7

Gas & Process

Chemical and petroleum processes have resulted largely from the ingenuity of the people who originated them. Ingenuity is defined as skill or cleverness in devising or combining. In this chapter it is hoped that a real appreciation of such ingenuity will be developed. One hundred and eight basic elements are available in nature. There are a few basic rules for combining these elements into compounds, all well defined in organic and inorganic chemistry Pressure and temperature control and the use of catalysts have let us manufacture the many chemical products that are now on the market.

An interesting observation, as an example, is that a certain process initially devised to produce a certain compound required very high reactor pressures. Consequently, the designers of gas compressors were challenged to produce equipment that would deliver the new component gas to a reactor at 40,000 psi. Twenty or thirty years later the same product required only one-hundredth of the original pressure. Nevertheless, compressor manufacturers have met and will continue to meet new challenges of many kinds in the design of compressors to supply gas at the required pressure, twenty-four hours per day, with a minimum of interruption imposed on the process by the compressor equipment.

The proper selection of the type of compressor for the service should be carefully analyzed. Compare past experience, efficiency at all operating flows and pressures, cost of installation, and compatibility between the compressor design and the gas to be compressed. After all, it is the product of a plant produced at the required rate that justifies the plant's existence.

The purchaser must accurately define the composition of the gas to be compressed, including any entrained liquids. Vapors that will condense out during the compression cycle must be part of the gas analysis. In many instances the gas analysis will vary, and this variation must be defined. Any characteristic of the gas that would influence the reliability of the compressor must be studied, and any known designs for similar problems should be investigated. For instance, chlorine and hydrogen chloride are successfully compressed, but the gas must be void of water and free of solids. Coke-oven gases can be successfully compressed, but in most cases require scrubbers to remove tarry substances first, and then static electric precipitators to remove fly ash.

Compressors handling gases other than air are broadly divided into two groups: process gas compressors and oil and gas field compressors. Process gas compressors are employed extensively in petroleum refining and in the chemical industry. Oil and gas field compressors have somewhat different characteristics from process gas compressors and are used throughout the oil and natural gas industry for producing oil and natural gas and in various phases of the treatment and transport of natural gas.

A relatively new use for gas compressors is in tertiary recovery and enhanced oil recovery. High energy prices, declining reserves, escalating exploration costs, pricing, and tax legislation have created incentives for enhanced oil and gas recovery. In this application, compressors are used to transport carbon dioxide, nitrogen, or hydrocarbon gases through pipelines for injection into wells.

In this chapter, typical applications of compressors in process services and in the oil and gas industry are discussed. These applications illustrate the potential and flexibility of gas compressors and demonstrate that the modern compressor has the capability to provide virtually any type of gas-compression service.

OIL REFINERY PROCESSES

Refineries are a part of many large cities and communities, such as Los Angeles, San Francisco, and Chicago. The cleanliness of the environment around refineries is improving and every effort is being made to reduce pollutants. Hydrocarbon vented gases in the refinery are collected, compressed to the necessary pressure, treated, and used in the refinery fuel gas system.

The raw material of crude oil contains polluting substances such as sulfur and chlorides, including salt. The modern refinery has processes that extract sulfur commercially, thus producing a needed chemical and, at the same time, reducing emissions into the atmosphere. Figures 7.1 and 7.2 outline the general flow and functions of the refinery processes.

The initial process plant in all refineries is the crude oil distilling unit. Here the crude oil is heated to its boiling point and separated by means of fractionating columns into various fractions from heavy to light (Fig. 7.1).

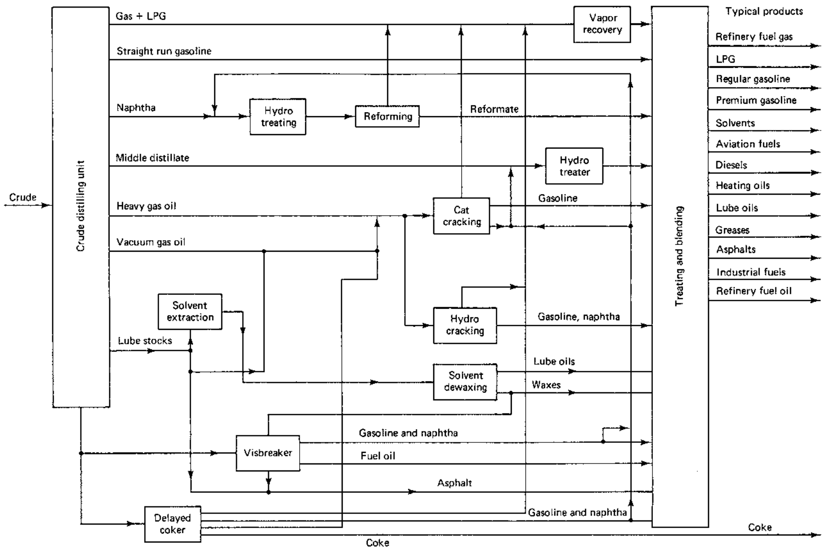

Figure 7.1 Flow diagram for a refinery.

Refinery Processes

1. Breaking Hydrocarbons Down
   Hydrogenation – Adding Hydrogen
   a. Thermocracking
      1. Coking (Delayed) – Flexicoking
      2. Visbreaking
   b. Catalytic Cracking
   c. Hydrocracking Fixed Bed Catalytic Process
   d. Hydrotreating
2. Building up Hydrocarbons
   a. Polymerization – Gaseous Olefins to Liquid Hydrocarbon
   b. Alkylation – Gases to Liquid
3. Modifying – Altering – Upgrading – Rearranging – Dehydrogenation
   a. Reforming
   b. Isomerization – Low Octane Liquid to High Octane Liquids
4. Separating Hydrocarbons – Light Ends Recovery
   a. Extraction – Physical Separation
   b. Distillation
      1. Atmospheric
      2. Vacuum

5. Cleaning Hydrocarbons
    a. Desalting
    b. Treating to Reduce H_2S, Acids, Etc.
        1. Hydrodesulfurization (HDS)
    c. Hydrotreating to Reduce Unsaturated Hydrocarbons
        1. Catalytic Hydrofining

Figure 7.2 All of these refinery processes utilize compressors.

Heavy Fractions

The heaviest fractions are boiled under a vacuum to make a product suitable for the manufacture of coke in the delayed coker. These heavy fractions are also used for heavy fuel oils, such as No.6 oil.

Gas Oil Fractions

The gas oils from the crude unit are feed stocks for gasoline and diesel fuel. These oils are sent to the catalytic cracking unit and the hydrocracking unit for making various forms of gasoline and light fuel oil and diesel oil. The fractionating columns in these plants separate these products after the cracking reaction takes place so that they can be segregated for final treatment.

Naphtha Fractions

The light naphtha components from the crude unit are sent to the reformer unit for making gasoline with a high-octane number. This product is mixed with gasoline from the catalytic cracking unit and hydrocracker to obtain the desired octane numbers of the various grades of gasoline: regular, premium, lead free, and so on. The alkylation unit also produces a high-octane component for blending into gasoline.

Liquefied Petroleum Gas (LPG) Fractions

The lightest fractions from the crude unit are propane and butane, which are used for LPG or bottled gas. They are mixed with the lighter components from the other process units to form the final products.

Lube-base Stocks

Only certain types of crude oil are suitable for manufacturing lubricating oil. The lube stock is sent to a solvent dewaxing process where it is mixed with methyl ethyl ketone, which dissolves the wax contained in the oil. The mixture of lube stock and solvent is refrigerated to from 0°F to -30°F, as it passes through tubes that

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have a rotating scraper. The outside of the tube is the evaporator of the refrigeration system. The cooled solution causes the wax to solidify, and the mechanical scrapers keep the solidified wax mixed with the oil. The lube stock then is pumped to a large rotating filter. The filter drum is constructed with radial segmented chambers. As the drum turns, the filter cloth is alternately pressurized or is exposed to a vacuum of 23 in. of mercury and then a positive pressure of 4 psig. As the tilter cloth passes the vacuum segment, the wax is extracted from the oil and is held to the cloth by its vacuum. As the drum turns the pressurized segment, the wax is blown off the cloth and removed from the system, and the oil is purified and wax free. A typical installation of this solvent dewaxing system uses horizontal, heavy-duty reciprocating ammonia refrigeration compressors. Practically all refining processes utilize centrifugal or reciprocating compressors.

It is important to note that each refining unit has a unique processing scheme determined by the equipment available, operating costs, and product demand. The optimum flow pattern for any refinery is dictated by economic considerations, and no two refineries are identical in their operations. The preceding description, however, will give some idea of the general nature of refinery processing.

Distillation of Crude Oil

Distillation of crude oil is the first major processing step in the refinery. It is used to separate the crude oil by distillation into fractions according to boiling point so that each of the downstream processing units will have the feedstocks that they require to meet the product requirements of the refinery. By changing the amount of heat and the temperature levels for the distillation process, variable boiling point materials can also be produced.

In many cases, the distillation is accomplished in two steps:

1. By fractionating the total crude supply at atmospheric pressure, a process called atmospheric distillation.
2. By feeding the heaviest fraction, called topped crude, from the atmospheric distillation to a second fractionator operating at a high vacuum, a process called vacuum distillation.

Atmospheric Distillation

Figure 7.3 shows the atmospheric distillation unit. The crude is normally pumped to the unit from storage at ambient temperature. After being heated in exchangers, it enters the desalting vessel at 250°F. Most crudes contain salt. The salt must be removed to minimize the fouling and corrosion caused by salt deposition on heat-transfer surfaces.

