the foundation rests. The following example illustrates this characteristic.

An oil storage tank was observed to settle when filled, and position when emptied. The test was made with a transit and repeated several times with the same results. There was a definite relation between load that is typical of elastic materials.

The foundation and the ground form an elastic system that will produce excessive vibrations if excited by periodic forces (the unbalanced forces in the reciprocating equipment). Having a frequency near the natural frequency of this bearing pressures keep the natural frequency of the foundation high, and also reduces the possibility of transmitted vibrations. A thorough treatment of this subject is beyond the scope of this book; therefore, it is recommended that a foundation engineer, familiar with the local area, be contacted. The compressor's unbalanced forces can be obtained from the manufacturer's drawings or by contacting the manufacturer.

Foundation Depth, Area, and Placement

Where possible, the foundation should be carried down to a firm footing. Where the foundation is exposed to freezing temperatures, its depth should extend below the frost line. Where the depth of the foundation is made greater than that shown on the manufacturer's drawing, the supporting area of the foundation should be proportionately increased by means of a mat. Such an increase in base area also is necessary whenever a compressor is raised above the floor line to an extent greater than that shown on the drawings. Where the sides of the foundation do not abut well-tamped soil, the base area must be increased. It also is advisable to isolate the foundation from any building footings, walls, or floors, to prevent any vibration being carried into the building structure. Where more than one compressor is being installed and they are relatively close to each other, it is advisable to cast their foundations en-bloc or separately on a common concrete mat.

Concrete Mix

It is recommended that a concrete engineer be consulted as to the proper mix for the location. In general, a foundation concrete mix of 3000 lb/in.^{2} compressive strength at 28 days of age and a slump not to exceed 6 in. should be suitable. Such concrete mix should be made with durable aggregates of such gradation that not over 40 gal total water, including moisture in sand and aggregates, be used per cubic yard of concrete. A cement dispersing agent may be used in the concrete mix. If aggregates are used that contain reactive silica, then cement must be used that contains less than 0.6% sodium and potassium alkalis.

The minimum amount of water described above should be adhered to. This should provide a ready placement and desired strength. No foundation should be poured when there is any possibility of freezing. For foundations, it is important that continuous pouring be used rather than section pouring.

The top surface of the foundation before the concrete takes its final set should be roughened by raking (preferably with a coarse rake) to provide an irregular surface for the bonding of the grout, which will be applied after the equipment is aligned and set on the foundation. The surfaces should be roughened to the extent that the aggregate is exposed and indentations in the surface are at least 2 in. deep and irregular. Figure 2.58 illustrates the proper surface texture.

The foundation should cure for at least a week before equipment is put on it. Curing can be accomplished by using a curing compound or by keeping the surface wet for seven days if normal temperatures exist or longer at lower-than-normal temperatures.

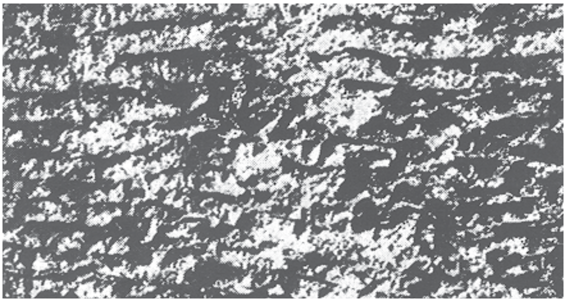

Figure 2.58 Proper surface to provide bonding of the grout.

Foundation Bolts

Foundation or anchor bolts serve two purposes: (1) they hold the compressor down firmly on the foundation, and (2) they prevent it sliding laterally on the foundation. The real function of the bolts then is to bind the compressor firmly so that, structurally, the compressor and foundation constitute a single mass.

Most manufacturers furnish a drawing that specifies anchor-bolt dimensions and their location in the foundation. Foundation bolts can be purchased from the manufacturer or can be made locally. In the event the bolts are obtained locally, the materials should not be of lower quality steel than AISI-C-1120, which has the following physical properties for hot-rolled material:

Tensile strength, 65,000 psi
Yield point, 38,000 psi
Elongation, 25%
Brinnel hardness, 117, approximate

Providing casings of ample length around the bolts permits lateral movement to compensate for slight inaccuracies in locating the bolts in the foundation and unavoidable shifts of the hole core during casting of the compressor frame and supports. Casings can be made of metal drain pipe, ordinary steel, or wrought-iron pipe. They should be at least 2 in. larger in diameter than the foundation bolts.

Foundation Template

The template (Figure 2.59) is a pattern or frame usually fabricated from wood strips. It supports the bolts and casings, while the foundation is poured around them. Engineering practice dictates using a template because it is more economical to build one than to endeavor to locate the bolts without it. It is best to build a template on the job. If the manufacturer furnished it, damage and warping during shipment might distort the framework and destroy its accuracy. Dimensions for building the template can be secured from the manufacturer's foundation drawing.

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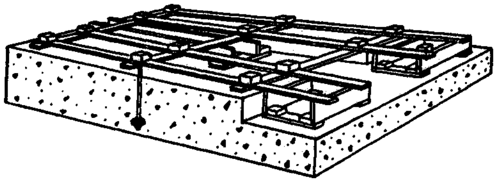

Figure 2.59 Foundation template fabricated from wood strips.

Alignment and Leveling

The compressor should be accurately aligned and leveled in accordance with the manufacturer's instructions. Before setting the compressor crankcase on the foundation, any glaze or loose scale from the exposed aggregate and any other particles not solidly bonded together must be removed with a star chisel. The surface must be swept and thoroughly cleaned to make sure that all loose particles are removed.

Leveling Wedges

Iron or steel is used in preference to wood for leveling wedges. This eliminates the possibility of swelling and thus disturbing the alignment. It is recommended that the wedges be removed after grouting because the grout may shrink away from contact with the bottom of the frame or base, thus allowing the wedges to support the whole load. When the wedges have been removed, the base must necessarily rest firmly on the grout.

Where the frame is provided with leveling set screws, wedges should not be used, but steel plates should be placed under the set screws. These plates can be left in place after the grouting has set, but the set screws must be backed off.

Grouting

Cement-type grout. The top of the foundation should be cleared, brushing and washing it off to remove any loose particles and checked to make sure that no tools or equipment used in leveling the compressor have been left.

A dam or form should be built around the foundation high enough to make a grout level inside the hollowed-out portion of the frame at least 1 in. above the bottom flange face. Forms should also be built around supports and motor sole plates or base plates. The foundation should be wetted thoroughly at least 30 minutes before the grout is poured and kept wet to ensure a good bond between the foundation and grout. The entire unit should then be rechecked for alignment and prepared for grouting, using a grout recommended by either a competent concrete engineer or by the compressor manufacturer.

The grout is poured, preferably from one side of the frame, to all points within the form. The grout should be spread evenly and worked beneath the frame flange to eliminate any tendency to form air pockets. If there is any danger of freezing, grouting should be postponed. After the grout is completely hard, all wedges and shims are removed and leveling screws retracted, if used. Voids left by the wedges are then filled with grout. At all juncture points between the compressor unit and grout, it is advisable to apply several coats of shellac, followed by paint, to prevent any seepage of oil into these points. In fact, the whole foundation should be similarly treated, as oil has a very harmful effect on the concrete or grout.

Plastic-type grout. The basic advantages of plastic-base, thermosetting epoxy resin and an inert filler type of grout, are oil resistance, higher tensile strength, very low shrinkage, fast hardening, and high bonding strength to metal and to rough concrete. It has chemical resistance and properties of impact strength.

The concrete surfaces must be dry and clean. Metal surfaces, where, a bond is desired, must be cleaned by sandblasting or with solvents, and all dust must be removed. Cleaned metal surfaces can be coated with a bonding film, as recommended by the manufacturers of plastic grouts.

Areas where bonding is not desired, such as forms or water hoses used for scaling around the oil pan, must be waxed. Specific instructions on the use of plastic grout must be requested from the equipment or plastic grout manufacturers or both. Because of its low viscosity, particular attention must be paid to scaling of forms and all possible seepage passages.

