Positive-Displacement Compressors

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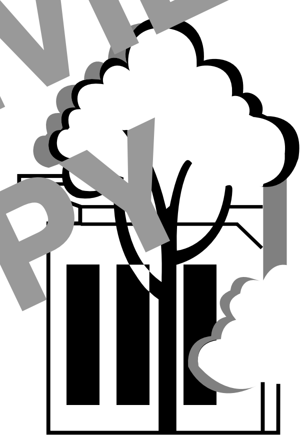
POSITIVE-DISPLACEMENT COMPRESSORS

Lesson One

Reciprocating Compressors



TPC Training Systems

46201

Lesson

1 Reciprocating Compressors

TOPICS

History of Reciprocating Compressors
Multicylinder Industrial Refrigeration Compressors
Design Features of Industrial Ammonia Reciprocating Compressors
Capacity Control
Reciprocating Compressor Lubrication
Typical Compressor Efficiency
Compressor Application Data
Compressor Units
Compound Reciprocating Compressors

OBJECTIVES

After studying this Lesson, you should be able to…

- Briefly describe the evolution of ammonia reciprocating compressors.
- Describe typical design features of today's reciprocating compressors.
- Explain how capacity control and proper lubrication are achieved in ammonia reciprocating compressors.
- Explain how to use volumetric and adiabatic efficiency data and the performance factor in sizing or selecting compressors for an application.
- Describe the function and basic design requirements of internally compounded reciprocating compressors.

KEY TECHNICAL TERMS

Capacity control 1.40 the action of loading and unloading cylinders, usually in response to suction pressure on industrial reciprocating compressors

Volumetric efficiency 1.51 the ratio of the actual volume that a compressor can pump at a given set of conditions to the theoretical displacement of the compressor, both in cfm

Compression ratio 1.52 the ratio of the discharge pressure to the suction pressure, both in psia

Adiabatic efficiency 1.54 the ratio of the power required for isentropic compression to the actual shaft power delivered to the compressor, both in bhp

Performance factor 1.55 a measure of the relationship between the actual power required at a specific condition to the capacity rating of the compressor

In the early- to mid-1800s, there was an interest in the artificial creation of cold temperatures for the production of ice and refrigeration purposes. At that time, natural ice was already being distributed worldwide, cut from the northern lakes of Europe and America. In the 1870s both David Boyle, an American mechanic, and Carl von Linde, a German scientist, independently patented and built practical ammonia compressors. Linde's compressors soon outsold Boyle's, and in 1881 there were 750 Linde refrigeration systems in 445 breweries.

About this time, in the 1880s, two machinery companies from southern Pennsylvania, Frick and York, entered the ammonia compressor refrigeration field. They had been making stationary steam engines and farm equipment and had the necessary facilities, knowledge, and skills for the manufacture of reciprocating machinery so that they could readily enter this new market.

In this Lesson, you will read about the commercial developments leading to today's industrial ammonia reciprocating compressors. Contemporary design features are discussed in detail, including a commonly used capacity-control system. This Lesson also discusses the lubrication system for reciprocating compressors and alternative compound reciprocating machines.

History of Reciprocating Compressors

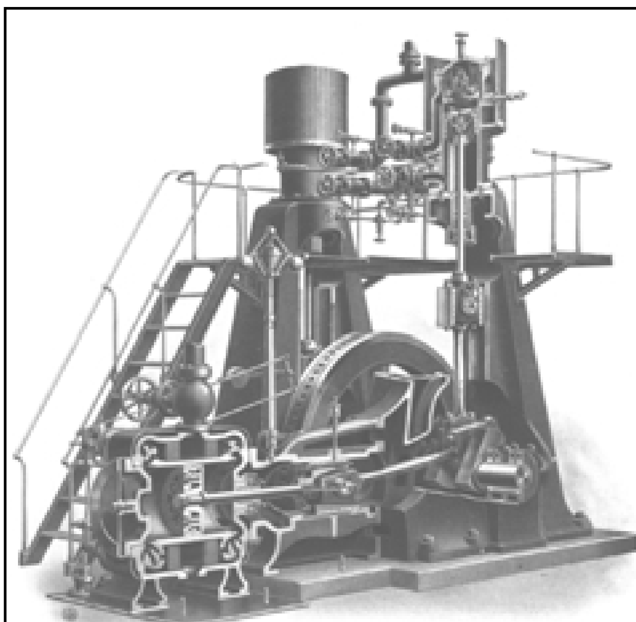
1.01 The refrigeration systems being developed in the mid-1800s were of three different forms:

- air machines, which cooled by the expansion of compressed air
- absorption machines, which used ammonia as the refrigerant and water as the absorbent
- vapor-compression machines, which initially used ether and later ammonia as the refrigerant.

The vapor-compression refrigeration system using ammonia reciprocating compressors soon became the dominant method for product cooling and making ice, far exceeding the earlier air-compression cycle and absorption systems in efficiency and equipment capacity. These early reciprocating compressors were of either vertical or horizontal design, driven by steam engines directly or belt connected. Operating speeds were about 150 to 300 rpm.

1.02 The earliest reciprocating refrigeration compressors typically consisted of two single-acting pistons arranged vertically. Figure 1-1 shows a two-cylinder vertical compressor arranged with a direct-drive steam engine. The vertical frame supports the cylinders and pistons and houses the vertical connecting rod and its guide bearing. The catwalk provides access to the cylinder valves and also to the open-topped wooden water jackets through which cooling water flows around the cylinders to provide cooling for the heat generated in the compression process. Note the connecting rod guide bearing located just below the catwalk deck and the piston rod packing gland at the base of the cylinder. Also noticeable is the long stroke of the piston and the valving at the top of the cylinder.

Fig. 1-1. Early Frick VSA two-cylinder compressor



1.03 *Single-acting* refers to the action of the piston and the compression cycle. The suction stroke occurs as the piston descends, filling the cylinder with low-pressure ammonia vapor at the evaporator pressure. This is followed by a compression and discharge stroke as the piston ascends to top dead center. During every revolution, there is one high-pressure discharge per cylinder. The two cylinders are displaced by 180° so that, for the entire compressor, there are two discharge strokes per revolution.

1.04 Concurrently with the development of the vertical single-acting machines, horizontal double-acting compressors were also introduced. These could also be either belt driven or directly connected with a steam engine. These compressors used both sides of the piston by compressing gas on one side while drawing low-pressure vapor into the expanding cylinder on the other side at the same time. Thus, each revolution produced two discharge strokes per piston. Characteristic of these machines is the dual suction and discharge piping to both ends of the cylinder. Figure 1-2 shows this arrangement in a cutaway view.

1.05 These early compressors were really nothing more than rigidly supported, large compression cylinders with most of the components—connecting rods, crankshafts, and moving parts—located outside the ammonia atmosphere. However, these machines developed into what became known as the enclosed compressor, typically arranged vertically, with two or four cylinders in line. These compressors were referred to throughout the industry as HDI (heavy-duty industrial) or VSA (vertical single-acting) compressors.

1.06 The enclosed HDI compressor required considerably less maintenance than the early compressors, which consisted mainly of a structure supporting the ammonia cylinders. The crankshaft and all running gear of the enclosed compressor was contained within the housing, which also included an oil sump from which the crankshaft, main bearings, connecting rods, and running gear were lubricated. This enclosure, which contained the ammonia atmosphere, eliminated the need to hand-oil the compressor wear components, except for the pedestal bearing outboard of the flywheel. Figure 1-3 shows a cutaway of an enclosed HDI compressor.

1.07 The water-cooling jacket was also an integral part of the cast compressor housing and surrounded the upper compression portion of the cylinders. The cooling jacket was no longer open, but now was closed with the top cover. Cooling water was piped to and from the jacket. The cylinder suc-