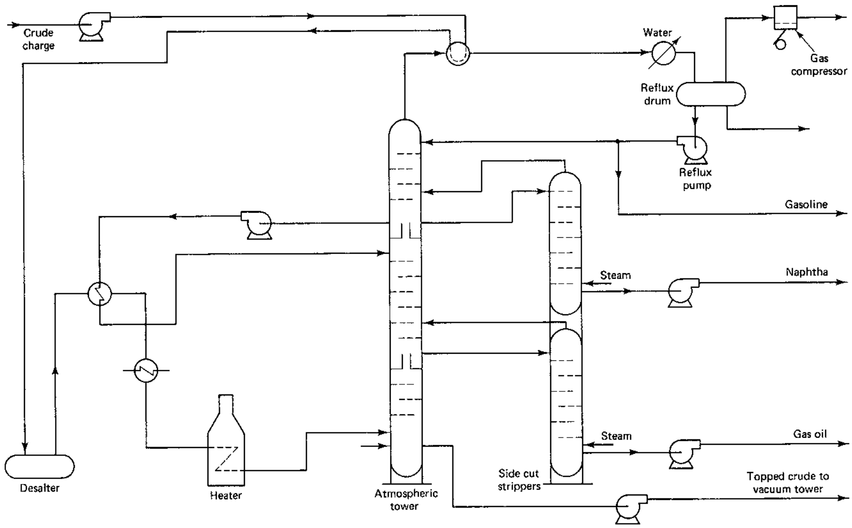

Figure 7.3 Atmospheric distillation unit for crude oil.

Desalting is carried out by emulsifying the crude oil with water. The salts are dissolved in the water, and in the water and oil phases, separated by using chemicals to break the emulsions or by developing a high potential electric field across the settling vessel to coalesce the water droplets.

The desalted crude is then heated by heat exchange from hot product and then further heated in a direct-fired furnace to about 750°F. At this temperature, it enters the flash zone of the fractionating column, where separation into fractions or “cuts” takes place. The fractionating column is a tall cylindrical vessel fitted with a large number of specially designed trays.

The mixture of vapor and liquid enters the fractionator near the bottom and rises gradually through the opening in the trays. As it rises, it cools, and a certain amount condenses on each tray until the tray is full of liquid up to the level of the overflow.

The level on each tray is kept just above the holes in the bubble caps so that all the vapor has to pass through the liquid. Each tray is a little cooler than the one below it, and lighter and lighter products will be present on each succeeding tray as the vapor passes up through the column. As the vapor bubbles through the liquid on the trays, that part of the vapor will condense that has the same boiling point as the liquid on the tray.

The temperature throughout such a column is controlled at the bottom by the furnace, which heats the incoming crude oil, and at the top by pumping back a certain amount of material, which leaves the top of the column after condensing. This material pumped back is called reflux.

In this way, by controlling the temperature at the top and bottom, the temperature variation throughout is kept under control so that the temperature of the trays varies gradually from the bottom to the top. The amount of liquid pumped back to the top of the column can be varied as required, in order to give the correct temperature at the top. This, in turn, controls the final boiling point of the gasoline leaving the top. Reflux can also be withdrawn and pumped back into the column at intermediate levels below the top for better control of temperature and distillation.

The atmospheric fractionation normally contains 30 to 50 fractionation trays. Side drawers are located to obtain the desired boiling range products from the fractionator; that is, gasoline off the top, topped crude off the bottom, and intermediate cuts, such as naphtha, gas, oil, and kerosene in between.

The liquid sidestream withdrawn from the tower will contain some material with a lower boiling point than the bulk of the material on the tray. These light ends are stripped from each sidestream in a separate, small stripping vessel containing four to five trays. As the liquid flows down over these trays, it meets an upward stream of steam, injected at the bottom of the stripper. The steam boils off the light ends, and this narrows the boiling range of the bulk of the liquid drawn off the side of the column.

From the side cut strippers the various liquids are pumped to storage or to the next downstream process-naphtha to the catalytic (cat) cracker or hydrocracker. The gasoline off the top of the column is treated and pumped to the gasoline pool for blending with other gasolines. Usually, at least four side strippers are provided in a plant to produce extra cuts, such as kerosene and diesel oil.

Some atmospheric crude units require a compressor to handle the gases from the reflux drum at the top of the column. As the gasoline vapors from the top of the column condense, the lightest vapors remain in the gaseous state and accumulate above the liquid level in the reflux drum. A compressor is required to pump this gas to the gas-treating unit for final disposition, either to fuel gas or other product. Because of the relatively small volumes, a reciprocating-type compressor is utilized and is called the vent-gas compressor. The molecular weight of this gas is 60 to 70 and it is normally pumped up to 100 to 200 psig pressure level.

Hydrocracking

Hydrocracking is a refinery process for making gasoline out of heavier feedstocks from the crude unit. The interest in hydrocracking was caused by two factors:

1. Economic demand for fewer, more efficient refineries to produce gasoline with an increase in the ratio of barrels of gasoline per barrel of crude oil.
2. Large quantities of by-product hydrogen were available from the many cat reformers that had been built, and hydrocracking requires a large amount of hydrogen.

In a number of refineries, cat cracking and cat re-forming work together. The cat cracker takes the more easily cracked oils as feed, while the hydrocracker can crack those oils that are not easily cracked in a cat cracker.

Although they work as a team, the processes are completely dissimilar. The cat cracker reaction takes place at a low pressure, 25 to 35 psig without hydrogen, and in the presence of a fluidized bed of catalyst. Hydrocracking occurs at a pressure of 1000 to 2000 psig, with a high concentration of hydrogen and in a fixed bed of catalyst.

The hydrocracking process is flexible. In addition to making gasoline from middle-distillate oils, the process is used to make distillates and light oils from residual oil. Of course, the catalyst and operating conditions are different; however, the same plant can be designed to operate in the alternate modes.

As in cat re-forming, the feedstock must be hydrotreated prior to entering the hydrocracker to remove the sulfur, nitrogen, and oxygen compounds, which are harmful to the catalyst. Thus, a hydrotreater may be required in conjunction with a hydrocracker, unless hydrotreating capacity exists within the refinery. In a two-stage hydrocracker, the first stage may perform the function of a hydrotreater.

A number of hydrocracking processes are available for licensing:

Isomax	Chevron and Universal Oil Products
Unicracking/SHC	Union and Exxon
H-G Hydrocracking	Gulf and Houdry
Ultracracking	Amoco
Hy-C, H-Oil	Hydrocarbon Research and
Shell	Cities Service

With the exception of H-Oil and Hy-C processes, all hydrocracking processes in use today are fixed-bed catalytic processes with liquid downflow through the reactors.

The hydrocracking process contains the most sophisticated rotating and reciprocating machinery that is found in refineries or petrochemical plants. Equipment for hydrocrackers can be classified as follows:

1. Barrel-type centrifugal compressors for hydrogen recycle service, 2000 psi pressure.
2. Large, balanced, opposed reciprocating compressors for hydrogen makeup, 2000 psi pressure.

Process Description

Figure 7.4 shows a typical two-stage hydrocracker. Reaction temperature is approximately 800°F and pressure in each reactor system is about 2000 psig. The fresh feed is pumped up to 2000 psi from the crude unit or cat cracker at elevated temperature of 600°F or higher. The feed mixes with makeup hydrogen, which is compressed into the plant from the hydrogen plant or the cat reformer. Since hydrogen is consumed in the hydrocracking reaction, it must be continuously added to the system. In addition, recycle hydrogen is mixed with the feed to supply the hydrogen atmosphere required by the reaction.

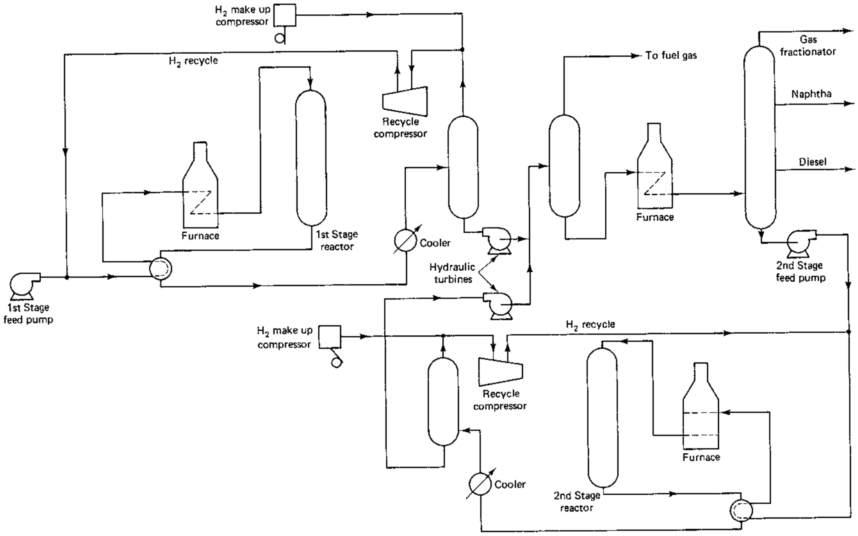

Figure 7.4 Two-stage Hydrocracker

The hydrogen recycle stream is circulated through the system by the recycle compressor, a harrel-type centrifugal machine. The mixture of feed, recycle hydrogen, and makeup hydrogen is heated in a furnace prior to entering the catalyst beds in the reactor.

From the first-stage reactor, the effluent travels through a bank of exchangers into a high-pressure separator where the hydrogen recycle gas is separated from the liquid product and fed to the recycle compressor for recirculation back through the first-stage reaction system. The liquid product from the high-pressure separator is let down through a hydraulic turbine into a low-pressure separator. The hydraulic turbine is utilized to drive the feed pumps.

From the low-pressure separator, the product is pumped to a fractionation column, where the gasoline comes off the top and is pumped to storage. The heavier fractions from the bottom of the column are pumped into the second-stage reaction system as feed.

The second stage is similar to the first, except that it operates at higher temperatures in order to crack the unconverted oil from the first stage. The second-stage product is combined with the first-stage product prior to fractionation. Thus, the second stage handles first-stage product plus some recycle of its own product.

Hydrogen Make-up Compressor

Make-up hydrogen is required in large quantities and in a relatively pure state (i.e., molecular weight of 2 to 3). It is usually supplied from two sources, hydrogen plant and a cat re-former, both at pressures in the vicinity of 200 psi. Thus, the hydrogen has to be boosted to a 2000 psi hydrocracker pressure level, and reciprocating compressors are normally required for the job.

Hydrogen consumption is about 200 scf (standard cube feet) per barrel of feed. Thus, on a 20,000 bpd (barrel per day) hydrocracker, 40 million standard cubic feet per day (scfd) of hydrogen must be compressed into the hydrocracker. For the same 20,000 bpd barrel per day unit, the horsepower required could be as high as 18,000, or 9000 for each 50% machine.