Air Intake

A clean, cool, dry air supply is essential to the satisfactory operation of a compressor. Wherever possible, the compressor inlet should be taken from the outside air. The open end of the intake pipe must be well hooded and screened to prevent rain and dirt or dust from entering. The filter should take air from at least 6 ft. or more from the ground or roof and should be located several feet away from any wall to minimize the pulsating effect on the structure. The pulsations immediately surrounding the intake may rattle windows and disintegrate a weakly constructed building wall.

It is recommended that the compressor intake not be located in an enclosed courtyard. In such a yard, the air compressor intake could cause pressure pulsations that would cause building vibrations even in solidly constructed buildings.

The air intake must always be located far enough from steam, gas, or oil engine exhaust pipes to ensure that the air will he free from dust, dirt, moisture, and contamination by exhaust gases.

The intake piping should be as short and direct as possible with long-radius elbows where bends are necessary. It should be the full diameter of the intake opening of the compressor. If the intake pipe is extremely long, a larger size should be used.

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If the air intake pipe is above the floor, aluminum, plastic galvanized pipe, standard steel pipe, or sheet-metal pipe can be used.

Glazed vitrified pipe and reinforced polyvinyl are convenient materials to use for underground air intakes, as long sweep elbows of the same material can be obtained. All joints should be cemented to make them watertight, as any water seeping into the intake is carried into the compressor, washing away the lubricant and causing the piston and cylinder to cut or wear. If a concrete duct is built for use as an air intake, it must have a smooth, hard interior surface, for if the concrete crumbles or disintegrates due to the air rushing through, the ingredients are carried into the compressor cylinder, causing rapid wear of the valve and piston, as well as possible scoring of the compressor cylinder. Glazed, vitrified pipe is preferred. Painting the interior of a concrete duct, if used, with a high-grade special waterproof paint or epoxy coating is advisable.

Discharge lines, steam lines, hot water lines, and the like, must never be put in the intake duct as such practice will raise the temperature of the intake air and cause considerable loss in the volume flow of the unit. A fact that must be remembered in this connection is that for every 5ºF reduction of the temperature of the intake air there is a gain of approximately 1 percent in air weight. Table 2.3 shows the effect of intake or initial temperature on the delivery of air compressors.

Where a compressor is used in or adjacent to a chemical plant, the air drawn into the compressor may contain acid fumes that attack iron and steel, causing corrosion and wear of the valves, pistons, and cylinders. Similarly, the exhaust from other industries may contain contaminants that will be injurious to the compressor. If these conditions are known to exist, the manufacturer of the compressor should be informed when the machine is purchased so that the proper precautions may be taken. Every effort should be made to locate the intake away from such fumes or other contaminants.

The suction line to the compressor should be thoroughly cleaned before the machine is first started to remove accumulation of pipe scale and grit or other foreign objects inadvertently placed in the line during the installation, It is recommended that the piping be fabricated with a sufficient number of flanged joints so that it can be dismantled easily for cleaning and testing. It is far better to clean and test piping in sections before actual erection than after it is in place. The use of chill rings for butt welds in piping is recommended. This prevents welding beads getting into the pipe and being carried through, not only on the original startup, but later during operation.

Table 2.3 Effect of Initial or Intake Temperature on Delivery of Air Compressors Based on a Normal Intake Temperature of 60ºF

Initial Temperatures			Initial Temperatures		
ºF	ºF abs.	Relative Delivery	ºF	ºF abs.	Relative Delivery
-20	440	1.18	70	530	0.980
-10	450	1.155	80	540	0.961
0	460	1.13	90	550	.0944
10	470	1.104	100	560	0.928
20	480	1.083	110	570	0.912
30	490	1.061	120	580	0.896
32	492	1.058	130	590	0.880
40	500	1.040	140	600	0.866
50	510	1.020	150	610	0.852
60	520	1.00	160	620	0.838

Depending on the material used and its condition, the cleaning can be accomplished by one of several methods or a combination of methods. These include wire brushing and blowing out, hammering and blowing out, sandblasting and blowing out, wiping with lintless cloth, washing or flushing, pickling and washing, or other methods practical at the site to ensure a clean suction line.

If it is impossible to mount the filter immediately adjacent to the compressor and if other than a short straight intake is to be used, it is recommended that a consulting engineer be contacted to make sure that the intake sizing and configuration will not introduce objectionable pulsation and excessive pressure drop.

Discharge Piping

The discharge pipe should be the full size of the compressor outlet or larger, and it should run directly to an aftercooler, if one is used. If no aftercooler is used, the discharge pipe should run directly to the receiver, the latter to be set outdoors if possible but kept as close to the compressor as practical. The discharge pipe should be as short and direct as possible, with a minimum of fittings and with long radius elbows where bends are necessary. Unnecessary pockets should be avoided. If a pocket is formed between the compressor and the aftercooler or receiver, it should be provided with a drain valve or automatic trap to avoid accumulation of oil and water moisture in the pipe itself.

The piping should be sloped away from the compressor with sufficient pitch to prevent either condensate or oil draining back into the compressor. A drop leg with a drain valve or automatic trap is a good idea at the compressor discharge to positively prevent liquids reaching the compressor.

The use of plug valves should be considered for the discharge line because these do not have pockets found in gate and globe valves. With outdoor installations in severe cold weather, this could eliminate freezing and breaking of valves.

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The hot discharge line should not contact wood or other flammable materials. Any gaskets in the discharge piping should be of asbestos, if permitted, or other oil-proof, noncombustible material. If the discharge line is more than 100 ft. long, pipe of the next larger diameter should be used throughout.

Under certain conditions of installation and operation, pipeline surges or pulsations may be set up in intake or discharge lines; these pulsations not only may cause vibrations of the pipe if it is not well anchored and supported, but may also influence the performance of the compressor.

A frequent cause of pressure surge in the discharge line is a rather long line with a receiver located at considerable distance from the compressor. The surge may be avoided by installing as near to the compressor as possible a surge drum of suitable size, which will damp out the vibrations. To isolate compressor vibrations from the system, it may be desirable to make the connection to the system by means of a short length of flexible hose. When in doubt, a competent engineer should be consulted who is familiar with the handling and piping of compressed air and gases.

Pipelines through which hot air passes should be kept clean to avoid the danger of a fire starting in the accumulated dirt and oil. It is recommended that a removable portion of the discharge pipe be installed directly out of the compressor so that this section can be readily removed when necessary, inspected, and cleaned of any buildup of carbon. Piping should drain toward the aftercooler and receiver.

All piping connected to the compressor should be arranged with flange fittings or unions close to the compressor to permit removal of the cylinder at any time without disturbing the piping. All overhead piping must be well supported to relieve the compressor of any incidental strains.

Cautions

A globe or gate valve may be placed in the discharge line between the compressor and aftercooler or between the compressor and the receiver when more than one compressor discharges into a single aftercooler or receiver. When such a shutoff valve is used, a safety valve of proper size positively must be placed in the line between the compressor and the shutoff valve and must be checked periodically. This is very important, since the compressor may at some time be started with the stop valve closed, and if no safety valve is used, sufficient pressure may build up to burst the cylinder. Figure 2.60 shows the wrong way and the right way to do this, if a stop valve must be employed. The safety valve or valves should have a total capacity more than sufficient to handle the entire output of the compressor. The globe valve on the safety valve branch is to allow for manual relief of pressure in the cylinder before opening it for inspection or repair.

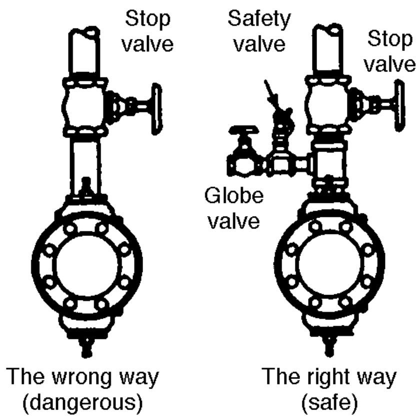

Figure 2.60 The right way and the wrong way to install a stop valve.