Fig. 1-2. Early Frick horizontal double-acting compressor

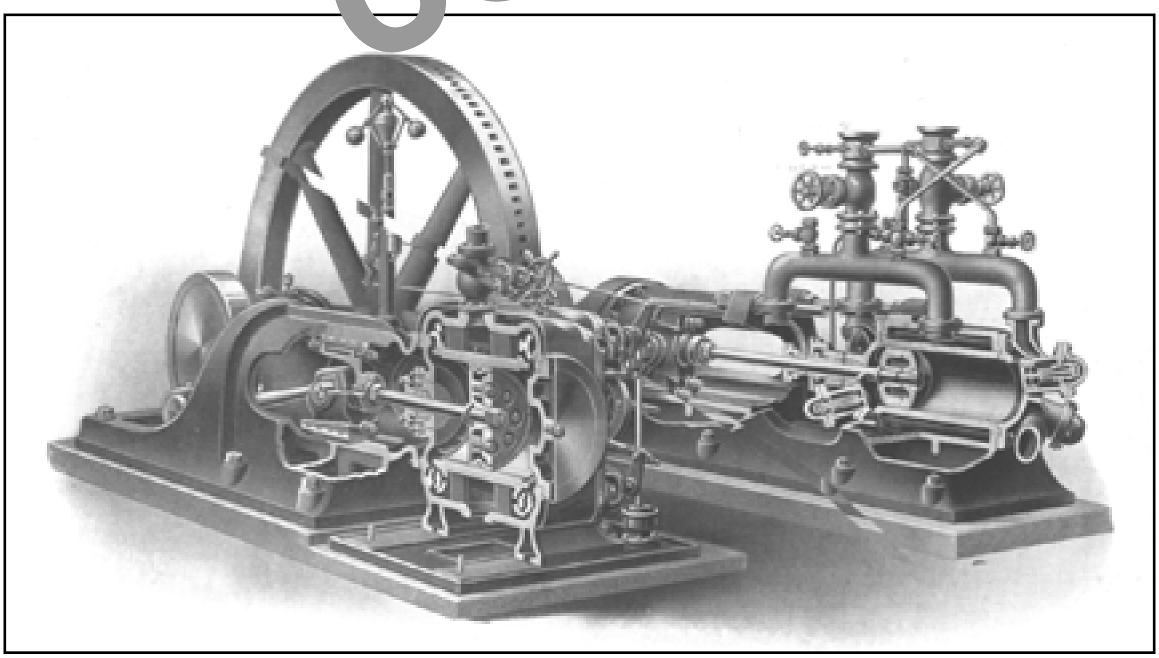
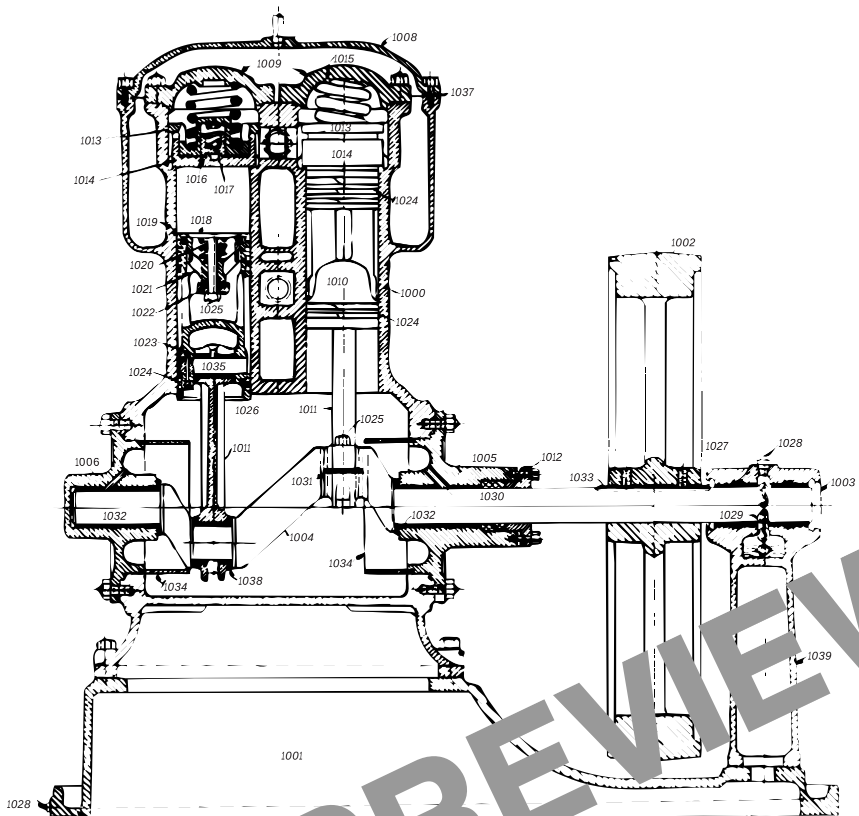


Fig. 1-3. Sectional view of Frick enclosed two-cylinder compressor



1000 COLUMN
1001 BASE
1002 BELT-WHEEL
1003 OUT-BOARD BEARING CAP
1004 CRANK SHAFT
1005 BEARING-STUFFING BOX END
1006 BEARING-OUTER
1007 HAND-HOLE COVER (NOT SHOWN)
1008 JACKET COVER 2-CYL.
1009 AMMO. CYLINDER HEAD
1010 PISTON
1011 CONN. ROD
1012 STUFFING BOX GLAND
1013 DISCHARGE VALVE CAGE
1014 DISCHARGE VALVE
1015 SAFETY HEAD SPRING
1016 DISCH. VALVE SPRING PLUG
1017 " " "
1018 SUCTION VALVE
1019 SUCTION VALVE CAGE SCREWS
1020 SUCTION VALVE SPRING
1021 " " CAGE
1022 CUSHION NUT
1023 CROSSHEAD PIN LOCK PIN
1024 PISTON RING
1025 CASTLE NUTS
1026 CONN. ROD BOX (CROSS HEAD END)
1027 HOLLOW CUP POINT SET SCREWS
1028 1/2" PIPE PLUG
1029 OIL CHAIN
1030 STUFFING BOX WASHER
1031 CONN. ROD BOLT LINERS
1032 DIE CAST BUSHING
1033 KEY (BELT WHEEL)
1034 END OIL SHIELD
1035 CROSS HEAD PIN
1036 CONN. ROD BOLT (NOT SHOWN)
1037 WATER COVER GASKET
1038 CONN. ROD BOX (CRANK END)
1039 PEDESTAL
1040 CONNECTING ROD LINER

tion valves, located in the head of the piston, drew suction vapor through the center slot in the side of the piston on the downstroke. The vapor was compressed on the upstroke and discharged through the valving above the cylinder into the area of the large safety head spring. Table 1-1 on the following page lists the refrigerating capacity of various enclosed compressors.

1.08 So far, the compressor prime movers were steam engines, either direct or belt driven. At the turn of the century, however, with the production and distribution of electric power and the commercial availability of electric motors, the electric motor soon became the compressor driver of choice. The motors, which were dependent on the power available, ranged from dc motors to large-diameter synchronous ac motors and eventually to the squirrel-cage ac induction motors commonly used today.

1.09 Most notable of the improvements during the first half of the 20th century was the use of mechanical shaft seals to replace the packing glands used in the original designs, thus reducing ammonia leakage and providing for longer run times without the need for periodic packing-gland adjustment and replacement. A second major improvement was the incorporation of an oil pump, driven at the crankshaft, which provided force-fed lubrication to the running gear and wear elements. A third improvement was the addition of methods of refrigeration capacity control.

Application 1-1

Don't think that all the old machines and drivers are a thing of the past. In 1998 the author witnessed a 1920s-vintage four-cylinder Frick vertical HDI compressor still in regular daily operation, being directly driven by a slow-speed synchronous motor. The motor was a fairly open design, about 12 in. wide and 6 ft in diameter. The machine ran smoothly and quietly, with only a characteristic click, click, click valve noise noticeable.

Table 1-1. Refrigerating capacity of enclosed compressors

Compressor bore x stroke	No. of cylinders	RPM	Capacity (tons at 0°F/96°F)
4 in. x 4 in.	1	290	1.6
4 in. x 4 in.	2	250	2.8
5 in. x 5 in.	2	250	5.6
6 in. x 6 in.	2	230	8.9
7 in. x 7 in.	2	220	13.1
8 in. x 8 in.	2	200	18.2
9 in. x 9 in.	2	200	25.8

Multicylinder Refrigeration Compressors

1.10 Probably the influence of automotive engine design led to the opportunity for increasing the capacity of the compressors while reducing the physical size and system cost. Higher production rates of common parts led to direct replacement and use of parts interchangeably in all like-model compressors, in turn improving reliability and serviceability.

1.11 The requirement for greater refrigeration capacity and smaller equipment size dictated smaller-diameter pistons and higher speeds. The early large-bore compressors were limited in capacity by the maximum speeds at which they could run. This limitation was due to several causes:

- Flow areas at the suction and discharge cylinder valves limited the volume of vapor flow into and out of the cylinder, and for the large machines, this volume was reached at the low running speeds already attained.

- The weights of the pistons and connecting rods and their physical size made operation at higher speeds unacceptable because of forces caused by unbalance vibration.

- The physical size of the two- and four-cylinder machines proved it was impractical to continue adding additional cylinders to obtain more capacity. This approach would have driven the compressor costs to a prohibitive level.

1.12 The direction in which to move became clear. If larger-size bores are a limiting factor, make the pistons smaller, decrease the stroke, run at a higher speed, and put more cylinders within a compressor housing—and all this was done. It was accomplished at first with cylinder bores of about 6 to 8 in. and strokes just equal to or less than the bore. These compressors were developed in the mid-1900s. Running speeds were considered breakneck at 500 to 800 rpm, and compressors were manufactured in V and W arrangements with up to eight or nine cylinders per compressor.