The following table indicates the horsepower required for typical hydrogen makeup to various-sized units, assuming two 50% machines:

Hydrocracker Capacity, bpd	Makeup Hydrogen, scfd	Approximate Compression Horsepower, bhp
5,000	10,000,000	5,000
10,000	20,000,000	9,000
20,000	40,000,000	18,000
30,000	60,000,000	27,000

The balanced, opposed compressor (Fig. 7.5) is the type required for this service. There have been different types of compressor arrangements for supplying the hydrocrackers with hydrogen.

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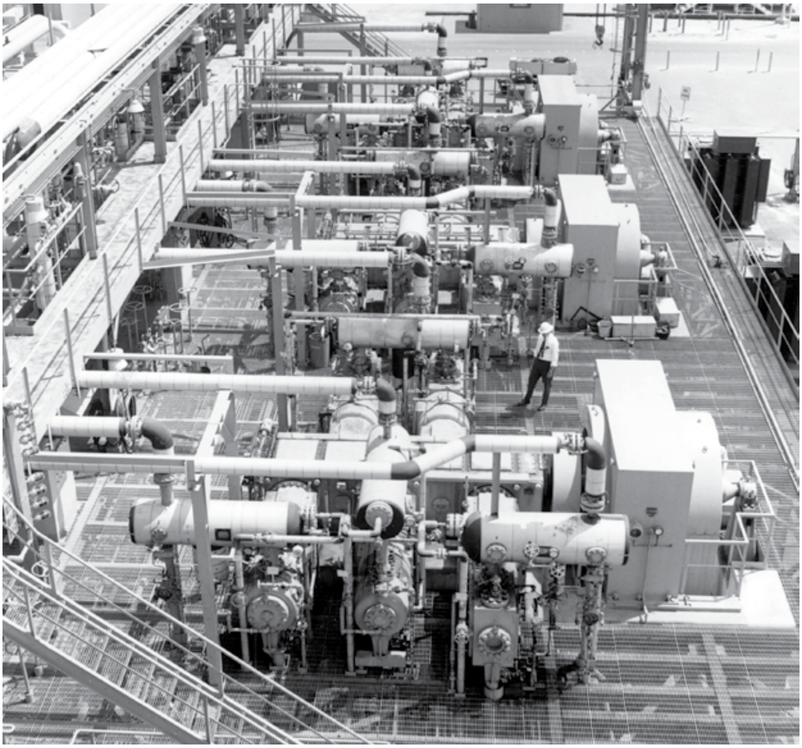

Figure 7.5 Three 5400 hp reciprocating compressors on hydrocracking gas service. The compressors are driven by engine type sychronous motors. The discharge pressure is 2873 psig.

Several plants have centrifugal compressors located within the hydrogen plant upstream of the CO_2 removal equipment. At this point, the gas has a molecular weight of 18, and two barrel compressors in series can reach 2000 psi pressure level. The CO_2 is removed at the higher pressure level and the hydrogen leaves the hydrogen plant in pure form at 2000 psi.

Hydrogen Recycle Compressor

The hydrogen recycle compressors are barrel-type centrifugal machines similar to cat re-former machines except for the higher pressure ratings of 2000 psi.

Catalytic Re-forming

The demand of today's automobiles for high-octane gasoline, without the addition of lead, has stimulated interest in catalytic (cat) re-forming. The major function of the cat re-former is to produce a high-octane product that, when blended with other gasoline streams from the cat cracker and hydrocracker results in an overall gasoline octane number within market specifications. As lead is legislated out of gasoline, more cat re-forming capacity is required to make up for the octane rating improvement obtained with lead.

In catalytic re-forming, the hydrocarbon molecules are not cracked, but their structure is rearranged to form higher-octane products. The reaction resulting in the molecular rearrangement takes place in a series of three or four reactors at a temperature of 900°F to 1000°F in the presence of a metallic catalyst containing platinum and in a hydrogen atmosphere. The feed stock is a straight-run gasoline or naphtha from the crude distillation unit. The feedstock is hydrotreated prior to entering the cat re-former to remove the sulfur, which is harmful to the platinum catalyst. Thus, a hydrotreating plant is usually built along with a cat re-former, and this unit will be described in another section.

There are several proprietary catalytic re-forming processes, all with somewhat similar operating conditions and all using some type of platinum catalyst. These include the following:

1. Platforming (Universal Oil Products)
2. Powerforming (Exxon)
3. Ultraforming (Amoco)
4. Houdriforming (Houdry)
5. Catalytic re-forming (Englehard)
6. Rheniforming (Chevron)

These processes utilize large, barrel-type, centrifugal compressors. There are approximately 1000 catalytic re-formers such as this throughout the world and many more are proposed as the lead phase-out occurs.

Equipment for catalytic re-forming can be classified as follows:

1. Barrel-type centrifugal compressors for hydrogen recycle service.
2. Barrel-type centrifugal compressors for hydrogen makeup service.
3. Single-stage overhung or horizontally split centrifugal compressors for regeneration gas circulation during catalyst regeneration.

Process Description

For illustration, the platformer process of Universal Oil Products is used for the process description in Fig. 7.6. Although operating conditions are slightly different, the other cat re-forming processes are similar in principle.

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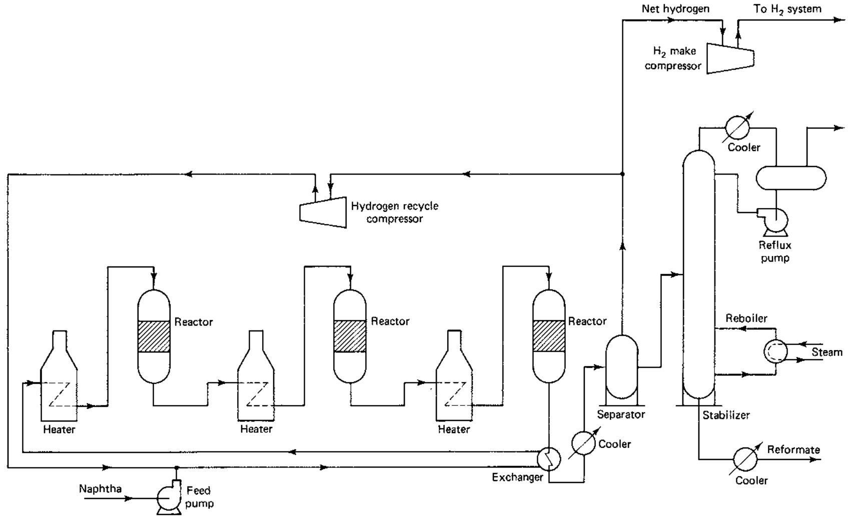

Figure 7.6 Catalytic Reforming Unit

The reaction takes place in three reactors filled with the platinum-bearing catalyst pellets. The mixture of hot oil and hot hydrogen is recycled through the beds to effect the molecular rearrangement. The reactors are steel vessels lined inside with refractory to insulate the metal from the 1000°F reaction temperature.

The process flow is as follows: the pretreated feed and recycle hydrogen are heated to a temperature of from 900 to 1000°F before entering the first reactor. In the first reactor, the major reaction is dehydrogeneration of naphthenes to aromatics and, as this is strongly endothermic, a large drop in temperature occurs. To maintain the reaction rate, the gases are reheated before being passed over the catalyst in the second reactor. Usually, three reactors are sufficient to provide the desired degree of reaction, and heaters are needed before each reactor to bring the temperature up to the desired reaction temperature.

As the mixture of product and hydrogen leaves the last reactor, it is cooled, and the hydrogen is separated from it in the hydrogen separator. The liquid products are condensed and sent on for further processing in the stabilizer column, where high-octane reformate (final product from the re-former) is accumulated and pumped to the gasoline pool.

The hydrogen leaving the hydrogen separator splits into two streams; one is recycled through the process to mix with feed going to the reactors, and the excess hydrogen is pumped away by compressors to be used in other processes such as hydrocracking. Since hydrogen is manufactured in the cat re-former process, there is always a sizable stream leaving the unit. It is one of the advantages of cat re-forming, since hydrogen is an expensive product.

Unfortunately, the catalyst deteriorates over a period of time and has to be regenerated. There are three types of regeneration.

1. *Semiregenerative:* Here, the plant is shut down after a 6-month to 2-year run, depending on the severity of operating conditions, and the catalyst is regenerated by high-temperature oxidation and chlorination. The hydrogen recycle compressor is used in this regeneration to pump air through the reactors to furnish oxygen for the regeneration reaction.
2. *Cyclic regenerative:* In the cyclic process, an extra reactor is installed in the train and it is used as a swing reactor. Every 24 to 48 hours, one reactor is valved out of service and the catalyst in it is regenerated. In this type of plant, a separate compressor is installed for circulation of air for regeneration. In many plants, a single-stage overhung centrifugal compressor is utilized.
3. *Continuous regenerative:* A development by Universal Oil Products features continuous regeneration. The reactors are stacked one on top of the other and the catalyst is continuously circulated; regeneration occurs almost continuously in a separate regenerator vessel. Again, a separate compressor is used for regeneration. It is a single-stage unit, but it handles gases at 900°F and requires special metallurgy.

Hydrogen Recycle Compressor

The hydrogen recycle compressor is the heart of the reaction system since it circulates the large quantities of hydrogen through the reactors and furnaces. The recycle compressor is a barrel-type compressor with approximately six or more wheels.

The first unit operated with pressure at approximately 1000 psi in the reactors, but, at present, the pressure level has been decreased to a level between 150 and 250 psig. The compressor differential pressure is between 100 and 150 psi, and the molecular weight of the recycle gas, which is about 90% hydrogen and 10% hydrocarbons, is from to 5 to 8. The polytropic head on a recycle compressor is usually less than 100,000 ft.-lbs/lb.

To ensure against hydrogen leakage, the barrel-type casing is used. Seals are of the oil-film type, with the small quantity of oil leakage discarded via the high-pressure oil trap in the seal oil system.

Drivers are electric motors or steam turbines. Since control of the recycle gas stream is not critical, a single suction throttling system on the motor unit and hand-set variable speed on the turbine are adequate. Small variations in recycle flow do not affect the reactions that occur. Of course, the less recycle, the lower the power consumption for the process.