Figure 2.61 indicates in a general way how the discharge of a compressor should be connected to an aftercooler, if one is used, and to the receiver; the receiver inlet should be near the top of the tank and the discharge near the bottom. The arrangement of air piping for large compressors is, in general, the same as shown here.

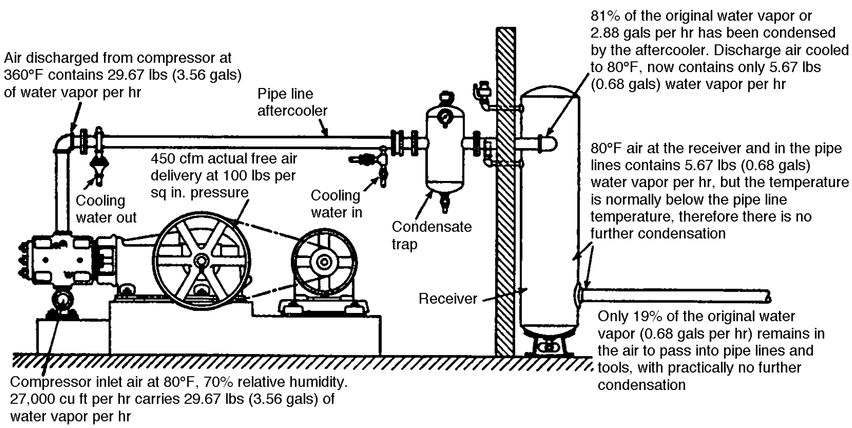

Figure 2.61 Compressor arrangement showing the piping to a receiver with an after-cooler.

A bypass valve or unloading valve should be provided in the discharge line to help in the starting of the compressor on occasions when receiver pressure is atmospheric or below the pressure necessary to permit the unloaders to function properly.

Circulating Water

A liberal supply of cooling water for cylinder jackets, cylinder heads, intercoolers, and aftercoolers must be provided. The operator should wait long enough after turning on the water before starting the compressor to ensure that water jackets are completely filled and the flow of water is established from the compressor. The use of dirty or scale-depositing water should be avoided, as it clogs the water passages and will reduce the cooling efficiency and result in considerable shutdown time to clean the jackets and maintain the overall efficiency of the compressor.

The inlet water connection should be located at the lowest point of the cylinder so that water can be easily drained from the cylinder when the compressor is shut down. The discharge connection should be at the highest point to ensure complete filling of the water jackets with no air pockets. The water piping should be provided with a valve for controlling the flow of water. The water flow control valve should be on the inlet water side to avoid water pressure on the cylinder when the unit is shut down.

When cooling water is very cold, condensation may form in the air inlet passage of the high pressure cylinder as the air enters from the intercooler, because the air may be much warmer than the water. Similarly, condensation may form in the first-stage cylinder of a multi-stage unit or in the cylinders of a single-stage unit handling saturated air if condensation forms. It will be carried into the cylinder and will destroy the lubricant, causing rapid cylinder and valve wear. To relieve this condition, it is advisable to pass the cold water through the intercooler first. This heats the water up considerably and allows a more normal relationship between water temperature and incoming gas temperature in the cylinder. The inlet water temperature to the cylinder jackets should never be less than the incoming gas temperature and, in general, should be 10 to 15ºF above the incoming gas temperature. Except in very special cases, water temperature should never be higher than 160ºF, with 120ºF being a preferred maximum.

Under ordinary circumstances, the cooling water is piped first to the intercooler and aftercooler in parallel. The intercooler saves power by reducing the volume of air handled by the high pressure stage, and the aftercooler moisture separator removes moisture and prevents carryover into the lines. Usually it is advisable, therefore, to supply each of them with the coldest water. From the intercooler, the water is then taken through the low and high pressure cylinder jackets.

If the jacket water flow is regulated automatically by thermostatic valves, the valves should be equipped with a bypass arranged so that at no time will the flow through the cylinder be completely stopped. This bypass should be designed to provide for enough circulated water to eliminate the formation of air pockets and hot spots and to make sure that the jackets are full of water at all times.

All jacket and cooler drains must be opened in freezing weather when shutting down the compressor.

Outlet water should flow into open funnels, allowing frequent temperature readings to be made; an excessive rise in temperature indicates insufficient water, carbonized discharge valves, leaking piston rings, or broken valve parts.

When closed water systems are used, sight flow indicators should be put on the water discharge from each cylinder and intercooler to show positively that water is circulating, and the discharge pipe should be bypassed to an open funnel so that the cylinders and coolers can be tested frequently to detect leaks in the water jacket or in the intercooler tubes. If the water discharge is opened to the overflow funnel and the valve on the discharge line shut off, any leakage between the air and the water spaces will be revealed by air blowing out with the water. Figure 2.62 shows how the water piping should always be arranged when a closed system is used. A relief valve should always be installed on a closed water system and should be between the shut-off valve and the cylinder jackets so as to prevent excessive pressure from building up in the jackets. The compressor manufacturer should be consulted for the proper setting of the water jacket relief valve.

The thermosyphon cooling system normally is used only in the case of portable or temporary installations where a continuous water supply is not available.

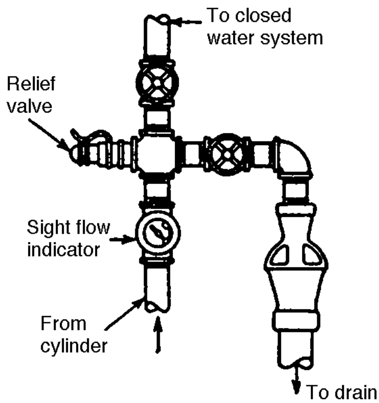

Figure 2.62 Arrangement of air-cylinder water piping when a closed system is used.

Table 2.4 shows the cooling water quantities recommended for the water coolers, cylinder jackets, and aftercoolers. It is expected that the temperature of the air leaving the intercooler or aftercooler will be within 10 to 15 °F, respectively, of the temperature of the water entering the cooler for ordinary working conditions.

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Table 2.4 Summary of Industrial Plant Compressed Air Systems (including ratio of air consumption to area of plant).

Type of Plant	Manufacturing Area (ft²)	Compressor Capacity (scfm at psig)		Horsepower	CFM per 1000 ft²
Automotive hardware manufacturing	580,000	9,600	110	1800	17
Laminated glass manufacturing	1,200,000	10,000	100	-------	8.3
Automobile component manufacturing	580,000	6,024	100	-------	10.5
Electrical switchgear manufacturing	252,000	500	100	250	2
Electrical switchgear manufacturing	135,000	400	100	100	3
Electronic computer manufacturing	750,000	1,525	100	300	2
Standard large electric manufacturing company	Any size	-------	100	-------	7
Glass bottle manufacturing, using automatic glass-blowing equipment	129,000	5,720 330*	50 100	800 15	45 2.6
Glass stemware manufacturing, using automatic glass-blowing equipment	234,000	7,160 1,400	50 100	500 250	30 6
Automated foundry	74,000	2,400	125	450	33

* Boosted from 50-psi system

DYNAMIC COMPRESSORS

Dynamic-type compressors are machines in which air is compressed by the mechanical action of rotating impellers imparting velocity and pressure to the air. The centrifugal type discussed in this chapter has a flow classified as being in the radial direction. For further information on dynamic compressors, including types, definitions, and applications, the reader is referred to Chapter 7.

Continued demand in the 1960s for more efficient compression with lower operating costs resulted in a transition to cooling between compressor stages. This was true for many centrifugal compressors used for plant air and other applications. The reason for this change is shown in Fig. 2.63. The figure shows the relative effect of additional coolers for a given compression ratio. The centrifugal compressor assumed is for a typical plant air, 100 psig application.