1.13 Machines in this new generation of industrial compressors were much smaller than the large HDI machines and, because of the speed and multiple cylinders, had a much greater refrigeration capacity. They used the latest technology regarding shaft-sealing techniques and had automatic cylinder-unloading capability for capacity control. In addition, the crankshafts contained counterweights and were balanced to run vibration-free. This made installation much easier, because the compressor package could be mounted on an existing sturdy concrete floor. The earlier HDI machines required that a huge mass of concrete be installed under the compressor to hold it in place during operation.

1.14 Following the mid-century mark, compressor bores again became smaller, speeds increased further, and some manufacturers increased the number of cylinders per compressor up to 16. The capacity of the compressors was also greater than before. Industrial compressors now had cylinder bores of 2.75 to 4.50 in. with maximum speeds ranging from 1200 for most and up to 1800 rpm for certain R-12 and R-22 applications.

1.15 This is where the industry is now. It should be noted that this discussion has dealt with industrial ammonia compressor development only. It has left untouched the concurrent development of the smaller commercial, residential, and appliance compressors, which are not suitable for industrial ammonia refrigeration service.

Design Features of Reciprocating Compressors

1.16 Figure 1-4 shows a contemporary Vilter Model 450XL series compressor. The 450XL series consists of six compressor sizes with 2 to 16 cylinders, all with the same bore and stroke and using common internal components. Most manufacturers typically standardize on certain bore and stroke dimensions for a line of compressors and then develop a range of compressors with two or four cylinders up to 12 or 16 cylinders to provide good coverage over a wide range of capacities.

1.17 The external view of the compressor, shown in Fig. 1-4A, is typical of most industrial refrigeration compressors. Each top head contains two cylinders. This six-cylinder compressor has three top heads—the third is located symmetrically on the reverse side of the machine. The large angle valve on the left end is the suction valve through which ammonia vapor enters the compressor from the evaporator. The flanges of the smaller discharge valve are just visible on the farther end of the compressor.

1.18 At that farther end below the valve is the belt-drive flywheel. The tank-type appendage below the suction valve is the oil filter, and the small tubular device attached horizontally alongside the compressor is the oil cooler. Note the lubrication oil lines connecting the oil filter housing cap to the oil cooler. The two capped connections at the near end of the oil cooler are for the coolant (typically water), which is connected at the jobsite.

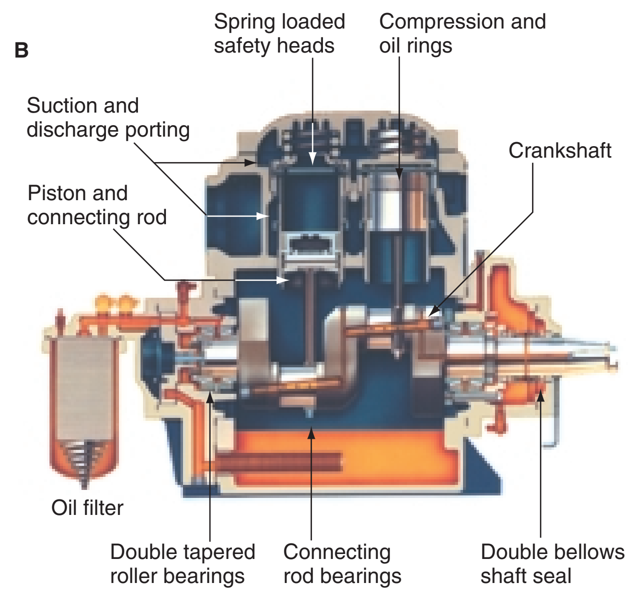
1.19 The cylinder top head is made of two sections. The lower thick section is the cylinder discharge head. The thinner domed section on top, containing the open elbow along its side, is the discharge cooling head, typically supplied with water or liquid ammonia as the coolant to assist in removing some of the high discharge temperature associated with ammonia compressors.

1.20 Figure 1-4B shows the internal section of the same 450XL compressor. Note that these large industrial refrigeration compressors from different manufacturers have quite similar construction and features, which you will also notice in sectional views later in this Lesson. All compressors include an oil-supply sump at the bottom of the housing. The oil is directed through a fine-mesh strainer located in the sump up to an oil pump, which is driven directly by the crankshaft. This provides for a pressurized or force-fed lubrication system whenever the compressor is in operation.

1.21 Because the compressor must be in operation to establish and maintain the oil pressure, there is no lubrication oil flow for the first several revolutions at each startup. All industrial compressors have oil-pressure cutout switches, sometimes with an alarm feature, which stop the compressor should the oil pressure differential fall below a preset minimum. For reciprocating compressors, the ammonia pressure in the crankshaft area, over the oil sump, is equalized to the suction pressure and typically requires a minimum differential oil pressure of about 30 to 40 psi above the suction pressure.

1.22 Because there is no oil pressure differential immediately at startup, the oil-pressure cutout switch has a timed bypass delay of at least 15 seconds. This delay permits the compressor to establish oil pressure and avoid nuisance cutouts.

Fig. 1-4. Vilter 450XL series compressor–external and internal views



CAUTION

Compressors that have been shut down for a prolonged period or for servicing must have some oil manually pumped (via a hand or mechanical pump) through the lubrication system to ensure a supply of oil to all critical bearing surfaces before startup. Failure to follow this procedure may result in excessive wear and early compressor failure.

1.23 The oil pump is generally an internal gear type with a crescent offsetting a center spur gear from the internal ring gear. Oil flow rates are about 3 to 5 gpm, depending on compressor size. As alternatives, some manufacturers use external spur gear or vaned rotary pumps. Oil in the Vilter compressor is directed through an oil filter, then to the main tapered roller bearings, and then through the crankshaft to lubricate the connecting rods and the shaft seal area. Oil pressure is also used to actuate the compressor cylinder load/unload (capacity-control) mechanism, which is discussed later in this Lesson.

1.24 The main bearings on the Vilter compressor are double tapered roller bearings located at either end of the shaft. These bearings support the crankshaft in the housing and carry the radial loads imposed by the compressing action of the piston/connecting rod set in each cylinder. The tapered roller bearings also lock the crankshaft in position axially to center the connecting rods at each cylinder. Most other manufacturers do not use tapered rollers, but instead use journal bearings for the radial loads and thrust washers or collars for axial positioning.

1.25 The crankshafts are typically of forged steel or of ductile (nodular) cast iron. The ductile cast iron has strength properties similar to those of steel forgings and permits economical production because it does not need forging dies, which are extremely expensive. Each crankpin carries from two to four connecting rods—three in the case of this Vilter 450XL. At either end of the shaft, just inside the main bearings, the shaft has counterweights that balance out the primary imbalance associated with the reciprocating action of the pistons and the upper portion of the connecting rods. The crankshaft bearing surfaces may be flame- or case-hardened or given a special surface treatment to provide a microthin but hard-wear surface.

1.26 To be able to attach the shaft to a driver—via belt or direct connection to a motor—the shaft must extend to the outside of the compressor. The extension to the atmosphere must be sealed from the ammonia environment within the compressor. This is accomplished with either a single or a double mechanical shaft seal. One portion of the seal assembly is fastened to the rotating shaft and makes a static seal at the shaft. The other portion of the seal is fixed to the nonrotating housing and also makes a static seal at the housing. The two highly polished, extremely flat, lapped mating surfaces are maintained in flexible (spring-loaded) contact with one another to provide the running seal. The seal assembly runs in an oil bath that provides the necessary lubrication and cooling.

1.27 The connecting rods may be either forged steel or aluminum, typically with steel-backed babbitt bearing shells at the crankshaft end and bronze or steel-backed babbitt bushings at the wristpin end. A lubrication hole is generally drilled through the shank of the rod to lubricate the wrist pin area at both the rod and the piston bosses.

1.28 The piston fits within the removable cylinder sleeve and typically carries three to four piston rings, which are a combination of compression rings at the top and one or two oil-scraper rings at the bottom. The manufacturer provides specific instructions regarding the location and orientation of piston rings. They are important in providing the necessary cylinder sealing for optimum capacity and minimum oil loss (carryover) from the compressor.

1.29 The shape of the head of the piston varies from manufacturer to manufacturer, depending on the design of the suction and discharge valve plates. Piston heads may be flat, as on the Vilter compressor in Fig. 1-4, or extended and dished, as on the Mycom compressor shown later in this Lesson. In either case, the piston at top dead center will come as close as practical to the cylinder valve service mounted on the top of the cylinder liner. This clearance runs in the range of 0.030 to 0.045 in. and accounts for normal manufacturing tolerances and some clearance for an oil film that might be carried on the top of the piston. It is important to come as close as possible to the valving, but not to touch it with either the piston itself or with any liquid film on the piston, because the film is essentially incompressible.

1.30 In the top head are two large safety head springs that hold the cylinder valve service assemblies in place. Their purpose is to provide a flexible clamping of the valve plate to the cylinder sleeve, permitting movement of the valve plate from the sleeve if excess liquid oil or refrigerant is in the cylinder during the discharge stroke. This is to prevent component damage during slugging. Note that not all manufacturers use this approach. Also, it is possible that this feature worked well on slow-speed (150 to 300 rpm) compressors, but is not effective at the higher speeds now common. Proper operation and design should minimize the need for this feature, which should not be depended on.