There is some variation in molecular weight with time as the catalyst deteriorates between regeneration, especially on semiregenerative units. Care must be taken to make sure the minimum molecular weight is considered in establishing compressor head requirements and the maximum molecular weight in calculating the bhp for driver sizing. On these units, the variable-speed turbine is an energy saver. Suction throttling on the motor unit, unfortunately, is not efficient at lower system head requirements associated with high molecular weights.

On the cyclic regeneration systems, where the swing reactor is regenerated daily, the molecular weight stays fairly constant and there is little choice between motor and turbine from the standpoint of energy consumption.

The following is a tabulation of compressor sizes and types for various-sized cat re-formers. The units are based on a molecular weight of about 8, a suction pressure of 125 psia, and a discharge pressure of 250 psia. These conditions will vary for the different types of cat re-formers. The recycle rate on most cat re-formers varies between 4000 and 6000 standard cubic feet of gas per barrel of feed depending on process conditions.

Cat Re-former Capacity, bpd	Recycle Compressor, acfm	Approximate Compression Horsepower, bhp
5,000	2,500	2,500
10,000	5,000	3,500
20,000	10,000	6,000

Cat Re-former Capacity, bpd	Recycle Compressor, acfm	Approximate Compression Horsepower, bhp
30,000	15,000	8,500
40,000	20,000	12,000
50,000	25,000	14,500

On small units, reciprocating compressors are applied and electric-driven, horizontal-balanced opposed units are utilized. Two compressors are usually furnished.

Hydrogen Make-up Compressor

The hydrogen make-up compressor transfers the excess hydrogen from the system of the hydrotreaters and other processes that use hydrogen. The amount of hydrogen that is made varies between processes but, in general, is about 1200 standard cubic feet of hydrogen per barrel of feed. Thus, on a 35,000 bpd unit, 40 mm scfd of hydrogen must be compressed for transportation to other plants.

The design of the make-up compressor is similar to that of the recycle unit, but the unit is of a smaller size. The following table estimates the compressor size and types for various-sized cat re-formers. It is assumed that the compressor suction pressure is 230 psia. Horsepowers are not shown because they depend on discharge pressure, which varies for each plant. Usually, nine or ten wheels are required for the compressor.

Cat Re-former Capacity, bpd	Hydrogen Make-up Compressor, acfm	Compressor Designation
5,000	375	Reciprocating
10,000	750	Reciprocating
20,000	1500	Centrifugal
30,000	2200	Centrifugal
40,000	3000	Centrifugal
50,000	3750	Centrifugal

Regeneration Recycle Compressor

In cyclic regenerative plants, the regenerator recycle compressor is usually a single-stage, centrifugal type of unit. Following is a rough estimate of the sizes of units for various cat re-former capacities. The gas handled by the compressor is, for the most part, air with an equivalent molecular weight of 28. Suction pressure is about the same as the recycle compressor suction pressure, and the increase in pressure through the compressors is about 50 psi to satisfy the system pressure drop.

Cat Re-former Capacity, bpd	Regeneration Compressor, acfm	Recycle Compressor Designation
10,000	1250	Centrifugal
20,000	2500	Centrifugal
30,000	3750	Centrifugal
40,000	5000	Centrifugal
50,000	6250	Centrifugal

Hydrotreating

Hydrotreating (Fig. 7.7) is a process for removing objectionable elements from products or feedstocks by reacting them with hydrogen in the presence of a catalyst. The objectionable elements include sulfur, nitrogen, oxygen, halides, and trace metals. Hydrotreating is also referred to as hydrodesulfurization (HDS).

Hydrotreating serves two major purposes in a refinery.

1. It removes impurities, such as sulfur, nitrogen, oxygen, halides, and trace metals, from marketable products, such as fuel oils, distillates, and lube oils.
2. It removes these same impurities from feedstocks to cat crackers, cat re-formers, and hydrocrackers. Since these impurities are detrimental to the catalysts in these processes, hydrotreating plays a vital role in refinery production.

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Many refineries have a number of hydrotreaters to perform the preceding functions. With environmental regulations dominating industrial processing, hydrotreaters will continue to be in demand.

Although a large number of hydrotreating processes are available for licensing, most of them have similar process flow characteristics and the flow diagram in Fig. 7.7 illustrates a typical process.

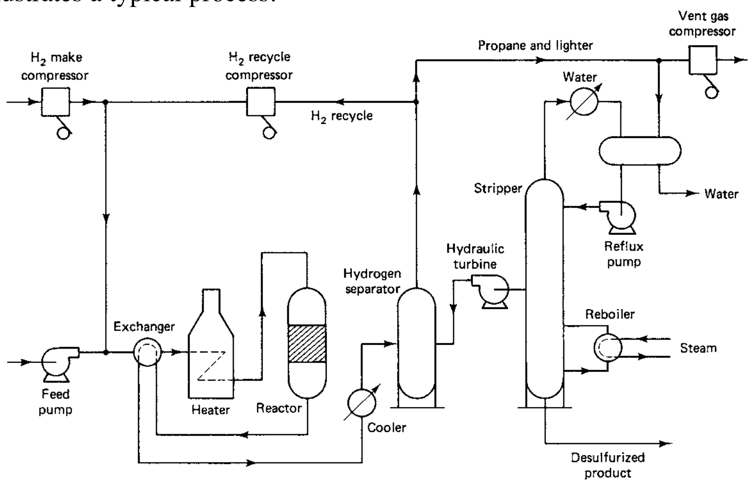

Figure 7.7 Hydrotreater

The charge pump pressurizes the feedstock up to about 1200 psi. The feedstock joins a stream of hydrogen recycle gas, and the mixture of hydrogen and oil is heated by exchangers in a direct-fired heater up to the reaction temperature of 700 to 800°F. The hot mixture enters the reactor, which is a pressure vessel with a fixed bed of catalyst. In the presence of the catalyst, the hydrogen reacts with the oil to produce hydrogen sulfide, ammonia, saturated hydrocarbons, and traces of metals. The trace metals remain on the surface of the catalyst, while the other impurities leave the reactor with the hydrogen-oil mixture. This mixture leaving the reactor is known as reactor effluent.

The reactor effluent is cooled before entering the high-pressure separator where the hydrogen-rich gas comes off the top and the oil off the bottom. The hydrogen-rich gas is treated to remove hydrogen sulfide and is used again in the process. It is recycled into the front end of the plant by the recycle compressor.

The oil from the bottom of the separator is throttled to a lower pressure and enters the stripper, where any remaining impurities are removed. In some plants a hydraulic turbine replaces the throttling valve.

The reaction consumes hydrogen at the rate of about 500 scf per barrel of feedstock. This figure varies with the process and with the amount of impurities that are to be removed. It can be as low as 200 or as high as 800 scf per barrel. The quantity of hydrogen-rich gas that is recycled is about 200 scf per barrel of feedstock.

There are three compressor applications on most hydrotreaters:

1. Recycle hydrogen-rich gas, molecular weight 6 to 8, with suction pressure of 1000 psig and discharge pressure of about 1150 psig.
2. Hydrogen makeup gas, molecular weight 2 to 6, suction pressure of 200 psig, and discharge pressure of 1150 psig.
3. Vent gas, small quantities at low suction and discharge pressure, with molecular weight varying from 20 to 40.

Quantities of recycle and makeup gas are estimated as follows for various-sized plants (since it is difficult to generalize on vent gas capacity as a function of plant capacity, it is not included).

Hydrotreater Capacity, bpd	Recycle Gas		Makeup Gas	
	scfm	acfm	scfm	acfm
10,000	15,000	225	3,500	250
20,000	30,000	450	7,000	500
30,000	45,000	675	10,500	750
40,000	60,000	900	14,000	1000
50,000	75,000	1125	17,500	1250

In most plants, all three compressor applications use reciprocating machines. The makeup hydrogen capacity may be large enough on larger plants, but the compression ratio is out of the range of the centrifugal compressor. The recycle compression ratio is low enough for a centrifugal application, but on plants smaller than 30,000 bpd, the capacity is too small. On larger plants, centrifugal recycle compressors should be evaluated against reciprocating units. Many centrifugal compressors are used in this operation.

Hydrogen Plants

Installation of hydrocrackers and hydrotreaters has resulted in a large demand for hydrogen in refineries. Cat re-formers, on the other hand, produce hydrogen as a by-product, but usually not in sufficient quantities to supply the bydrocracker and hydrotreaters in a refinery. Thus, supplemental hydrogen is often required. Two processes are available for hydrogen production:

1. Partial oxidation of heavy hydrocarbons, such as heavy fuel oil.
2. Steam re-forming of methane (from natural gas).

Since steam methane re-forming is currently much more widely used than partial oxidation, it will be described in this section.

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Process Description

The flow diagram is shown in Fig. 7.8. The process takes place in three steps, as follows:

1. *Re-forming:* Natural gas (methane, CF_4) is pumped into the plant at approximately 200 psig pressure along with a supply of steam. With the addition of heat, the steam-gas mixture reacts in the presence of a catalyst to produce carbon monoxide and hydrogen:

$$CH_4 + H_2O \rightarrow CO + 3H_2$$

This is the re-forming reaction and it takes place at 1500°F. The reaction is carried out by passing the gas and steam mixture through a bank of catalyst-filled furnace tubes The furnace consists of one or two rows of numerous vertical tubes, fired on each side, to obtain even heat distribution around the tube and along the length of the tubes because of the extremely high tube wall temperature 1700 to 1800°F. Special alloys are used for the furnace tubes.