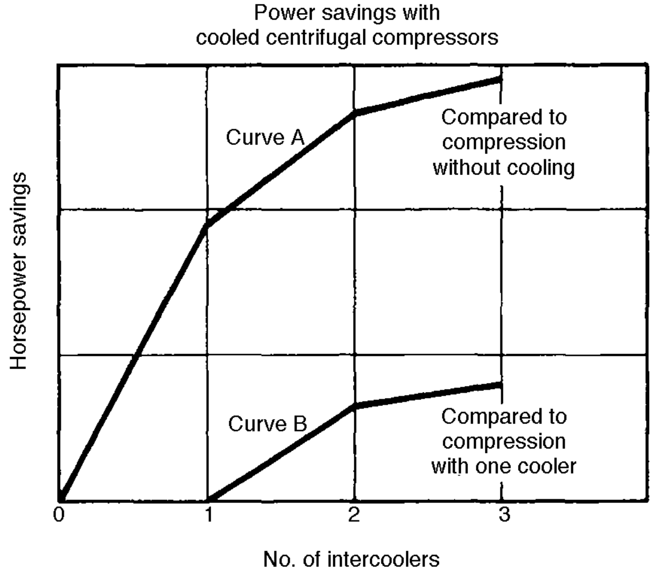

Figure 2.63 Relative power savings for a compressor with coolers compared with a compressor having no cooler and with one having one cooler.

The rising cost of energy in the 1970s resulted in a further drive to reduce the power requirements and led to a considerable number of arrangements, extending from one- to three-cooler compressor designs for plant air to three- or four-cooler compressor designs for 350 psig soot-blower applications in coal-fired boilers. Impeller and stage designs were improved aerodynamically to provide maximum air supply for minimum operating costs in standard product lines.

Typical of such intercooled compressors is the integral-gear-type centrifugal compressor seen in Fig. 2.64. It consists of a low-speed gear directly connected to the motor drive and two high-speed pinions having extended shafts that carry four centrifugal compressor impellers. Two shaft speeds are used to provide selection of more optimal impeller speeds and, consequently, improved efficiency. The use of axial entry to each centrifugal compressor stage and the adaptation of matching single-stage centrifugal compressor scrolls surrounding each impeller provided an ideal flow pattern and an efficient conversion of the velocity head leaving the impellers.

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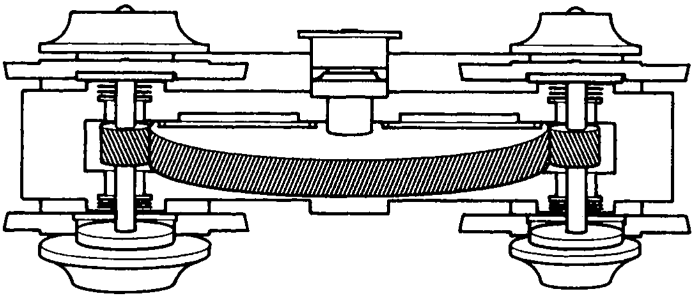

Figure 2.64 The input full gear of atypical integral-gear unit runs at motor speed. The low-speed pinion drives the first-stage impeller (upper right) and the second-stage impeller (lower right). The high-speed pinion drives the third-stage impeller (upper left) and the fourth-stage impeller at higher speed to match the reduced volume flow due to pressure increase and cooling.

Further input power reduction is obtained in the compression cycle by inter-cooling between the four stages of compression. Figure 2.65 shows the flow path and the cooling between stages. The open impeller design generally is used for air compressors up to and over 100,000 cfm. This impeller design is seen in Figure 2.66. Closed or shrouded impellers are used for very large flow volumes.

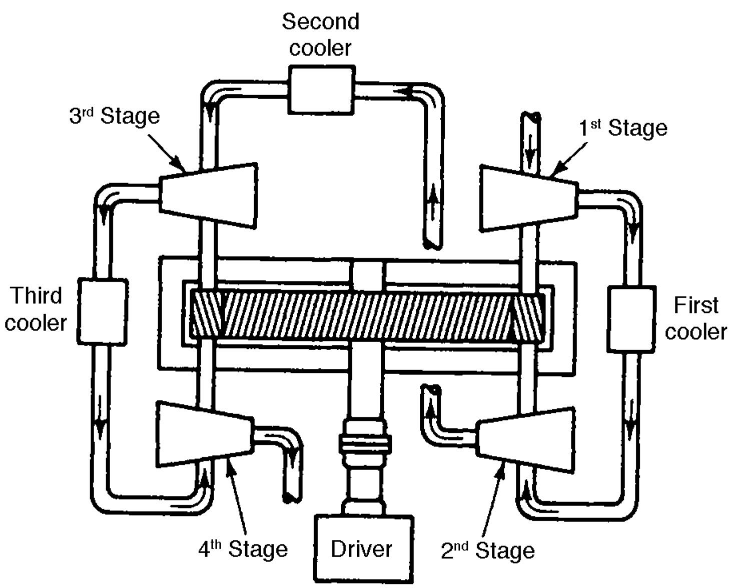

Figure 2.65 Flow diagram of an integral-gear-type compressor showing stages of compression and including the cooling arrangement.

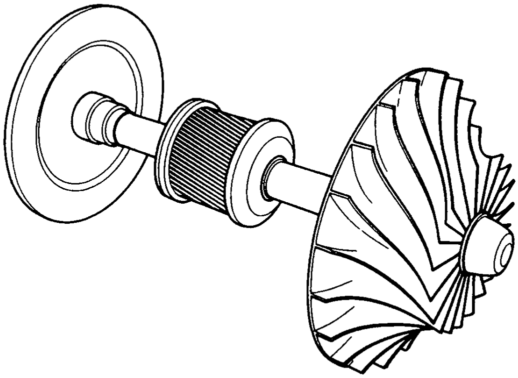

Figure 2.66 Pinion of an integral-gear unit having open, backward-curve-bladed impellers.

Opposed impellers on each pinion shaft (Fig. 2.67) help to balance the aerodynamic thrust load. The unit is equipped with special radial load bearings, which provide stability for the lightweight, high-speed pinions. An integral thrust bearing or thrust transfer ring on the pinion shaft absorbs the remaining net thrust between impellers. The low-speed gear shaft has full sleeve bearings and an integral-type thrust bearing. The gears shown are precision single helical type.

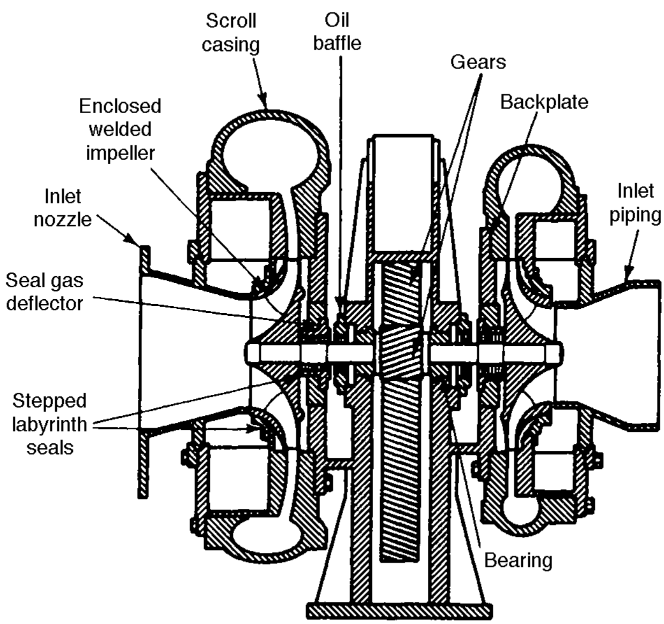

Figure 2.67 Opposed impellers help balance thrust on pinions.

COMPRESSED AIR USES

The low-speed gear shaft has full sleeve or anti-friction bearings and an integral-type thrust bearing. Principle uses are for the operation of hand tools, actuation of control devices, textile weaving, air separation, cleaning, and prevention of contamination and to power many motions required in automatic and semiautomatic production equipment. The design of the compressed-air system of a plant depends on the use to which the compressed air is to be put.

How Many Compressors? How Large?