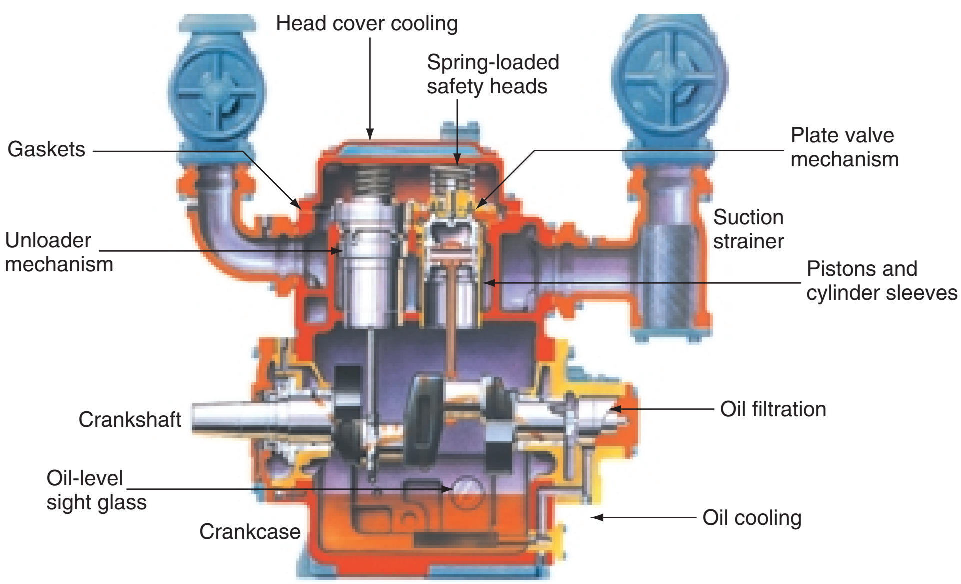
1.31 Figure 1-5 shows the internal section of a Mycom 6WB compressor, including some additional detail that is not included on the Vilter 450XL section, but is typical of all industrial compressors. One feature is the suction strainer screen basket on the compressor inlet. The other is the oil-level sight glass shown at the oil sump. Overall, the compressors look quite similar, except that the main crankshaft bearings are the journal type and the piston tops are sculptured and scalloped to mate with the valving.

1.32 The journal bearings, which are commonly used by most manufacturers, are easily seen on the section of the Mycom 6WB. Also note that the inner ends of the journals become the thrust bearings, which locate and maintain the axial position of the crankshaft. This compressor also has a commercial filter with a spin-on cartridge (not shown), which is located externally along with the shell-and-tube oil cooler. Note the relative sizes of the suction and discharge shutoff valves. The suction valve is always the larger because the inlet gas is at a lower pressure, is less dense, and takes up more volume than the refrigerant at the discharge after compression.

1.33 The housings for virtually all industrial compressors—as well as the various top heads, cover plates, and bearing housings—are made of fine-grain, nonporous, leak-tight, high-strength gray iron or meehanite semisteel castings. The compressors are tested to be leak-tight and are given a strength test at 150% of the design working pressure of the compressor.

1.34 The compressor suction area surrounds the cylinders, and ammonia vapor enters the cylinder on the piston downstroke. At bottom dead center the suction valve closes, and on the upstroke the cylinder diminishes, compressing the gas. As the piston approaches top dead center, the pressure increases above the discharge pressure, opening the discharge valve. The compressed gas enters the top head through the cylinder discharge valve, is manifolded to the discharge service valve, and continues on to the condenser.

Fig. 1-5. Mycom 6WB compressor



1.35 The cylinder suction valve assembly consists of a flat suction valve disk or ring, which rests on inner and outer valve seats that are formed into the top and are part of the cylinder liner. The suction valve ring is held in place with a number of small helical springs that are recessed into the suction valve plate. The suction valve plate contains a recessed pocket into which the suction valve ring rises during the suction stroke to permit suction gas to enter the cylinder. The depth of the recess is critical, because it controls the flow rate of refrigerant gas and also the impact stresses and motion of the ring.

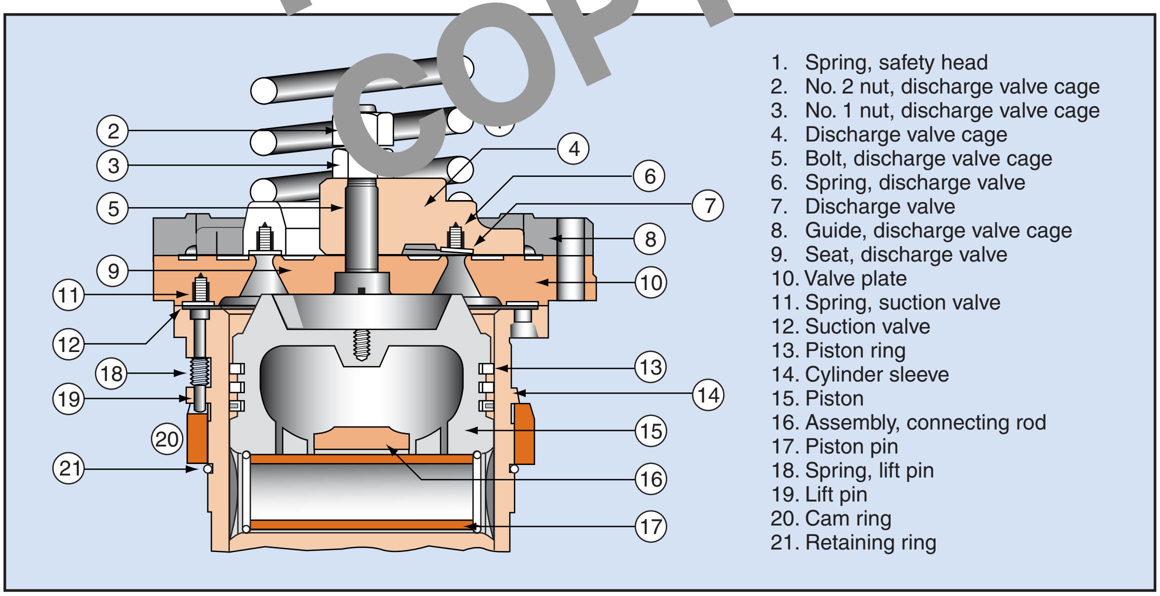
1.36 The discharge valve assembly is mounted above the suction valve plate. The discharge assembly consists of the discharge valve cage, a mushroom discharge seat, a flat discharge valve disk or ring (smaller than the suction ring), several small helical springs, and a bolt to clamp the mushroom, ring, and springs to the cage. The inner discharge seat is located on the mushroom, but the outer discharge seat is machined into the top of the suction valve plate. The discharge cage also has a machined recess that permits the discharge valve ring to rise the desired lift amount, controlling the rate of discharge gas flow and the motion of the valve ring.

1.37 Figure 1-6 shows how the top of the piston is contoured to mate with the shape of the cylinder valve service. This is the clearance space, which must be kept as small as practical to minimize the trapped high-pressure gas that cannot be discharged out of the cylinder. The space must be sufficiently large, however, to avoid physical contact between the piston and valve plates even with some liquid oil or refrigerant on the piston.

1.38 Regardless of the manufacturer, certain operating characteristics are signs of valve failure. Discharge valve failure is indicated by high discharge temperatures. Suction valve failure is indicated by reduced power consumption.

The Programmed Exercises on the next page will tell you how well you understand the material you have just read. Before starting the exercises, remove the Reveal Key from the back of your book. Read the instructions printed on the Reveal Key. Follow these instructions as you work through the Programmed Exercises.

Fig. 1-6. Mycom GWB cylinder sleeve and valving



1-1. Important improvements to early compressors included mechanical ________ seals, a(n) ________ pump, and a ________-control system.	1-1. SHAFT; OIL; CAPACITY Ref: 1.09
1-2. Increased refrigerating capacity from smaller machines resulted from using ________ pistons that were smaller and running at ________ speeds.	1-2. MORE; HIGHER Ref: 1.12
1-3. Today's ammonia reciprocating compressor has up to ________ cylinders and runs at a maximum speed ranging up to about ________ rpm.	1-3. 16; 1200 Ref: 1.14
1-4. The oil-pressure cutout switch must have a timed bypass ________ to permit compressor startup without nuisance ________.	1-4. DELAY; CUTOUTS Ref: 1.22
1-5. Most manufacturers use journal bearings for the ________ loads and thrust washers or collars for ________ positioning.	1-5. RADIAL; AXIAL Ref: 1.24
1-6. When used, ________ springs hold the cylinder valve service assemblies in place.	1-6. SAFETY HEAD Ref: 1.30
1-7. Ammonia vapor enters the cylinder on the piston ________.	1-7. DOWNSTROKE Ref: 1.34
1-8. The discharge valve assembly is mounted above the ________.	1-8. SUCTION VALVE PLATE Ref: 1.36

Capacity Control

1.39 Notice the long slender pins (item 19 in Fig. 1-6 on page 12) that rest on a rotatable band located on the outside of the cylinder sleeve. The pins fit through holes in the cylinder sleeve and have a captured spring that forces the pins down against the band. The upper end of the pins extend into the suction-gas entry area just under the suction valve ring. The purpose of these lift pins is to permit loading and unloading of individual cylinders to control the capacity of the compressor and thus match the system load requirements.