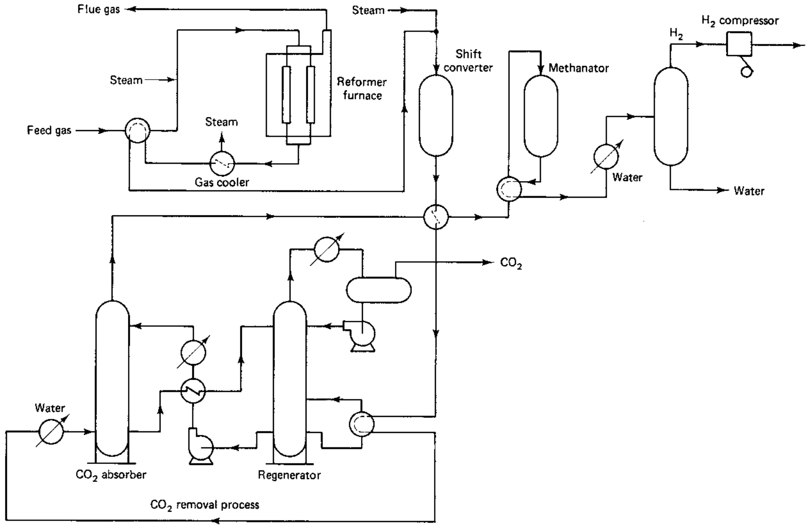

Figure 7.8 Hydrogen Plant, Process Flow Diagram

2. *Shift conversion:* The product from the re-forming reaction is then cooled to about 650°F and enters the shift converter along with additional steam. In the shift convertet, a fixed-bed catalyst reactor, the carbon monoxide in the gas reacts with the steam in the presence of shift catalyst to produce more hydrogen and to convert the CO into CO_2:

$$CO + H_2O \rightarrow CO_2 + H_2$$

The CO_2 is then removed from the gas by absorbing it in a special solution and boiling off from the CO_2 solution to the atmosphere or to a CO_2 collector for further use or sale.

3. *Methanation:* The remaining small quantities of carbon monoxide and carbon dioxide are converted to methane in another fixed-bed catalytic reactor for the final purification step in the process.

These process steps are required to obtain very pure hydrogen as a product, since impurities are detrimental to the expensive catalyst in the hydrocracker. Purities of 95% and higher are obtained in the steam methane reformer.

From the methanator, the gas, with a molecular weight of 2 plus, leaves the plant at about 150 psig and is ready to he compressed to the 2000 psig pressure level for supply to the hydrocracker and hydrotreater.

Compression Equipment

There are two major requirements for compressors in a hydrogen plant:

1. Compressors to move the methane into the process.
2. Compressors to move the nearly pure hydrogen from the process to the hydrocracker and hydrotreater.

Methane Compressor

Methane, or natural gas, is usually available at about 25 psig or higher and must be compressed to the 200 psig level in the steam methane re-former. Since hydrogen plants are sized by the volume of hydrogen produced (5 million, 10 million, or even 75 million sfcd), the size of the methane compressor for a given plant is not evident from the plant size. However, by calculating how many mols of methane are required for the reaction to produce 1 mol of hydrogen, the quantity of methane feed gas can be determined. The following table should be helpful in estimating compressor size.

H_2 Plant, scfd	Methane, scfm	Methane acfm, at 15 psig
10,000,000	3,300	1200
25,000,000	8,000	3000
50,000,000	16,000	6000
75,000,000	24,000	9000

From this table, based on capacity, centrifugal compressors are applicable for plants as small as 10 million scfd. With respect to head, a compression ratio of 51/4 is attainable in a single casing since the molecular weight of the methane is 20. A compression ratio of 5 1/4 will produce a discharge pressure of about 215 psia, which is adequate for most plants.

Conventional, horizontally split centrifugal compressors, motor or turbine driven, are suitable. Metallurgy is standard since the gas is noncorrosive, and sealing with carbon rings or oil seals is satisfactory.

The one problem with the centrifugal compressor application is turn-down ratio. The hydrogen plant capacity often fluctuates, especially if it is operating in parallel with one or more cat re-formers to furnish hydrogen to the refinery system. It is important to make sure that an adequately sized bypass line is available for partial-flow operation.

On plants smaller than 10 million scfd, reciprocating equipment should be considered. Heavy-duty, balanced opposed compressors would be applied to this service. With multiple compressors, the turn-down problem becomes less troublesome.

Hydrogen Compression

The compression of hydrogen from the hydrogen plant to the hydrocracker is discussed in the hydrocracker description. Normally, reciprocating compressors are required to pump the hydrogen, with a molecular weight of 2, from the 125 to 200 psig intake pressure to the 2000 psig pressure level required at the hydrocracker.

INDUSTRIAL REFRIGERATION

Chemical plants, refineries, dry-ice and meat-packing plants, along with breweries, all need refrigeration compression equipment. This refrigeration cycle is driven by screw or reciprocating compressors, and the power consumed adds considerably to the cost of the items produced.

Selecting the right compressor requires that the following be considered:

1. Refrigerant to be used
2. Refrigeration cycle:
    a. Whether single-stage or multi-stage
    b. Whether a cascade system will be selected
       (two refrigerants used in series)
3. Loading demand of the system
4. Monitoring of the refrigeration cycle loop to ensure that a malfunction within the cycle loop will he discovered and corrected

The refrigerant used in most industrial systems is ammonia because of certain thermodynamic properties. The low molecular weight of ammonia reduces the friction losses in the loop, and the saturation pressure is such that lower pressure is required to liquefy the ammonia in the condenser.

Typical ammonia [specific gravity 0.594 at 60°F] and 14.7 psia cycle conditions are as follows:

Evaporator: pressure, above atmospheric in the range of 25 psig; temperature, - 28°F and higher.

Condenser: pressure, 150 to 200 psig; temperature, 86°F.

The approximate value of bhp/ton of refrigerant is as follows for reciprocating compressors:

Ammonia	1.172
Refrigerant 11	1.111
Refrigerant 12	1.23
Propane	1.256
Propylene	1.262
N-butane	1.193

The Refrigerant 11 shows a lower bhp/ton; however, the price of R11 and R12 is much more than that of ammonia. Since most industrial systems are large, the cost to charge a system with other gases is higher than that of ammonia.

While R11 is used in some refrigeration cycles because of its specific gravity [4.78 at 60°F and 14.7 psia], the systems are more costly to install than those using ammonia.

The typical performance of an ammonia refrigerant system is as follows:

Performance, Based on an Ammonia Refrigerant						
Evaporator temperature	-15°F	-1°F	+12°F	-26°C	-18°C	-11°C
Evaporator pressure	21 psia	30 psia	40 psia	1.5 bars	2.1 bars	2.8 bars
Condenser temperature	96°F	96°F	96°F	35.6°C	35.6°C	35.6°C
Condenser Pressure	200 psia	200 psia	200 psia	13.8 bars	13.8 bars	13.8 bars
Comp. Ratio	9.5	6.7	5.0	9.5	6.7	5.0
BHP/ton, stage 1	1.89	1.48	1.22	1.89	1.48	1.22
BHP/ton, stage 2	1.74	1.48	1.22	1.74	1.48	1.22

A typical ammonia compressor is shown in Fig. 7.9.

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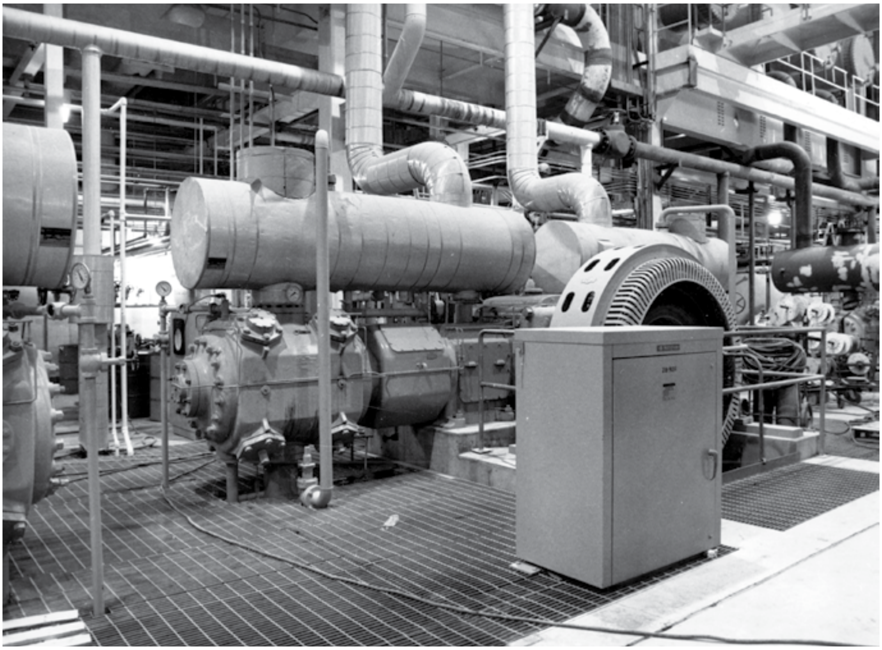

Figure 7.9 A heavy frame, 2500 hp electric-driven ammonia compressor.

CHEMICAL INDUSTRY PROCESSES

Synthetic Ammonia (NH_3) Production

Large quantities of ammonia are produced throughout the world, since it is an essential ingredient of chemical fertilizers. Nitrogen is necessary for living cells to produce protein. Very few living plants can produce amino acids that contain the elements carbon, hydrogen, nitrogen, and other elements in complex combinations that are essential foods for man and animals. The few living plants that can produce nitrogen are called legumes, including clover, alfalfa, and beans. These plants have a bacteria that fixes, or extracts, nitrogen directly from the air. Large quantities of nitrogen are formed during severe storms where lightning produces nitrogen from the air and this nitrogen is then absorbed by rain.

The first attempt to produce nitrogen utilized a high-voltage discharge and was made in 1780 by Lord Cavendish. A commercial plant using electric discharge was established at Nottoten, Norway, in 1900 and at Niagara Falls, New York, in 1902. Haber, a German scientist, was the first to synthesize ammonia by combining three parts hydrogen and one part nitrogen at 100 atm and cooled to 1112°F. The process used a catalyst material of asmium. The chemical equation $3H_2 + N_2 \rightarrow 2NH_3 + 26{,}000$ cal describes the reaction. The commercial development of Haber's discovery was carried out by Bosch. A chemical firm was established called Badische Anilin und Soda Fabrik (BASF). By 1913, it proceeded to construct a large synthetic ammonia plant. The main problem was to obtain the two constituents in a free state.