Due consideration should be given to the need to have standby or future capacity when establishing the size and number of compressors. Equipment may be sized for less than the peak demand, provided the peak will be of short enough duration that the receiver and compressor together can carry the peak without compromising the pressure available at the points of use, and providing further that the valleys are deep and broad enough so that the receiver pressure can be restored before the next peak.

Where the size of compressors is great enough that such a feature is available, capacity modulation should be specified. For sizes at which capacity modulation is not available, the compressor capacity should be adequate to permit the unit to carry the anticipated eight-hour operating load with not over six hours of operation. For such installations, the capacity of the receiver should be such as to assure at least 2 minutes of operation of the compressor between the pressure control cut-in and cut-out pressures, with zero draw from the receiver. Table 2.4 shows typical data for the capacity required in cubic feet per minute per 1000 ft^2 of plant area for a variety of facilities.

Some attempt has been made to indicate in this table the general nature of the facility and the principal use made of compressed air therein. It will be noted that a relatively broad range of capacity is shown, even for closely similar plants. This indicates that the table should be used primarily as a check on the computed size or to establish approximate plant capacity if data are not available for more accurate computation.

Sizing Considerations

Centrifugal air compressors normally are specified on the basis of required air flow volume. However, there are several ways to calculate volume, and serious problems can result unless both user and manufacturer use the same method. At the very least, the user can have trouble comparing bids from competing manufacturers. At worst, he may choose the wrong compressor.

These problems can be avoided by specifying capacity in terms of the actual inlet conditions and by understanding how compressor capacity is affected by variable ambient conditions, such as inlet pressure, temperature, and relative humidity.

Also, factors such as cooling-water temperature and motor load must be considered before a compressor and its drive motor can be sized.

The accompanying data sheet shows tabulated factors that must be considered when choosing a centrifugal air compressor. Some factors are fixed; others vary fairly frequently, daily, or seasonally.

Compressed Air and Gas Institute
Data Sheets
for
Packaged, Integrally Geared Centrifugal
Compressors
Plant-Process-Instrument

1.0	Customer site considerations:			
1.0.1	Barometer (psia)			*______
1.0.2	Inlet pressure at first stage flange, psia			______
1.0.3	Inlet temperature, °F			*______
1.0.4	Relative humidity, %			*______
1.0.5	Cooling-water temperature, °F			*______
1.0.6	Cooling-water pressure, psig			*______
1.0.7	Source of cooling water (i.e., tower city, etc.)			*______
1.0.8	Outlet pressure from aftercooler psig/psia (if supplied)			*______
2.0	Vendor model designation			______
3.0	Number of stages			______
4.0	SCFM at 14.5 psia, 68°F, 0 percent RH (dry):			
4.0.1	Required by purchaser			*______
4.0.2	Vendor's rating			______
5.0	Weight flow:			
5.0.1	Inlet weight flow, lb/min dry, based on SCFM at 14.5 psia, 68°F, 0 percent RH (dry)		For 4.0.1 ______	For 4.0.2 ______
5.0.2	Specific humidity (vapor content) at site conditions, lb water vapor/lb dry air		______	______
5.0.3	Inlet total weight flow wet (5.0.1 x 5.0.2 + 5.0.1 = 5.0.3)		______	______
6.0	ICFM at inlet flange			______
7.0	Horsepower at compressor coupling:			
7.0.1	For 4.0.1			______
7.0.2	For 4.0.2			______
8.0	Cooling water required and pressure drop:		GPM	Pressure drop, psi
8.0.1	Intercoolers, total		______	______
8.0.2	Aftercooler		______	______

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				Type
8.0.3		Oil cooler	______	______
	9.0	Bearings:		
9.0.1		Bull gear bearings:		
9.0.1.1		Journal		______
9.0.1.2		Thrust		______
9.0.2		Pinion bearings:		
9.0.2.1		Journal		______
9.0.2.2		Thrust		______
10.0	Drive Coupling:			
10.0.1		Make		______
10.0.2		Type		______
11.0	Impeller shaft seal:			
11.0.1		Air seal		______
11.0.2		Oil seal		______
			Pressure	Flow
11.0.3		Air seal requirements	______	______
11.0.4		Oil seal requirements	______	______
12.0	Impellers:			
12.0.1		Type, backward leaning or radial blades		______
12.0.2		Material		______
13.0	Compressor casings:			
13.0.1		Type, horizontal or vertical split		______
14.0	Gear case:			
14.0.1		Type, horizontal or vertical split		______
14.0.2		Material		
15.0	Pinions:			
15.0.1		Material		______
15.0.2		AGMA quality number		______
16.0	Bull gear:			
16.0.1		Material		______
16.0.2		AGMA quality number		______
			Pressure	Flow
17.0	Oil reservoir retention time		______	______

The main item that must be specified is inlet air volume. Standard cubic feet per minute (scfm) is a common unit of measure for compressor capacity in the United States; however, several definitions of this unit exist. The definition of Standard Air adopted by the Compressed Air & Gas Institute, PNEUROP and ISO, is air at 14.5 psia, 68˚F and 0% relative humidity. The petrochemical industry commonly uses 14.7 psia, 60˚F and 0% relative humidity. A third (metric) definition specifies a standard(or normal) cubic meter of air at 1 atmosphere, 32˚F, and 0 percent relative humidity.

A packaged integrally geared, multistage, centrifugal compressor typically includes the following:

1. Multistage centrifugal compressor
2. Prime mover (motor or turbine)
3. Coupling and guard
4. Lube oil system including:
   a. Single oil cooler
   b. Single oil filter
   c. Auxiliary oil pump (full capacity)
5. Intercoolers
6. Aftercooler-moisture separator
7. Vibration monitoring system
8. Controls and instrumentation
9. Control panel
10. Inlet filter-silencer
11. Discharge check valve
12. Inlet valve
13. Blowoff silencer
14. Discharge blowoff valve-antisurge valve
15. Base plate

To avoid the confusion caused by these variable standards, some users have adopted a simpler unit that expresses inlet volume in terms of the actual inlet pressure, temperature, and humidity. This inlet cubic feet per minute (icfm) indicates the actual volume entering the first stage of a multistage compressor at the expected operating conditions. This volume, in turn, determines the impeller design, nozzle diameter, and casing size that provides the most efficient operation.

The relationship between icfm and equivalent scfm can be expressed as:

$$Q_s = Q_i \frac{T_s}{T_i} \frac{P_i - P_v}{P_s}$$

where:

Q_i = inlet air volume, icfm
Q_s = standard air volume, scfm
T_i = inlet temperature, degrees Rankin
T_s = standard temperature, ºR
P_i = inlet pressure, psia
P_s = standard barometric pressure, psia
P_v = partial vapor pressure, psia

This last term is equivalent to the saturated steam pressure at temperature T_i multiplied by the relative humidity. This equation indicates how ambient air pressure, inlet temperature, and relative humidity affect capacity.

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Inlet pressure is determined by taking the barometric pressure and subtracting a reasonable loss for the inlet air filter and piping. A typical value for filter and piping loss is 0.3 psig.

The need to determine inlet pressure at the compressor flange accurately is particularly critical in high-altitude installations. Because barometric pressure varies with altitude, a change in altitude of more than a few hundred feet can greatly reduce compressor capacity. Often, the lost capacity can be restored by using larger-diameter impellers, but occasionally a different-sized compressor must be used.

Other variables that influence volume flow include temperature and relative humidity of the inlet air. These must be considered over the range of conditions expected in service. Air volume is lowest at the highest expected operating temperature, and vice versa. Therefore, the impellers must be designed to deliver the required flow at the highest temperature expected. This guideline also applies to the temperature of the cooling water, which controls the temperature of the air delivered to the stage following an intercooler.

Relative humidity also affects the useful volume of air available at the compressor inlet. The higher the humidity, the less is the effective air volume available; thus the impellers must be sized for the highest humidity expected.

A typical multistage centrifugal compressor for plant air service compresses air in several stages, with intercooling between each pair of stages. The relationship of pressure versus volume flow of a typical compressor is such that the pressure decreases at an increasing rate as volume flow rate increases.