1.40 The action of loading and unloading cylinders is called *capacity control*. Capacity control on industrial reciprocating compressors is generally in response to the refrigeration system suction pressure. The compressor capacity-control system can be set to respond to a predetermined pressure, and deviation caused by a varying system load will result in the compressor automatically loading and unloading to maintain the required pressure and therefore the evaporating temperature.

1.41 To *load* a cylinder or a bank of cylinders, oil pressure is directed to a small capacity-control piston, which moves a rod attached to the rotatable band. Rotating the band causes the pins to follow down the angular ramp on the band, withdrawing the lift-pin upper surface below the suction valve ring. In this position the suction valve is free to actuate freely in response to the piston stroke position, drawing suction gas on the downstroke and closing and permitting compression and discharge on the upstroke. The term for a cylinder in this position is *loaded*. This condition is due to oil pressure acting on the capacity-control piston.

1.42 To *unload* a cylinder, the oil pressure is removed from the capacity-control piston and a suitably sized spring returns the rod, rotating the band, causing the lift pins to rise up the ramp on the band. The lift pins lift the suction valve ring off its seat and force it against the backstop. In this position, suction gas enters the cylinder on the piston downstroke, but the valve cannot close at bottom dead center, remaining open during the upstroke. No compression or discharge takes place, because the gas is merely pushed out of the cylinder as the piston rises back through the open suction valve. The term for a cylinder in this position is *unloaded*. This condition is due to the removal of oil pressure from the capacity-control piston and the action of the return spring.

1.43 It should be noted that the compressors do not have unloaders on all cylinders, except in some special cases, to prevent operation with no flow through the compressor. Two- and four-cylinder compressors have a minimum of one cylinder loaded. Six- and eight-cylinder compressors have a minimum of two cylinders loaded. Twelve- and sixteen-cylinder compressors have at least four cylinders loaded at all times.

1.44 Compressor starting torque, and therefore motor starting requirements, are also minimized by this unloading feature. There is no oil pressure immediately at start. The pump does not function until the compressor turns, and a few seconds are required to establish sufficient oil pressure. In other words, whether or not the system demands that all cylinders be loaded, the compressor always starts on the minimum number of cylinders—those that do not have unloader mechanisms—reducing starting torque for the critical seconds of transition from 0 rpm to operating speed.

Reciprocating Compressor Lubrication

1.45 Compressor lubrication is essential to the successful operation of all industrial compressors. The lubrication is provided under pressure, as force-fed or pressure-fed lubrication, which results from a mechanical oil pump that is located at the crankshaft and directly driven by the crankshaft. Oil is drawn from the oil sump in the compressor crankcase, typically through a wire-mesh strainer to prevent foreign material from entering the pump. Many systems also include a coalescing oil separator, as is discussed in Lesson Five.

1.46 The remainder of the lubrication system consists of the internal and external oil passages from the pump to the various points of lubrication—the crankpins, main journals, connecting rods, and shaft seal. The lubrication system on most compressors also provides the hydraulic pressure to activate the capacity control. In addition, most compressors have an oil filtration system, which often filters all the oil pumped, or a bypass system, which filters only a bypass stream of the total oil flow. Some manufacturers provide oil filtration only as an option. If your compressor does not have a filter on the lubrication system, take measures to add one, even if it is only on a bypass. This can be accomplished with a suitable spin-on 15- to 20-micron pleated paper filter handling from 0.5 to 1 gpm of oil. Within a few minutes of operation, the entire oil sump quantity will have passed through the filter and the compressor will be safe from the grit in the unfiltered oil.

1.47 The oil also sprays from the connecting rods at the crankpins and both lubricates and removes heat from the cylinder sleeves, inside the pistons, and the inner deck area of the compressor. The heat pickup by the oil increases its temperature enough that most ammonia compressors (except boosters) have some provision for oil cooling. This oil cooling is generally provided by a water-cooled or refrigerant-cooled shell-and-tube heat exchanger.

1.48 The lubricants used for ammonia reciprocating compressors have a viscosity of 300 to 350 SUS (Saybolt universal seconds), about the same viscosity as an SAE 20 motor oil. This viscosity is also indicated as ISO VG 60, the International Standards Organization designation of 60 centistokes viscosity grade. The lubricants may be of naphthenic or paraffinic crude oil origin and should be specifically designated for refrigeration compressor service. The more acceptable lubricant is a hydro-treated paraffinic oil, which has stable characteristics, a high coking temperature, a low pour point, and a low oil carryover rate through the oil separator. More specific information about lubricants will be covered in Lesson Five of this Unit.

1.49 Reciprocating compressors typically contain a crankcase oil heater inserted below the oil level in the oil sump. The purpose of the heater, which is energized only when the compressor is not running, is to keep the oil in the crankcase slightly warmer than the machine room or surrounding area so that during prolonged off cycles, liquid ammonia does not condense in the oil sump. These heaters are typically rated at about 300 watts and often contain inherent thermostats to turn them off when the oil is sufficiently warm, at about 120 to 140°F. Heaters directly immersed in the oil must have sufficient surface to prevent the oil from coking on the heater. (Coking is a particular form of oil degradation.) This is accomplished by specifying heaters with a maximum watt density of about 15 to 17 watt/in^2.

Typical Compressor Efficiency

1.50 Two efficiency terms used in conjunction with positive-displacement compressors are important to understand—volumetric efficiency, VE, and adiabatic (ideal) efficiency, N_a. However, a third relationship, called the performance factor, or PF, is the most useful and practical for comparing similar-sized industrial compressors from several manufacturers.

1.51 You may recall from the previous Unit that *volumetric efficiency*, VE, is the ratio of the actual volume (Acfm) that a compressor can pump at a given set of conditions to the theoretical displacement of the compressor, Dcfm.

$$\mathrm{VE} = \frac{\mathrm{Acfm}}{\mathrm{Dcfm}}$$

1.52 The volumetric efficiency data is generally presented in graphical form plotted against compression ratio, where the *compression ratio*, CR, is equal to the ratio of the discharge pressure (psia) to the suction pressure (psia), as follows:

$$\mathrm{CR} = \frac{\text{discharge pressure}}{\text{suction pressure}}$$

Figure 1-7 on the following page is a generalized plot of volumetric efficiency for industrial reciprocating compressors.

1.53 Note how the VE decreases as the CR increases and approaches 10.0. At that point the CR is down to 50% of the theoretical displacement of the compressor. This coincides with the limitation of operation of most ammonia reciprocating machines, because at these conditions the discharge temperature is in excess of 300°F, and the oil will char through the valving, leaving a black sooty deposit that results in frequent servicing and loss of performance.

1.54 *Adiabatic efficiency*, N_a, is the ratio of the power required for isentropic (constant entropy, ideal) compression to the actual shaft power delivered to the compressor. For industrial open compressors, the actual power is given in motor brake horsepower, not the motor power input.

$$N_a = \frac{\text{isentropic power}}{\text{shaft input power}}$$

The adiabatic efficiency is determined by manufacturers by test and is used to establish power requirements

Fig. 1-7. Typical industrial ammonia reciprocating compressors volumetric efficiency