Nitrogen is obtainable from an air-separation plant. This same process produces oxygen as well. The oxygen by-product is then reacted with methane and steam to produce hydrogen. The chemical equation is $CH_4 + 2H_2O \rightarrow CO_2 + 4H_2$ for this process.

Other sources of hydrogen can be natural gas, water gas (steam passed over glowing coal or coke), chlorine cells, coke-oven gas, water electrolysis, liquid hydrocarbons, and refinery gas, such as catalytic re-forming as described previously. The most practical processes for obtaining hydrogen and nitrogen components are the partial oxidation system and the steam or natural gas re-forming system, shown in Figs. 7.10 and 7.11.

The partial oxidation process gave many problems in the design of furnace equipment to handle high temperatures and high-pressure gases or oils. In 1954, Texaco introduced its synthesis gas generation process. This process had the advantages of complete independence from hydrogen sources, whether gas or liquid, and operated at line pressure of 400 psig. Its biggest disadvantages were the requirement for an air-separation plant to provide oxygen for the furnace and its relatively short burner life. The furnace temperature was 2000°F, thus requiring a water quench as shown in Fig. 7.10.

The natural gas or steam re-forming process (Fig. 7.11) requires the feedstock he heated to 1200°F. At the secondary reformer the reaction is endothermic. The nitrogen component enters the process as a component of air. The water quench, plus nickel catalyst, converts the hydrocarbons to carbon dioxide and free hydrogen or, in a concurrent reaction, carbon monoxide and free hydrogen. The shift converter reacts carbon monoxide with additional steam to form carbon dioxide and free hydrogen. The methanator reacts residual CO and CO_2 with hydrogen and thus forms methane and water vapor. Methane is less objectionable as an impurity than CO or CO_2.

High design reactor pressures will increase the conversion rates and reduce the size of the converter required.

Early synthetic ammonia compressors installed in the United States in 1925 were designed for final discharge pressure of 4500 psig. In 1939, a new process named the Claude (French) process used a design final pressure of 15,000 psig. In 1980, a low-pressure system was introduced that required pressures of only 400 to 600 psig. This new system has a rating of 1000 tons. The new process design utilized dynamic compressors, while prior to 1970 nearly all compressors were reciprocating, positive-displacement units. The new low-pressure process has not, however, replaced many of the existing high-pressure plants.

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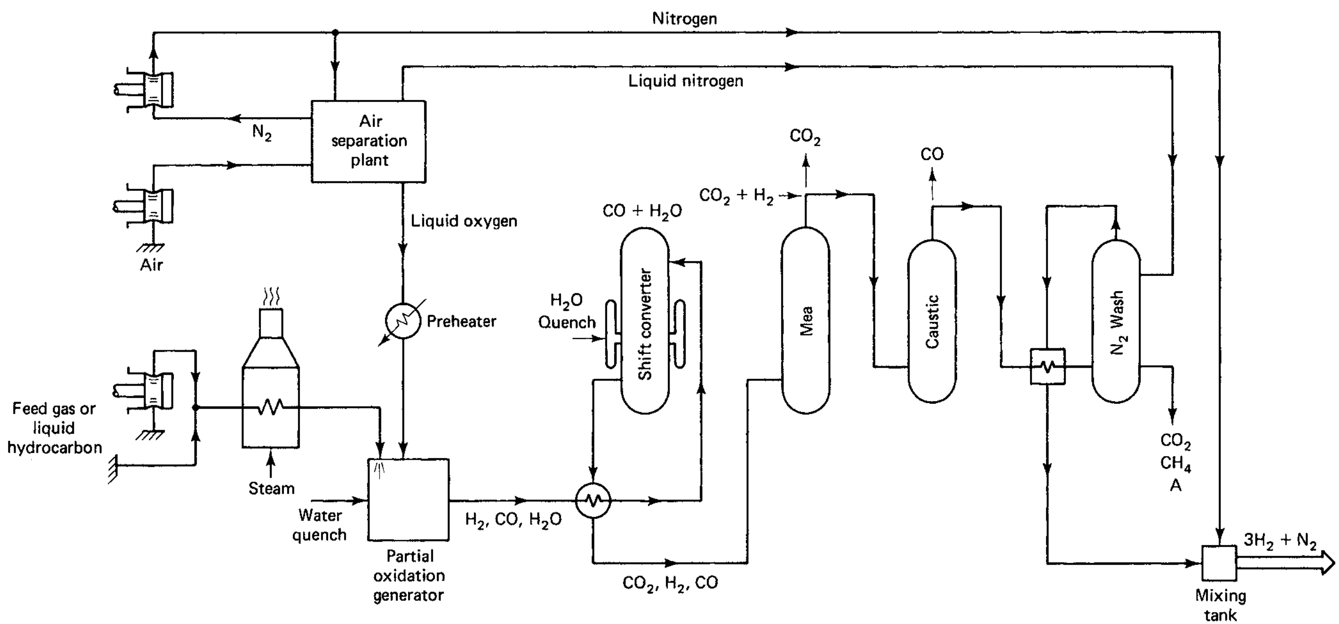

Compression requirements

A. Air – 600 to 2500 psig (depending on air plant)
B. Nitrogen – 330 psig or 2500 psig with sidestream
C. Feed gas – 400 to 450 psig
D. Possibly ethylene, propane, or ammonia refrigeration for air plant

Figure 7.10 Flow Diagram for the Partial Oxydation Process

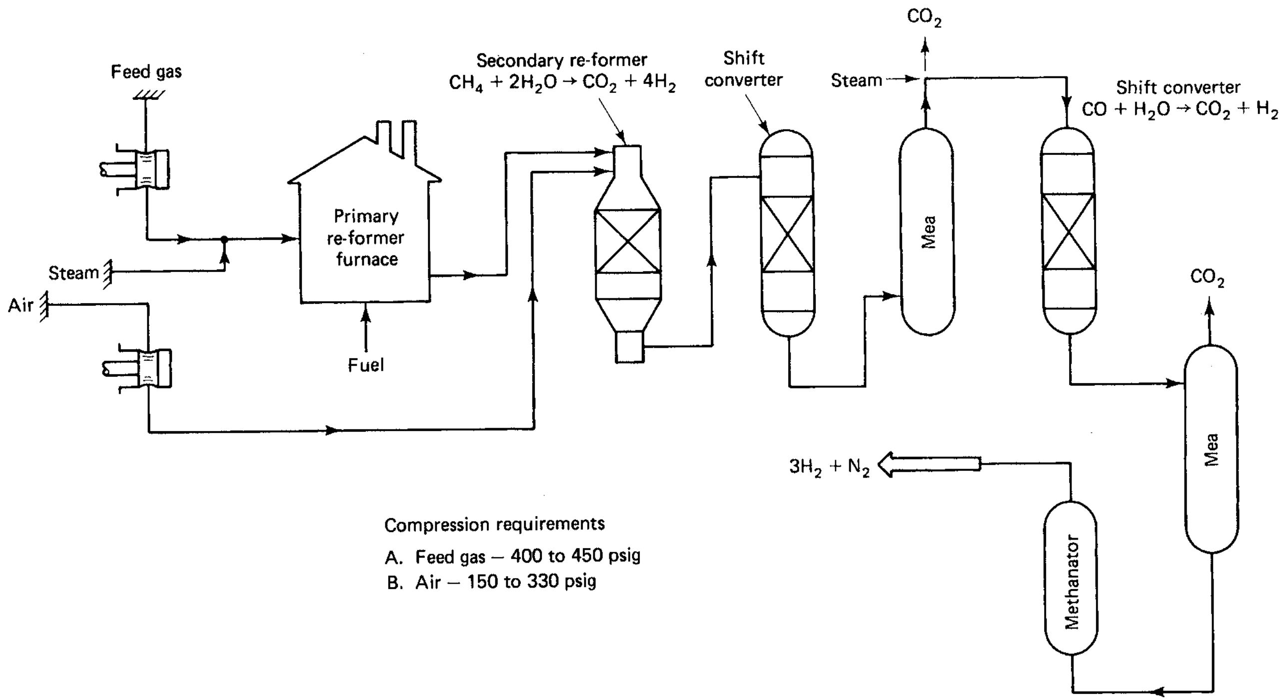

Figure 7.11 Natural Gas Reforming

Urea Production

The urea compound has a high nitrogen content and is an excellent fertilizing agent. Combining urea and formaldehyde yields a useful plastic. The formation of the compound is accomplished by the reaction of ammonia and carbon dioxide. The reaction pressure varies from 2000 to 6000 psig depending on the process.

The chemical reaction is described by the chemical formula $CO_2 + 2NH_3 \rightarrow CO(NH_2)_2 + H_2O$. The initial product is carbonate solution and the process is exothermic. By holding the solution in the reactor under elevated pressure and temperature, most of the carbonate is converted to urea. The conversion process is endothermic, but more heat is formed in the carbon dioxide and ammonia reaction than is absorbed in the conversion process. The higher the reactor pressure, the greater is the conversion. What carbonate is not converted is decomposed at a lower pressure. The ammonia gas is compressed to a pressure at which it will condense. This ranges between 200 and 270 psig. The ammonia feed and ammonia recovered from the decomposition is pressurized to the reactor pressure by using high-pressure reciprocating pumps.

Urea can be crystallized into pellets by a falling-film evaporation process. Urea for agricultural purposes also can be used as a liquid. Combining urea and formal-dehyde yields a very useful plastic. The characteristic of this plastic is that it has brighter colors, is oil resistant, and is odorless and tasteless.

Heavy-duty, horizontal, electric-driven reciprocating compressors are used for urea production. These compressors often handle the carbon dioxide and ammonia service on the same compressor frame. A typical urea process is shown in Fig. 7.12.