Compressed air is often used in some sort of pneumatic device or is involved in chemical-processing operations. When it is used in a machine to do work, the amount of work done depends on the mass flow of air passing through the device. Mass is also a common denominator in cases in which it is involved in a chemical reaction and becomes part of the product. For the chemical equation to balance, a specific mass of product requires a specific mass of air. Therefore, in the final analysis, the mass flow (weight flow) of air delivered by a compressor should be the fundamental factor in specifying its capacity.

Weight Flow of Air Delivered

The key word is delivered, that is, air available for use at the discharge flange of the compressor. The work that can be done is based on what comes out of the compressor. Given a properly defined specification, the manufacturer is responsible for making sure that the compressor takes in enough air to make up for seal losses and the like so that the required weight flow is available at the discharge.

Importance of Air Density and Volume Flow

Volume flow does not tell very much because the weight of air in each cubic foot depends on the temperature and pressure of the air. In other words, the weight flow is related to the density of the air as well as the volume flow. The following formula relates weight flow to volume flow:

$$W = Q\rho$$

where:

W = weight flow, lb/min
Q = volume flow at the given air density, cfm
ρ = weight density, lb/ft^3

Air density (weight of a cubic foot of air) is inversely proportional to its absolute temperature. Thus, the higher the temperature, the less weight flow in each cubic foot. The weight flow delivered in summer is less than in winter. Therefore, the specification for a compressor should provide for the required weight flow to be delivered on a hot summer day. A slightly larger compressor will be required if the air temperature is 90˚F rather than 68˚F. If the manufacturer's rating (based on air at 68˚F) is accepted, the compressor will not deliver the same weight of air per minute at 90˚F.

Another reason that the volume flow by itself must be qualified is that air density also depends directly on air pressure. Because atmospheric air pressure depends on altitude, a compressor installed at a higher elevation (above sea level) gets less weight of air in each cubic foot of intake air than the same compressor installed at sea level. This change in weight flow due to differences in barometric pressure can be significant. For example, because of the lower atmospheric pressure, a compressor in Kansas City will deliver nearly 5% less air than the same compressor installed in Miami.

For dry air, the relationship of density to temperature and pressure is:

$$\rho = \frac{144P}{RT}$$

where:

P = absolute air pressure, psia
R = gas constant of dry air
T = absolute air temperature, °R

However, barometric pressure is not the only factor that affects inlet pressure. The effects of air filter, inlet valve, and piping leading to the compressor should also be considered. Because these components can cause significant pressure drop, a cubic foot of air measured just ahead of the compressor flange will contain less air by weight than a cubic foot measured ahead of the filter. Then the density of air entering a compressor becomes:

$$\rho = \frac{144(p_b - \Delta p)}{R_m T}$$

where:

p_b	=	absolute barometric air pressure, psia
Δp	=	pressure loss in inlet air filter, piping, and inlet valves, psia
R_m	=	gas constant of air mixture (i.e. with water vapor)

Relative Humidity Is Important

Another variable that often causes confusion in sizing an air compressor is relative humidity. Atmospheric air always contains water vapor. As a result, the compressor takes in a mixture of air and water vapor. This affects compressor operation and performance because the higher the pressure of the air, the less water vapor it can hold. And what it can no longer hold condenses in the intercoolers and aftercooler and is drained as water. So, once again, the weight of the cubic foot in is not the same as the weight out. The compressor must be sized slightly larger to allow for the water vapor loss, which, although it is part of the inlet flow, is not part of the delivered weight flow.

A portion of the inlet volume is attributable to water vapor. This depends on the relative humidity of the intake air and can be calculated by:

$$E = \frac{P_b}{P_b - rh(p_{vs})} - 1$$

where:

E	=	factor representing the amount of increase in inlet volume due to water vapor
rh	=	relative humidity, percent
p_{vs}	=	vapor pressure at saturation at given air temperature, psia (from steam tables; selected data are given in Table 2.5)

Table 2.5 Pressure of Water Vapor at Saturation

Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)	Temperature (°F)	Pressure (psia)
32	0.08854	60	0.2563	86	0.6152
34	0.09603	62	0.2751	88	0.6556
36	0.10401	64	0.2951	90	0.6982
38	0.11256	66	0.3164	92	0.7432
40	0.12170	68	0.3390	94	0.7906
42	0.13150	70	0.3631	95	0.8153
44	0.14199	72	0.3886	96	0.8407
46	0.15323	74	0.4156	98	0.8935
48	0.16525	76	0.4443	100	0.9492
50	0.17811	78	0.4747	102	1.0078
52	0.19182	80	0.5069	104	1.0695
54	0.20642	82	0.5410	106	1.1345
56	0.2220	84	0.5771	108	1.2029
58	0.2386	85	0.5961	110	1.2748

These relationships provide all that is needed to relate the real need, that is, weight flow of dry air to a quantity (volume flow), which is generally used by compressor manufacturers for performance rating purposes.

Many relationships become apparent from the following formula:

$$W = 144Q\frac{P_b - \Delta P}{R_m T(1+E)}$$

where:

P = pounds per square foot
= $144p$

Specifying Ambient Conditions

Another point that can be inferred from the last equation for Q, is that in order to be sure to have enough air, the compressor buyer must be careful in specifying ambient conditions. These should tend toward the minimal conditions, that is, high air temperature, normal barometric pressure, and high humidity. This does not mean that the specification should be based on the maximum air temperature on record. The result would be an unnecessarily large compressor.

For example, an air-conditioning guide gives a 1 percent confidence limit on a summer temperature of 95˚F in Chicago. This means there is only one chance in 100 that a 95˚F temperature will be exceeded, and not much more chance that even 90˚F will often be exceeded by a significant amount. On the other hand, specifying a lower air temperature such as 60˚F will result in running short of air on the days when this temperature is exceeded.

A related item to be considered in selecting a compressor of the correct capacity is cooling-water temperature, since most air compressors are intercooled. Water temperature has much the same effect as air temperature. This is easy to understand because the water cools the air before it enters the next compression stage. The warmer the cooling water, the warmer the air, and the less dense it will be. Therefore, water temperature should be specified at the highest anticipated temperature; otherwise, the compressor will deliver less air than expected. When cooling-water temperature is lower than specified, water flow can be reduced.

Centrifugal Air Compressor Characteristics

Centrifugal air compressor performance can be represented by a characteristic curve of discharge pressure versus flow. This is a continuously rising curve from right to left (Fig. 2.68). The effect of environment on performance requires understanding of two phenomena associated with this curve: choke (stonewall) and surge.

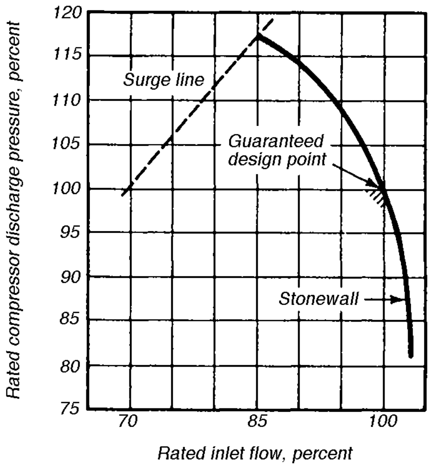

Figure 2.68 Typical Centrifugal Compressor Performance Curve

When the compressed-air system pressure decreases, a centrifugal compressor delivers an increased volume of air. As the system pressure continues to decrease, the air delivery from the compressor continues to increase until the air velocity somewhere in the compressor reaches the speed of sound. At this point, the flow is said to be choked because further reduction in system pressure does not result in additional air delivery by the compressor.

On the other hand, the maximum discharge pressure of the centrifugal compressor is a function of the intersection of the surge line and the sloping performance curve. When the compressed-air system pressure increases, the compressor furnishes less air as higher pressures are encountered until the system resistance is matched. This relationship may continue until the compressor is unable to maintain a steady flow of air into the system.