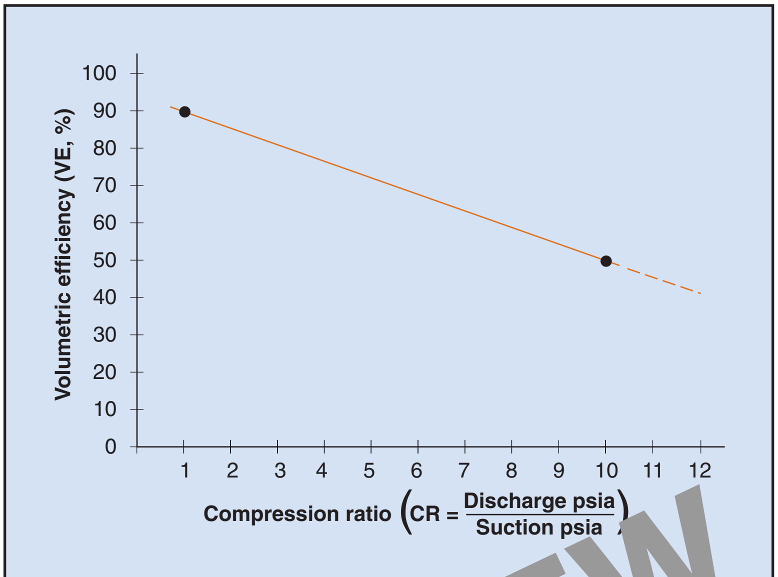


Fig. 1-8. Capacity/bhp ratings for typical ammonia compressors

A. York model RW-4A compressors

Unit model: Saturated discharge temp. °F	Saturated suction temp. °F	RW44A Tons	RW44A Bhp	RW44A MBh	RW64A Tons	RW64A Bhp	RW64A MBh	RW84A Tons	RW84A Bhp	RW84A MBh	RW124A Tons	RW124A Bhp	RW124A MBh	RW164A Tons	RW164A Bhp	RW164A MBh
	-15	17.6	[illegible]	295	26.4	[illegible]	[illegible]	[illegible]	[illegible]	587	52.8	97.2	881	70.4	129.1	1173
	-10	20.8	35.2	338	[illegible]	[illegible]	[illegible]	41.5	69.3	674	62.3	104.0	1012	83.0	138.2	1348
	-5	24.3	37.3	[illegible]	[illegible]	55.4	[illegible]	48.5	73.5	769	72.8	110.3	1154	97.0	146.5	1537
	0	28.1	39.2	[illegible]	[illegible]	58.3	653	56.1	77.4	870	84.2	116.1	1305	112.2	154.3	1739
	5	32.3	41.0	[illegible]	[illegible]	[illegible]	735	64.5	81.0	980	96.8	121.4	1470	129.0	161.4	1959
	10	36.9	42.6	[illegible]	[illegible]	63.4	825	73.7	84.2	1099	110.6	126.3	1649	147.5	167.9	2197
	15	41.9	44.1	[illegible]	[illegible]	65.6	920	83.7	87.1	1226	125.6	130.7	1840	167.5	173.8	2452
80	20	[illegible]	[illegible]	683	[illegible]	[illegible]	[illegible]	94.7	89.6	1364	[illegible]	134.4	2047	189.5	178.7	2728
90	25	51.0	51.4	742	76.5	76.6	1113	102.0	101.8	1483	153.0	152.7	2224	204.0	203.1	2965
	30	57.3	52.8	822	86.0	78.7	1232	114.7	104.6	1642	172.0	156.8	2463	229.4	208.6	3283
	35	64.1	53.6	906	96.2	79.8	1357	128.3	106.1	1809	192.4	159.2	2714	256.6	211.8	3618
	40	71.5	54.3	996	107.2	80.9	1493	143.0	107.6	1989	214.5	161.4	2984	286.0	214.7	3978
	45	79.5	55.0	1093	119.2	82.0	1639	158.9	108.9	2184	238.4	163.4	3277	317.9	217.4	4368
	50	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	176.1	110.3	2394	[illegible]	165.5	[illegible]	352.3	220.1	4788

B. Mycom N2-12W compressors at 95°F condensing

Refrigerant	Model	Displacement at 1,200 rpm (Cfm)	Refrigeration capacity at 95°F condensing temperature, U.S. tons: –20	–10	0	10	20	30	Brake horsepower at 95°F condensing temperature, Bhp: –20	–10	0	10	20	30
R-717	N2WA	41.8	3.9	5.6	7.7	10.3	13.4	17.1	9.8	11.3	12.7	13.9	14.9	15.5
	N4WA	91.2	8.4	12.2	16.8	22.5	29.2	37.2	21.4	24.7	27.7	30.4	32.5	33.7
	N6WA	136.8	12.6	18.3	25.2	33.7	43.8	55.8	32.1	37.0	41.6	45.6	48.7	50.6
	N8WA	182.4	16.9	24.4	33.7	44.9	58.4	74.4	42.9	49.4	55.5	60.8	65.0	67.5
	N4WB	224.4	20.8	30.1	41.5	55.3	72.0	91.7	52.8	60.8	68.4	74.9	80.1	83.1
	N6WB	337.2	31.2	45.1	62.2	83.0	108.0	137.6	79.2	91.3	102.6	112.4	120.1	124.7
	N8WB	450.0	41.6	60.1	82.9	110.7	143.8	183.4	105.6	121.7	136.7	149.9	160.1	166.3
	N12WB	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	249.4

for their equipment throughout the range of operation. It is normally not published.

1.55 The *performance factor*, PF, is a measure of the relationship between the actual power required at a specific condition to the capacity rating of the compressor.

$$PF = \frac{\text{actual bhp}}{\text{rated capacity in tons}}$$

The performance factor—bhp/ton—provides a practical way to analyze and assist in choosing between several alternative compressor selections for a given operating condition. The calculation is easily accomplished directly from the manufacturer's rating information. Simply determine the brake horsepower (bhp) required for the application and then divide the bhp by the refrigeration rating in tons. Note that you must be sure to correct the manufacturer's data for your specific condition—including actual speed, superheat, liquid subcooling, and so on. The lower the performance factor, PF in bhp/ton, the more energy-efficient is the selection.

Compressor Application Data

1.56 Manufacturers typically supply compressor application data, limits of operation, and rating tables or graphs of capacity in their engineering-guide literature. Some manufacturers also provide computer disks with complete rating information so the customer can obtain ratings at his or her specific conditions, without interpolation of tabular data. Sample sections of typical rating and application data from various manufacturers are presented in Figs. 1-8, 1-9, and 1-10 on the following page.

Fig. 1-9. Capacity/bhp rating graph for Vilter 454XL compressor at 1200 rpm and 95° F condensing