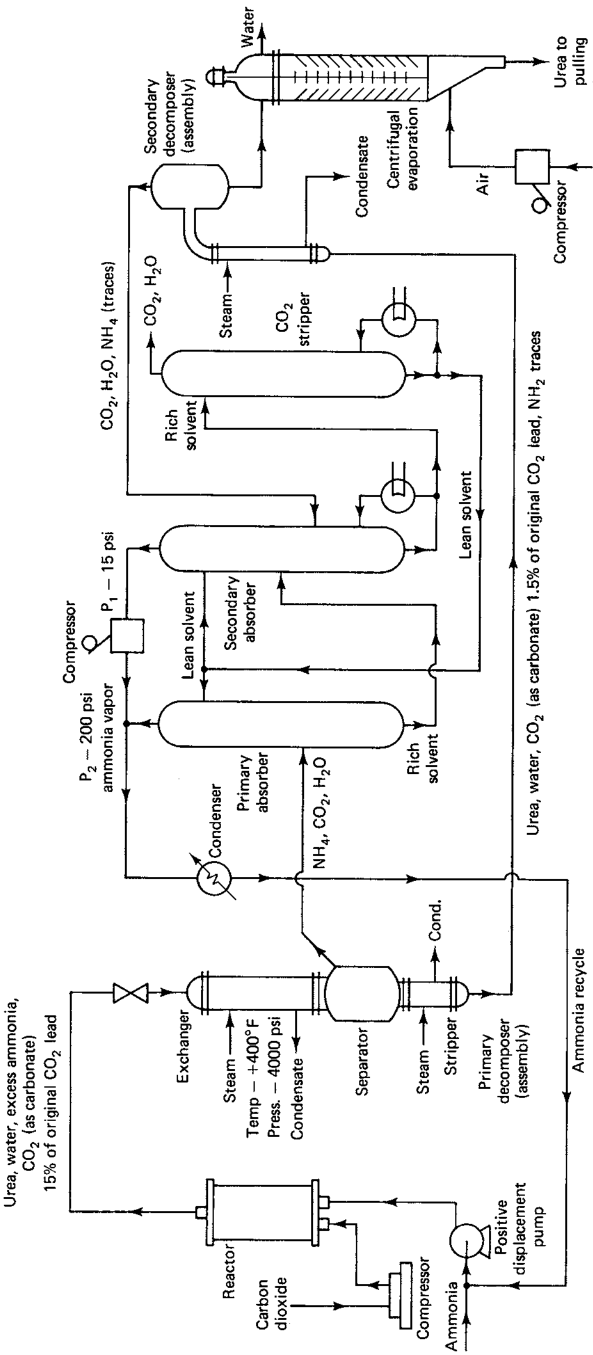

Figure 7.12 CPI-Allied Chemical Company, Urea Process

HALOGEN COMPRESSORS

The halogen family consists of elements that occur in salt beds or in sea water. The word halogen is a Greek word meaning salt forming. Chemically these elements are very active and are therefore not found in the free state, but in combination with other elements. The following table lists some of the important properties of the halogens.

Physical Properties of Halogens

Formula	Element	Color and State	Molecular Weight	Melting Point, °C	Boiling Point, °C
F_2	Fluorine	Pale yellow gas	19	-223	-188
Cl_2	Chlorine	Greenish-yellow gas	35.46	-103	34.7
Br_2	Bromine	Dark red liquid	79.92	-7.2	58.0
I_2	Iodine	Purplish-black solid	126.92	113	183.0

Fluorine

Fluorine occurs in nature combined with calcium as an insoluble salt (CaF_2), in igneous rocks such as cryolite (Na_3AlF_6), and as a double salt, apatite [$Ca_5(PO_4)3F$].

Chlorine

The main source of chlorine is common salt. It is not found in the free state because of its active chemical behavior. Chlorine is separated from sodium by electrolysis of brine.

$$2NaC1 + 2H_2O + \text{direct current} \rightarrow Cl_2 + H_2 + 2NaOH$$

Many small chlorine plants utilize single-cylinder, single-stage reciprocating compressors. A very common arrangement is to have a multiple number of first-stage units in one row and, on the opposite side, a row of second-stage units. This provides flexibility in the production rate since the number of cells on-stream can vary. Plants that are designed for output of 250 tons per day or more use centrifugal units. The chlorine leaves the cell from the anodes at 176°F, saturated with water vapor. The moisture is removed by a cooling and countercurrent bubble cap tower, with concentrated sulfuric acid scrubbing; the chlorine leaves the tower at about 68°F. The gas is pulled through the drying tower by the slight vacuum created by the first-stage compressor(s). The chlorine gas is then compressed in the required number of stages to condenser pressure at which the chlorine is liquefied, at 120 psig if water is used.

The oceans of the world contain an estimated 18 trillion tons of chlorine. Salt beds, salt domes, and volcanoes, all formed during past geological ages, represent the major commercial sources.

Bromides

Bromides are found in sea water as compounds. The dissolved bromides yield a pint of bromide per 7 1/2 tons of sea water. Salt wells in Michigan and other areas of the world are rich in sodium bromide.

Iodine

Iodine can be extracted from tissues of seaweed (kelp). The more commercially viable sources are found in brines, in oil wells, and as sodium idodate $NaIO_3$ as a 0.2 percent impurity in the sodium nitrate mines of Chile.

Chemically, any halogen will oxidize any halogen ion that comes later than itself in the periodic table. Commercially, bromide is produced chemically in this way: $C1_2 + 2NaBr \rightarrow 2NaCl + Br_2$.

Hydrogen Chloride

The HCl gas produced by burning pure hydrogen and chlorine contains contaminants that form deposits on hot surfaces such as valves used in reciprocating compressors. The operating life of these parts is extended if the gas is scrubbed and filtered. The design is similar to that of chlorine compressors. The HCl gas entering the compressor has more impurities and deposits than is usual for chlorine gas (HCI is highly hydroscopic). The major use of HCI is as basic feedstock to make polyvinyl chloride plastics, silicone rubber products, urethane foams, and the like.

Synthetic Polymers

A variety of synthetic polymers is produced throughout the world. The chemical industry has classified different polymers as follows:

Ordered polymer: A very large molecule in which the arrangement of the atoms is closely controlled.

Condensation polymerization products: The material produced by the chemical reaction condenses or solidifies at a given pressure and temperature. These are characteristics of nylon, Dacron, Mylar, and polyurethane.

Additional polymerization products: Materials similar to Saran, Orlon, polyvinyl chloride, Teflon, polystyrene, polyethylene, and polypropylene.

Gas compressors are necessary for feeding gases to the reactor (Fig. 7.13). Gases usually encountered in the production of synthetic polymers are ethylene, hydrogen, hydrogen chloride, methyl chloride, phosgene, and butane. In some cases, the gas is compressed to a pressure at which it can be liquefied and then pumped into the process.

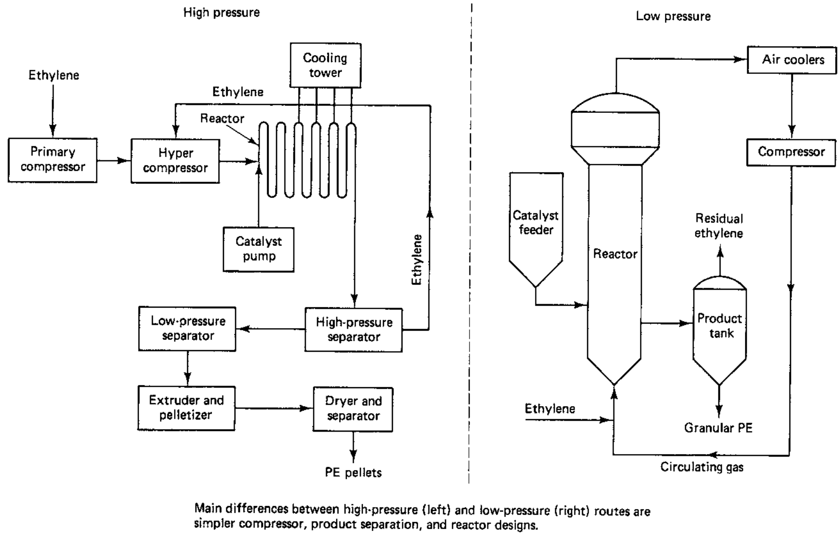

Figure 7.13 Synthetic Polymer Reactor

Polyethylene Production

Polyethylene polymer is, as stated, an ordered polymer. There are basically three varieties of this polymer presently manufactured. The low-density polyethylene product was originated by Imperial Chemical Industries in England in 1933. By 1938, 1 ton of polymer had been produced. In 1977, the world production, excluding the USSR, was 22 billion lb/year. The U.S. production alone was 8.3 billion lb/year. The polyethylene produced has a density of from 0.91 to 0.935 g/cm^3.

The low-density polyethylene process uses large reciprocating compressors compressing ethylene gas to 15,000 to 35,000 psig generally and as high as 50,000 psi in at least one case. The high-pressure ethylene in contact with the catalyst in a timed exposure period by flow rate is converted to a polymer. The rate of conversion is about 20 to 25%. The gas and molten polymer are separated. The unreacted ethylene is piped back to the inlet of the high-pressure compressor. In each plant there will be a primary or feed gas compressor, a secondary or hyper compressor (Fig. 7.14), and generally a small booster compressor. The normal operating pressures of a polyethylene plant are as follows:

Unit	Inlet (psig)	Discharge (psig)
Booster	0 – 70	150 – 500
Primary	150 – 500	1700 – 5000
Hyper	1700 – 5000	19,000 – 35,000

The end uses of low-density polyethylene are molded articles, approximately 14%; electrical insulation, approximately 11%; film, approximately 34%; pipe, approximately 15%; paper coatings, approximately 8%; bottles and tubes, approximately 11%; and miscellaneous, approximately 6%.

Various properties of polyethylene are important, depending on the application, and the product is manufactured and controlled to promote the best obtainable quality of one or more specific characteristics: crystallinity, relative rigidity, softening temperature, tensile strength, elongation, impact strength, and density. The high-pressure, low-density polyethylene process produces a material having the characteristics needed for most markets.