When the compressor cannot maintain a steady flow of air, backflow from the system through the compressor occurs until a momentary equilibrium is established between the compressor and the system. This backflow is commonly referred to as surge. This phenomenon is roughly equivalent to the stalled condition of an airfoil. Under this condition, compressor operation moves from surge to some point below the operating point shown on the performance curve. When the compressor continues to operate against sustained excessive system pressure, compressor operation moves up the curve and surge occurs again.

Neither of these conditions is desirable and both should be avoided. Control systems that allow the compressor to function without reaching the choked or surge condition must be based on prevailing environmental conditions. Therefore, it is helpful to examine individual environmental factors that can affect compressor performance. Further detailed information on compressor performance can be found in Chapter 7.

Weight or Volume Flow

The compressed-air system is in reality a vessel that stores energy in the compressed air, energy that can be withdrawn by instruments and air-powered tools. When a portion of this stored energy is withdrawn from the system, it must be replenished by the compressor.

The performance of the compressed-air system is measured by the pressure of the air in the system. Air pressure in the system for a steady air usage and relatively constant system temperature depends on the weight of the air in the system. Boyle's law states that, for a constant gas temperature,

$$p_1 v_1 = p_2 v_2$$

where:

v = specific volume, ft^3/lb

Therefore, $$p_1 = \frac{V_1}{w_1} = P_2 \frac{V_2}{w_2}$$

where:

V_1 = total volume
w = total weight

The compressed-air system volume is constant, so

$$V_1 = V_2$$

Therefore, assuming no change in temperature or relative humidity,

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$$\frac{P_1}{w_1} = \frac{P_2}{w_2}$$

or

$$p_2 = p_1 \frac{w_2}{w_1}$$

As air is withdrawn from the compressed-air system, the weight of the air, if not replaced, and the air pressure decrease. The performance of the compressed-air system therefore depends on the weight of the air delivered by the compressor. Because each centrifugal compressor has a fixed volume design capacity, in cubic feet per minute or cubic meters per hour, the weight flow capacity is determined by the pressure temperature and relative humidity of the air entering the compressor.

The compressor characteristics curve and work input are related to flow as a function determined by physical geometry, blade angle, speed of rotation, molecular weight of gas, and other factors to a minor degree. Once this characteristic has been established for a compressor, it can be affected by inlet air pressure, temperature, relative humidity and cooling water temperature.

Effect of Inlet Air Temperature

The head relationships discussed in Chapter 7 can be used to explore the effects of inlet air temperature. Aerodynamic work input to a centrifugal compressor is proportional to polytropic head and weight flow of air to which the head is imparted. Polytropic head is measured in foot-pounds (work) per pound of air or, more simply, as feet of head. Power is then obtained by multiplying head times total weight flow and considering mechanical losses and efficiency.

Polytropic head is obtained by the equation:

$$H_p = ZRT\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)(n-1)/n - 1\right]$$

where:

H_p	=	polytropic head, ft-lb/lb
Z	=	supercompressibility factor for air; $Z = 1.0$ except at cryogenic temperatures
R	=	gas constant (1545/molecular weight)
T	=	inlet air temperature, ºR (ºR = ºF + 460)

$$\frac{n}{n-1} = \eta x \frac{k}{k-1}$$

where:

n	=	polytropic exponent
η	=	polytropic efficiency, percent
k	=	ratio of specific heats
p_2	=	discharge pressure, psia
p_1	=	inlet pressure, psia

The head, H, required to raise the air from the inlet pressure p_1 to the discharge pressure p_2 is:

$$H = C_1 \times T\left[\left(\frac{p_2}{p_1}\right)^{c_2} - 1\right]$$

where:

$$C_1 = \text{constant} = ZR\frac{n}{n-1}$$

$$C_2 = \text{constant} = \frac{n-1}{n}$$

For a fixed geometry and constant speed, air compressor head per stage is constant. The only variables are the inlet temperature and the pressure ratio. Therefore, if inlet pressure is constant and inlet temperature is increased, discharge pressure must necessarily drop to maintain the equality. Conversely, when inlet temperature decreases, discharge pressure must increase.

Inlet air temperature also affects the weight flow through all types of compressors:

$$W = \frac{QP}{RT} = \text{constant} \times \frac{P}{T} = \text{constant} \times \rho$$

where:

W	=	weight flow, lb/min
Q	=	volume flow rate, cfm
ρ	=	weight density, lb/ft^3

Weight flow through a centrifugal compressor is proportional to inlet volume and inlet pressure and indirectly proportional to inlet air temperature. Because P1/RT1 is weight density, another way of stating this relationship is that weight flow is proportional to density. As inlet temperature decreases, weight flow through the compressor increases, and vice versa, although volume flow remains constant.

Often the compressor manufacturer states the capacity for a standard air temperature, which may be as low as 60˚F. Then, when the compressor operates with 90˚F inlet air, for example, the weight *flow* is reduced by the ratio of the absolute temperatures:

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$$\frac{60 + 460}{90 + 460} = 0.945$$

For a fixed air usage, the compressed-air system pressure is reduced by the ratio of the weight flows or 5.5 percent, other factors remaining unchanged. In SI units, capacity stated in terms of a standard temperature of 68°F with an actual inlet temperature of 86°F would mean a weight reduction of:

$$\frac{20 + 273}{30 + 273} = 0.967$$

Proper performance of the compressed-air system requires that the compressor rating be guaranteed for the summertime air inlet temperature or that a weight *flow* rating be guaranteed at the same conditions.

The effect of wintertime air temperature on air density must also be considered. A compressor rated for 90˚F summertime inlet air temperature, for example, will have a 17% higher weight flow when operating with a 10˚F wintertime inlet air temperature:

$$\frac{w_2}{w_1} = \frac{90 + 460}{10 + 460} = 1.17$$

or in S.I. units:

$$\frac{32 + 273}{-12 + 273} = 1.17$$

This increased weight flow will not impair the pressure performance of the compressed-air system because the resultant increase in pressure can be relieved through a relief valve or through more frequent cycling of the compressor. But this increased weight flow will add to the cost of compressing the air because of the increased power required:

$$P = C_1 \text{ x} W$$

where:

P is the power required. Therefore, for this example, the power required by the compressor will increase 17%. Alternative expressions for power are

$$P = C_2 \text{ x} \frac{p}{t}$$

and where C_1, P_2 and C_3 are constants:

$$P = C_3 \text{ x } \rho$$

Figure 2.69 shows the effect of inlet air temperature. Increasing temperature means decreasing flow and power requirement, and decreasing temperature means increasing flow and power requirement. The implications to the buyer are twofold. First, the compressor must be rated at a sufficiently high temperature so that the plant does not run short of air on a hot day-perhaps not the highest temperature of the year, but a mean temperature based on a reasonable confidence level. Second, controls must be provided to prevent the compressor from drawing excessive additional power when the air is cooler.

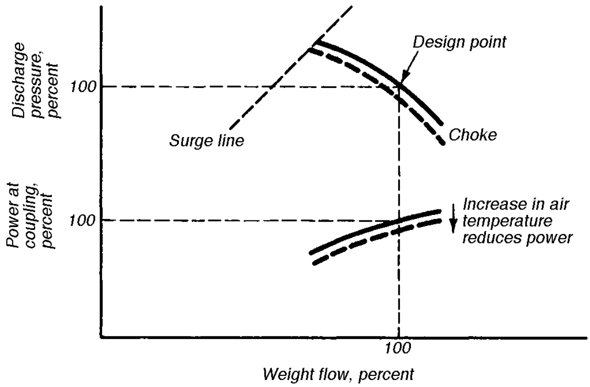

Figure 2.69 Inlet air temperature has an inverse relationship with flow and power in a centrifugal compressor. Decreasing temperature at the inlet increases flow, and more power is required to compress the denser air.

Effect of Inlet Air Pressure

A change in the inlet pressure does not affect the established pressure ratio, but the discharge pressure varies directly with changes in the inlet pressure (Fig. 2.70). Reducing the inlet pressure also reduces the weight flow through a compressor, but volume flow remains the same. Because weight flow is reduced, the power requirement is also lower.