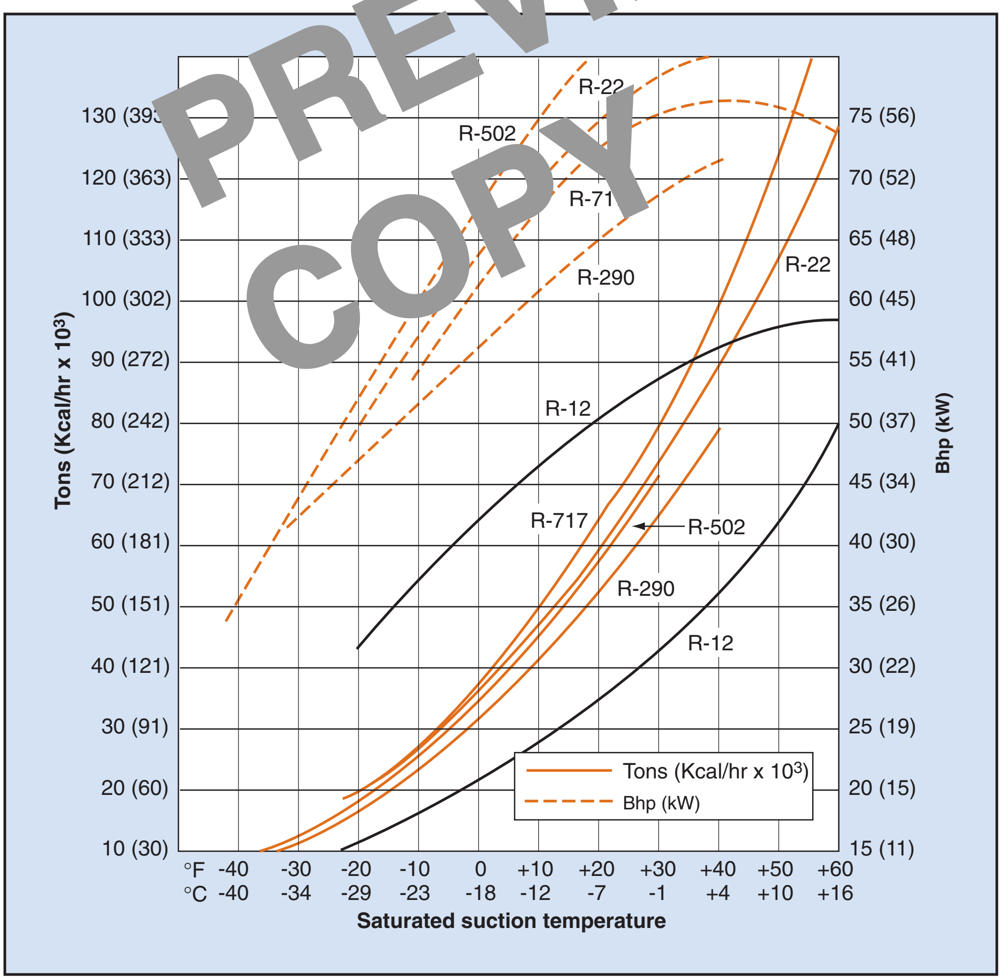


Fig. 1-10. Vilter general specifications and limitations

ITEM		452XL	454XL	456XL	458XL	4512XL	4516XL
Number of cylinders		2	4	6	8	12	16
Maximum rpm		1200	1200	1200	1200	1200	1200
Bore & Stroke – in.		$4^1/_2 \times 4^1/_2$	$4^1/_2 \times 4^1/_2$	$4^1/_2 \times 4^1/_2$	$4^1/_2 \times 4^1/_2$	$4^1/_2 \times 4^1/_2$	$4^1/_2 \times 4^1/_2$
(mm)		(114 x 114)	(114 x 114)	(114 x 114)	(114 x 114)	(114 x 114)	(114 x 114)
Cfm @ maximum rpm (m^3/hr)		99.4(169)	199(338)	298(507)	398(676)	597(1014)	796(1352)
Tons (Kcal/hr x 10^3) Refrigeration @ 95°F condensing	R-717 (10°F)	24(73)	49(148)	73(221)	97(293)	146(442)	195(590)
	R-12 (40°F)	27(82)	55(166)	82(248)	109(330)	164(496)	219(662)
	R-22 (20°F)	29(88)	59(178)	88(266)	117(354)	176(532)	235(711)
	R-502 (–20°F)	9(27)	19(57)	28(85)	37(112)	56(169)	75(227)
	R-290 (0°F)	16(48)	31(94)	47(142)	62(187)	94(284)	125(378)
Suction connection – in. (mm)		$2^1/_2$(64)	3(76)	4(102)	4(102)	5(127)	5(127)*
Discharge connection – in. (mm)		2(51)	$2^1/_2$(64)	3(76)	3(76)**	(2)3(76)	(2)3(76)
Unit weight less motor – lb (kg)		1900(862)	2700(1225)	3100(1406)	3400(1542)	5300(2404)	5800(2630)
Oil charge – gallons (liters)		5(19)	7(27)	7(27)	7(27)	14(53)	14(53)
Standard steps of unloading (%)		0	50	33/66	25/50	33/66	25/50
Option 1 steps of unloading (%)		50	25/50/75	–	25/50/7[illegible]	–	25/50/75
Option 2 steps of unloading (%)		100	50/100	33/66/100	25/[illegible]5/1[illegible]0	33/66/100	25/50/75/100
Maximum discharge temp. – °F (°C)		300(149)	300(149)	300(14[illegible]	[illegible]	300(149)	300(149)
Crankcase oil temp. range – °F		110-130	110-130	1[illegible]130	[illegible]130	110-130	110-130
(°C)		(43-54)	(43-54)	(43[illegible]	(4[illegible]4)	(43-54)	(43-54)

*Add 1 in. (25.4 mm) for halocarbon units. **Add 1 in. (25.4 mm) for [illegible] units.

Compressor Units

1.57 Most of this Lesson has discussed reciprocating compressors, but except for service replacements and those sold for further packaging into refrigeration systems, most compressors reach the customer as part of a compressor unit. The compressor unit is a much more convenient way to obtain the compressor, ready for tie-in to your system.

1.58 The compressor unit typically consists of the following:

- compressor
- base assembly—belt or direct drive
- controls—electromechanical or microprocessor with operating and safety devices
- motor and drive package installed
- optional equipment required.

Figure 1-11 shows a factory-packaged compressor unit.

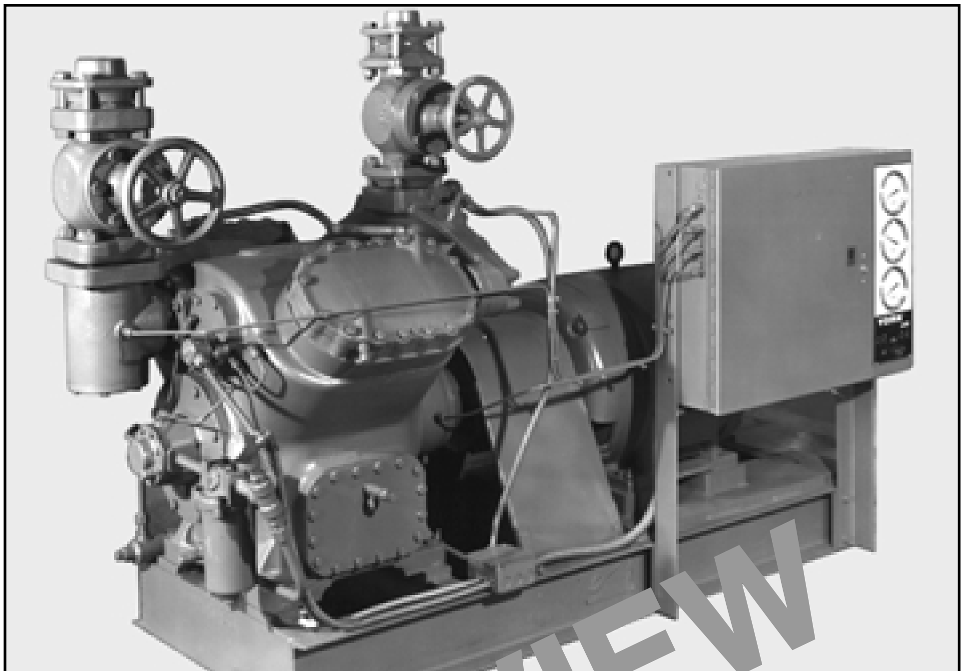
Compound Reciprocating Compressors

1.59 There is a special class of reciprocating compressors known as compound or internally compounded compressors. These compressors are unique in that they contain within a single housing both the low (booster) and high stages of a two-stage compressor system. This is accomplished by the following:

- internal divisions within the compressor body to separate the two pressure stages
- provisions for both the booster low-pressure and the intermediate high-stage suction valve connections
- provisions for compressor discharge valve connection to the condenser.

1.60 Companies that manufacture compound compressors typically use a modification of their existing compressor housings and running gear (crankshaft, liners, rods, pistons, cylinder valving, and so on) to provide the internally compounded machine economically. The advantage of these compressors is for the smaller industrial plant where one or two compound

Fig. 1-11. York model R compressor unit



compressors are all that are required to maintain the low-temperature load. The system is simplified as are the controls. It takes only a single control panel and one motor to operate each compressor.

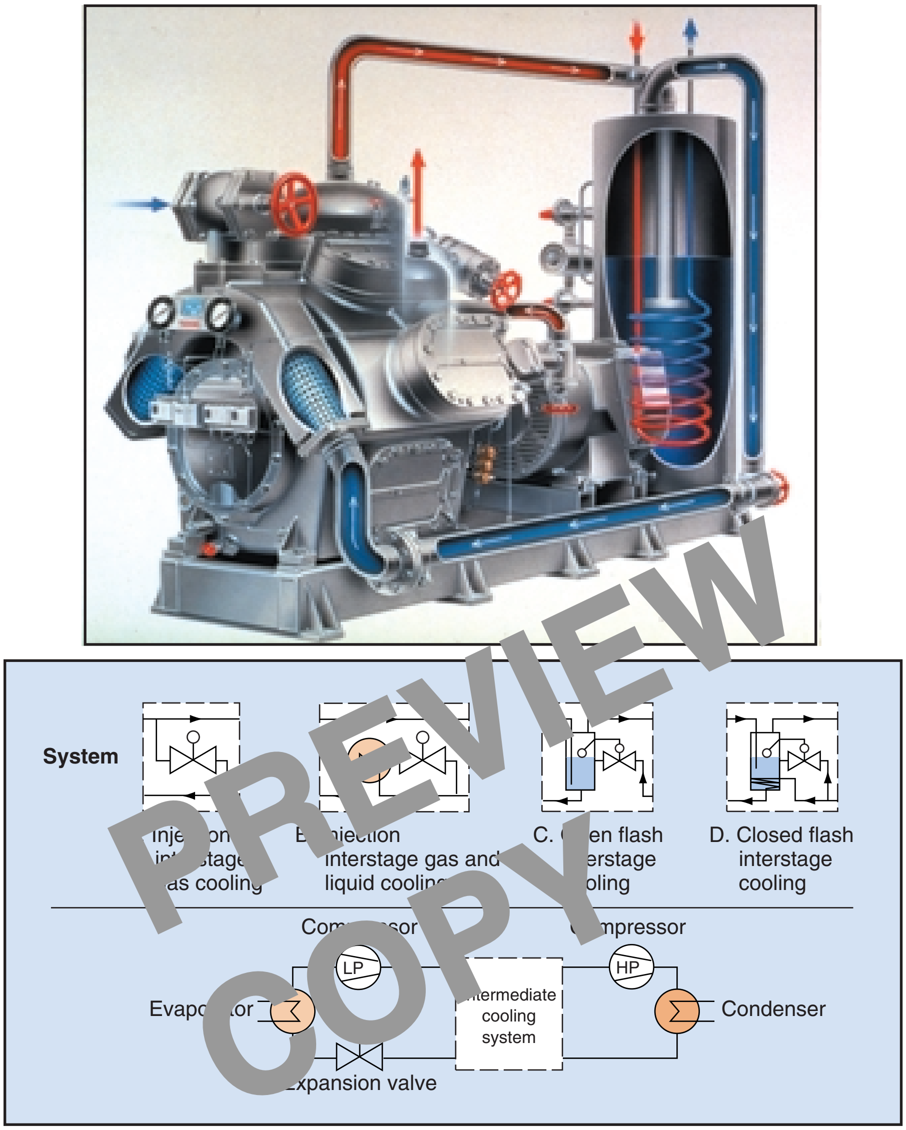
1.61 The two criteria that mark two-stage systems are prepackaged with compound compressors. They contain an interstage desuperheater and a liquid subcooler, both within the same heat exchanger, and use some condenser liquid via a thermal expansion valve to accomplish both desuperheating and subcooling. The intercooler is mounted between the low-stage cylinder discharge and the high-stage suction. The booster discharge is typically cooled to within about 10°F of the interstage saturation temperature. The high-pressure liquid is also subcooled to within about 10 to 15°F of the interstage saturation. This is similar to the operation of a standard two-stage refrigeration system with two compressors.

1.62 The basic difference, except for size, is that the interstage pressure is determined by (1) the ratio of low-stage to high-stage cylinders per compressor and (2) the absolute low-side suction pressure. It is not normally possible to introduce additional high-side loads into the intermediate pressure (high-stage suction). Six- and twelve-cylinder compressors typically have a basic 2:1 ratio, with cylinder ratios of 4:2 and 8:4 respectively. Eight- and sixteen-cylinder compressors are typically divided into a 3:1 ratio with 6:2 and 12:4 low:high cylinders.

1.63 The crankcase of the compressor is normally equalized to the suction (low-stage) pressure. This permits the low-stage cylinders to operate much the same as a single-stage compressor. For the high-stage cylinders, however, the load on the wrist pin and crankpin bearings is considerably higher, because even the lower intermediate pressure at the intake of the high-stage cylinders is above the crankcase (suction) pressure. The force on the piston and connecting rod is continually down toward the shaft at all times. It becomes necessary to use pressure-lubricated needle or roller bearings at the small end of the connecting rod to obtain adequate bearing life.

1.64 Figure 1-12 on the following page shows a compound compressor. The illustration also shows the four possible intermediate cooling systems for this compressor.

Fig. 1-12. Sabroe compound compressor



1-9. Compressor capacity control is achieved by means of loading and unloading cylinders, usually in response to ________ pressure.	1-9. SUCTION Ref: 1.40
1-10. No compression takes place on the upstroke in a(n) ________ cylinder.	1-10. UNLOADED Ref: 1.42
1-11. All ammonia reciprocating compressors should have a(n) ________ installed to ensure that oil fed to the bearings is _______.	1-11. OIL FILTER; CLEAN Ref: 1.46
1-12. Crankcase oil heaters, which work only when the compressor is ________, usually warm the oil to about ________°F.	1-12. NOT RUNNING; 120 TO 140 Ref: 1.49
1-13. The most useful efficiency term is the performance factor, the relationship between actual power in ________ to the compressor capacity rating in ________.	1-13. BHP; TONS Ref: 1.50, 1.55
1-14. Limits of compressor operation are provided in graphs and tables from manufacturers' ________ or ________.	1-14. LITERATURE; COMPUTER DISKS Ref: 1.56
1-15. A compound compressor requires ________ control panel(s) and ________ motor(s).	1-15. ONE; ONE Ref: 1.60
1-16. The booster discharge of a compound compressor is typically cooled to within about ________°F of the interstage saturation temperature.	1-16. 10 Ref: 1.61

Answer the following questions by marking an "X" in the box next to the best answer.

1-1. Force-fed lubrication to the compressor wear elements became possible with the addition of

- ☐ a. lift pins
- ☐ b. packing glands
- ☐ c. the oil pump
- ☐ d. the shaft seal

1-2. All industrial reciprocating refrigeration compressors are equipped with a(n)

- ☐ a. oil cooler
- ☐ b. oil filter
- ☐ c. oil-supply sump
- ☐ d. suction strainer

1-3. For reciprocating compressors, the minimum differential oil pressure is usually about

- ☐ a. 10 to 15 psi above suction pressure
- ☐ b. 10 to 15 psi below discharge pressure
- ☐ c. 30 to 40 psi above suction pressure
- ☐ d. 30 to 40 psi below discharge pressure

1-4. The devices that provide cylinder sealing to prevent oil carryover are the

- ☐ a. piston rings
- ☐ b. safety head springs
- ☐ c. suction valve plates
- ☐ d. unloader elements

1-5. The suction valve is always larger than the discharge valve because the inlet gas is

- ☐ a. at a higher flow rate
- ☐ b. at a higher pressure
- ☐ c. less dense and takes up more volume
- ☐ d. more dense and takes up more volume

1-6. The discharge valve opens when the piston

- ☐ a. approaches bottom dead center
- ☐ b. approaches top dead center
- ☐ c. begins the downstroke
- ☐ d. begins the upstroke

1-7. When lift pins lift the suction valve ring off its seat, the cylinder is ________ and the valve is ________ during the upstroke.

- ☐ a. loaded; closed
- ☐ b. loaded; open
- ☐ c. unloaded; closed
- ☐ d. unloaded; open

1-8. Suitable viscosity for lubricants used with ammonia reciprocating compressors is ________ SUS or ISO VG ________.

- ☐ a. 20; 60
- ☐ b. 60 to 65; 325
- ☐ c. 225; 100
- ☐ d. 300 to 350; 60

1-9. Which of the following is the equation for the most practical way to compare the suitability of various compressors for an application?

- ☐ a. CR = discharge pressure/suction pressure
- ☐ b. N_a = isentropic power/shaft input power
- ☐ c. PF = actual bhp/rated capacity in tons
- ☐ d. VE = Acfm/Dcfm

1-10. Which of the following components are required on compound reciprocating compressor connecting rods because of additional forces resulting from higher pressures?

- ☐ a. Journal bearings
- ☐ b. Needle or roller bearings
- ☐ c. Safety head springs
- ☐ d. Thrust collars

SUMMARY

Early ammonia reciprocating compressors were slow-speed vertical or horizontal single-acting machines with either belt or direct drive, followed by enclosed HDI or VSA compressors. Later improvements included shaft seals, oil pumps, and capacity-control systems, followed by reduced piston size and stroke and increased running speeds and number of cylinders. Today's ammonia reciprocating compressors typically have 2 to 16 cylinders, an oil-supply sump, an oil pump, bearings, shaft seals, pistons with varying numbers of compression and oil-scraper rings, sometimes safety head springs, usually a suction strainer and oil filter, and often an oil cooler. The suction valve is always larger than the discharge valve.

A typical capacity-control method works by loading and unloading cylinders as oil pressure is applied and removed. Almost all compressors have a minimum number of cylinders loaded at all times, which prevents operation with no flow through the compressor and also reduces starting torque. Reciprocating compressor lubrication is achieved by means of a crankshaft-driven oil pump. The oil filtration system removes grit. Most non-booster ammonia compressors provide for oil cooling. Lubricants have a viscosity of 300 to 350 SUS (ISO VG 60). A crankcase heater warms the oil when the compressor is not running.

Three efficiency terms are used with positive-displacement compressors—volumetric efficiency (VE), adiabatic efficiency (N_a), and performance factor (PF). PF, the actual bhp divided by the rated capacity in tons, is most useful for comparing the operation of various compressors. The lower the PF, the more energy-efficient is the selection. Also, manufacturers provide compressor application data.

Most compressors are provided as part of a compressor unit, which includes the base, drive, controls, and other components. Compound compressors contain both booster and high stages in one housing, along with a desuperheater/subcooler, thermal expansion valve system, and intercooler.

Answers to Self-Check Quiz

1-1. c. The oil pump. Ref: 1.09

1-2. c. Oil-supply sump. Ref: 1.20

1-3. c. 30 to 40 psi above suction pressure. Ref: 1.21

1-4. a. Piston rings. Ref: 1.28

1-5. c. Less dense and takes up more volume. Ref: 1.32

1-6. b. Approaches top dead center. Ref: 1.34

1-7. d. Unloaded; open. Ref: 1.42

1-8. d. 300 to 350; 60. Ref: 1.48

1-9. c. PF = actual bhp/rated capacity in tons. Ref: 1.50, 1.55

1-10. b. Needle or roller bearings. Ref: 1.63

Contributions from the following sources are appreciated:

Figure 1-1.	Frick Company
Figure 1-2.	Frick Company
Figure 1-3.	Frick Company
Figure 1-4.	Vilter Manufacturing Corp.
Figure 1-5.	Mycom America Corp.
Figure 1-6.	Mycom America Corp.
Figure 1-8.	York International; Mycom America Corp
Figure 1-9.	Vilter Manufacturing Corp.
Figure 1-10.	Vilter Manufacturing Corp.
Figure 1-11.	York International
Figure 1-12.	Sabroe Refrigeration A/S