If a given discharge pressure is required, a higher pressure ratio is required when the inlet pressure is lower, which, in turn, causes a higher work input. This factor should be considered when the compressor operates at high elevations.

Under normal operating conditions, the daily change in inlet air pressure is relatively small, except when the inlet air filter becomes dirty and needs cleaning.

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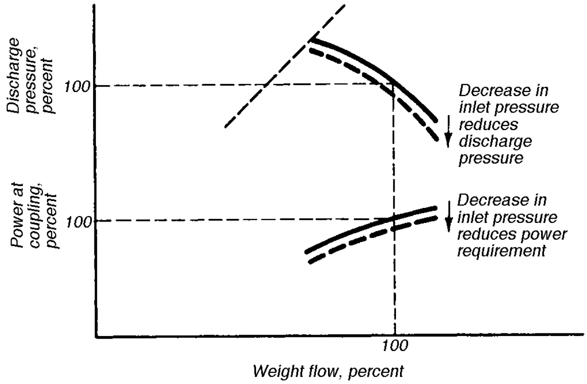


Figure 2.70 Inlet pressure effects.

Effect of Cooling-water Temperature

First-stage performance is not affected by cooling-water temperature. However, all successive stages undergo a change in performance similar to that related to air temperature (Fig. 2.71). Changes in cooling-water temperatures directly affect the temperature of the air entering the second and third or any later stages.

A reduction in cooling-water temperature increases the discharge pressure, weight flow, and the power consumption. Conversely, a higher cooling-water temperature decreases the discharge pressure, weight flow, and power consumption.

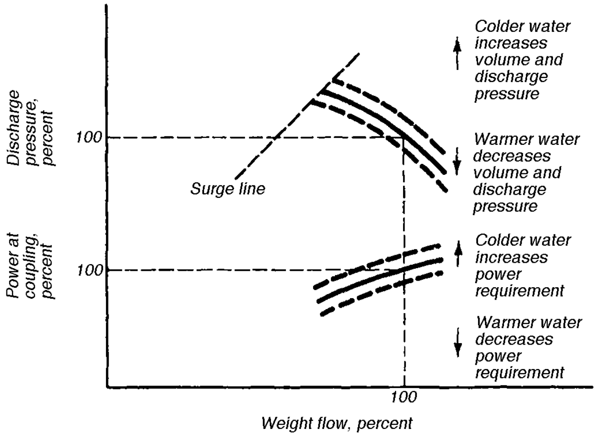


Figure 2.71 Cooling-water temperature effects.

Most significant are the combined effects of simultaneous changes in water and air temperatures, because in most plants they tend to increase or decrease together (Fig. 2.72). On a summer day, higher air and water temperatures are normal, whether the water comes from cooling towers, a public supply, a river, or the sea, although changes in air temperatures are more extreme than changes in water temperature because of the moderating effect of heat storage capacity in water. The combined effect of higher air and water temperatures is to depress the compressor characteristic, resulting in lower discharge pressure, lower weight flow, and lower power consumption. Colder temperatures increase weight flow and power requirements. This discussion is related to an uncontrolled compressor. It also describes what happens to an installed compressor selected for rated ambient conditions at a given site.

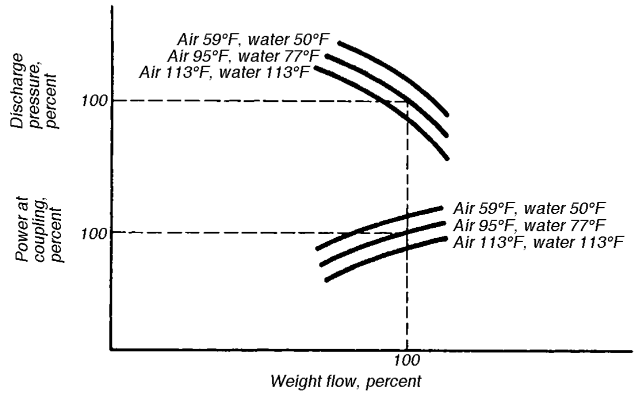


Figure 2.72 Combined effects of cooling water and inlet air temperatures.

Motors and Controls

Impellers are sized to deliver the required flow at the highest expected operating temperature. However, during winter conditions of colder air and cooling water, air density increases and more power is required to handle the increased weight flow at the same discharge pressure. The motor rating must therefore be chosen with the increased demand of the coldest expected air temperature in mind.

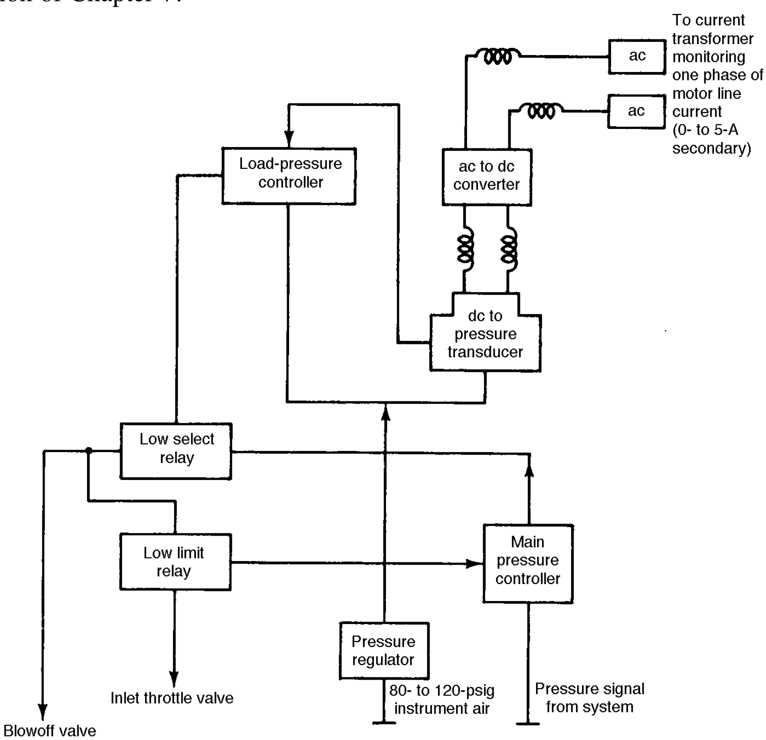
The extra cold-weather output required from a motor often can be covered by its service factor. Thus, for instance, a 2100 cfm compressor with a required horsepower close to 450 hp can be fitted with a motor rated at 450 hp and a 1.25 service factor. Under winter conditions, the service factor provides the extra horsepower required.

However, motors rated above 450 usually have a 1.15 service factor, which may not cover the extra horsepower requirement. One alternative in this situation is to use a motor with a horsepower rating higher than the compressor nominal rating, but this approach can be expensive and can require the use of controls to limit compressor output under normal operating conditions.

A more economical approach may be to use a motor load control so as to more closely match the motor and compressor. The cost of a load control, obviously, must be balanced against the cost of a larger motor.

Motor controls may also be necessary, regardless of cost, in applications that cannot accommodate the extra weight flow at the cold conditions. This is particularly true of process-control installations, where the process can accept only a certain weight flow of air. In addition, load controls may be needed in installations where the electrical system cannot supply the inrush current required by a large motor.

Typical load controls measure the current applied to the motor and close the compressor inlet throttle valve when the current exceeds a maximum allowable value. Other types of controls may be a control based on air flow or a device to adjust the inlet throttle valve in response to inlet air temperature, as seen in Figure 2.73. More detailed information on compressor controls is included in the Gas Section of Chapter 7.



A typical motor load control limits motor horsepower by adjusting the opening of the compressor inlet throttle valve. The control measures motor line current with current transformers, and the transformer signal is led to a current-to-pressure transducer that converts the signal to a control pressure. This pressure signal is compared to the output of the main pressure controller. The lower of the two signals is transmitted to the inlet throttle valve, adjusting its setting to limit the motor current and horsepower.

Figure 2.73 Controlling motor load.