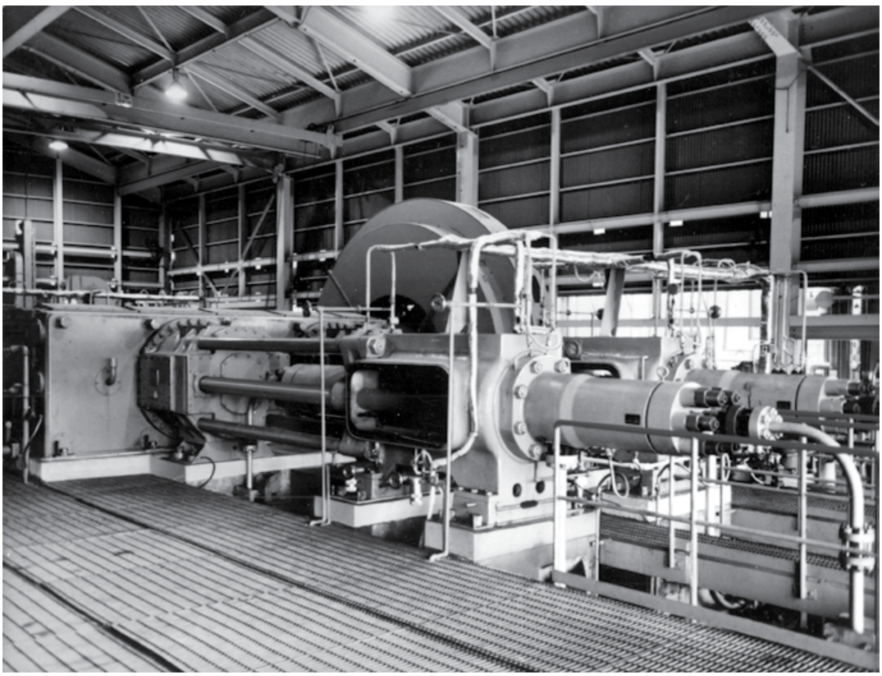

Figure 7.14 A 5000 hp heavy duty horizontal, electric driven, high pressure (hyper) compressor for the production of polyethylene.

Resin properties are controlled by varying throughput rates, the method of initiating conversion or reactor configuration. Polyethylene is formed by stringing the ethylene molecules together in a long chain and then constructing cross branches to tie these chains together. Normally, an increase in reactor pressure increases the number of cross branches. The chain branching with especially short branches is probably the single most important factor that affects the physical properties of polyethylene.

An entirely different process produces high-density polyethylene. This compound is only a linear arrangement of the chains. The fewer the cross branches, the closer together the strings will lie and, therefore, the higher the density will be. High-density polymers are produced at a process operating pressure of 15 to 3000 psi. The product has a density of 0.945 to 0.970 g/cm^3. The polymer produced is ideally suited as a synthetic fiber material since the crystallinity is 75% or higher. Copolymerization of ethylene with relatively small amounts of other olefins such as *I*-butane and propylene produces polymers with slightly lower densities and degrees of crystallinity. As ordered polymers, it is very possible for such plastics to possess a range of properties at higher pressures. On the other hand, the feed gases of propylene and f-butane are more expensive than ethylene.

Thirty-five percent of ethylene is produced by cracking ethane and 31% by cracking propane. The remainder comes from cracking heavier hydrocarbon material, such as naphtha or gas field condensates, off-gas from catalytic cracking units in petroleum refineries containing ethylene, and other light hydrocarbons. Ethylene production has increased to the point that there now exist ethylene pipelines extending over long distances in Louisiana and Texas. Pipeline compressors are nonlubricated, reciprocating, positive-displacement machines.

Production of Low-density Polyethylene Using a Low-pressure Process

In this process (Fig. 7.13), gaseous ethylene and a solid, supported, chromium-containing catalyst are fed to the reactor. Gas is recirculated through the reactor to keep the growing polymer particles fluidized. The recirculating gas passes continuously through coolers to remove reaction heat. The polymer is withdrawn as a dry granular product, which passes directly to the pelletizing extruder. Reaction pressure is 100 to 300 psi, and reaction temperature is controlled at less than 100°C (212°F). Only one compressor is required to feed the fluidized reactor. This process has energy savings and capital-cost advantages.

OIL AND GAS INDUSTRY

Gas Gathering

Gas gathering is defined as the collection of natural gas from the well head and moving it to a gas plant or to a major transmission line (Figs. 7.15 and 7.16). This application can involve either individual, small well-head compressors as small as 5 horsepower or larger-power central units of several thousand horsepower handling several wells or an entire field. As the field pressure drops due to depletion, the suction pressure at the compressor will drop correspondingly, increasing the ratio of compression. The discharge pressure can also vary greatly depending on the flowing pressure of transmission pipelines in the area. Typical ranges of pressure for a reciprocating compressor in gas-gathering services are suction from about 10 to 150 psig and discharge from 150 to 1200 psig. Machines are expected to handle a wide range of these conditions and keep the driver reasonably well loaded.

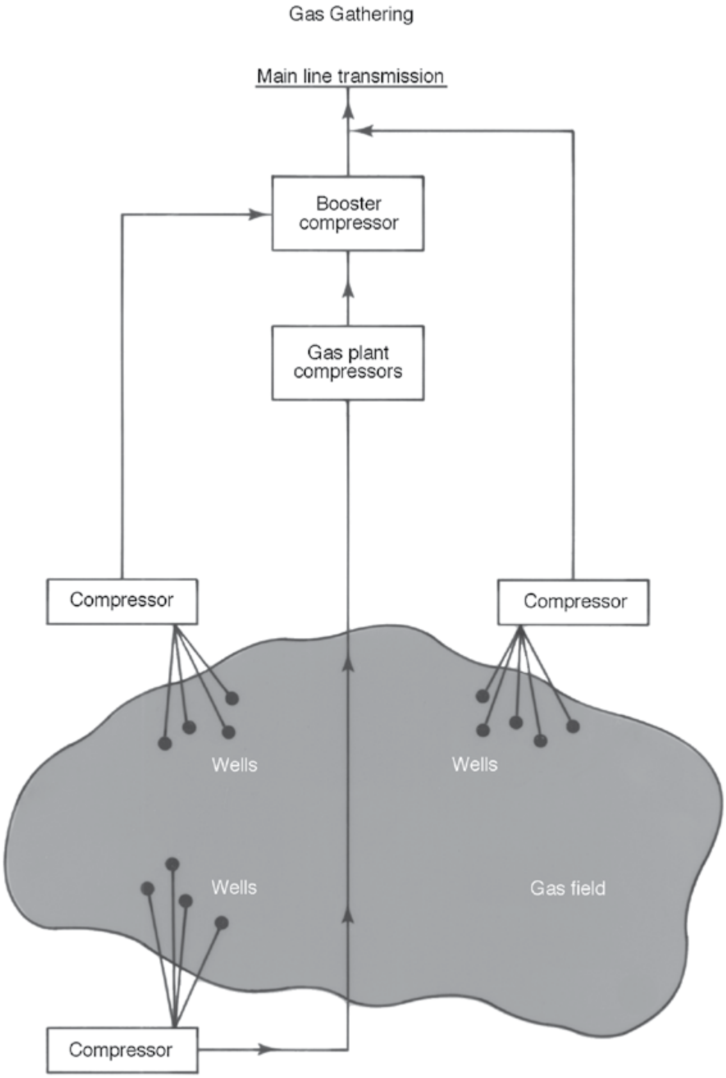

Figure 7.15 Gas gathering is the moving of natural gas from the well head to a gas plant or a major transmission line.

Gas-gathering applications probably account for the majority of installed reciprocating compressor horsepower in the oil and gas industry. Compressor units are generally sold to the user; however, the rental of compressor units is growing rapidly and is itself becoming a viable service industry.

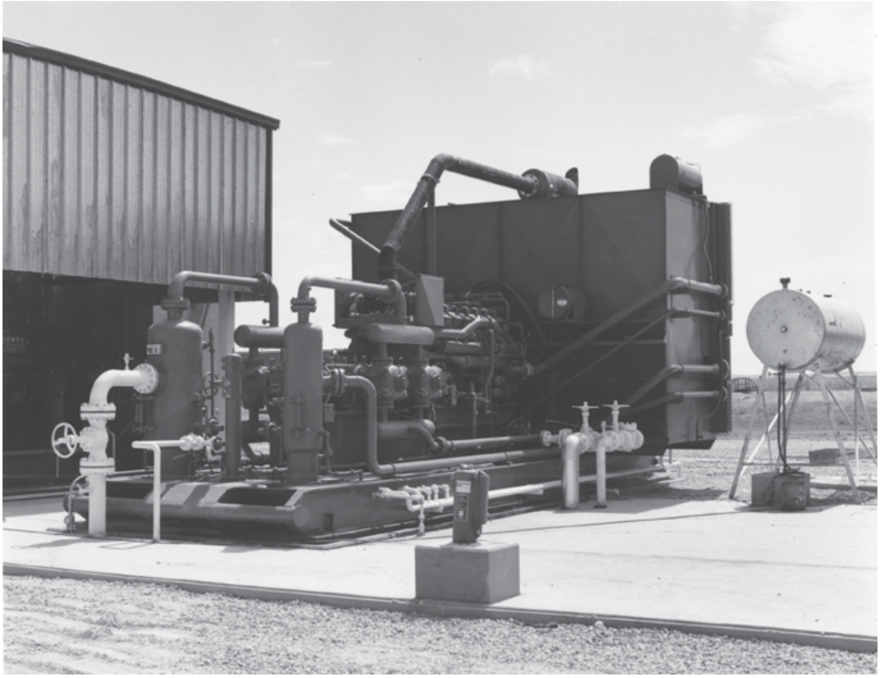

Figure 7.16 Field installation of a balanced opposed compressor in gas gathering service.

The nearer the compressor is to the well head, the more likely the gas will be saturated. Some tolerance of this liquid is desirable in the design of a field unit. Additionally, even the standard machine should be capable of compressing gas with as much as 2 percent H_2S without serious corrosion problems. Reciprocating compressors are generally utilized in gas-gathering applications; however, some rotary positive units are used in low-pressure applications.

Gas Lift

Gas lift (Fig. 7.17) utilizes natural gas to produce oil. Bottom hole pressures are maintained by forcing gas down the well casing. Oil is then lifted upward through the production tubing. The gas-oil mixture passes through a separator where the gas is separated from the oil, and the gas is then piped to the compressor suction, where it is recompressed. Typical conditions of service are suction pressures ranging from 25 to 65 psig and discharge pressures from 800 to 1200 psig.

Gas lift is utilized where electricity to run pumps is not practical or economical and gas is readily available. Reciprocating compressors are generally utilized, such as those in Fig. 7.18.

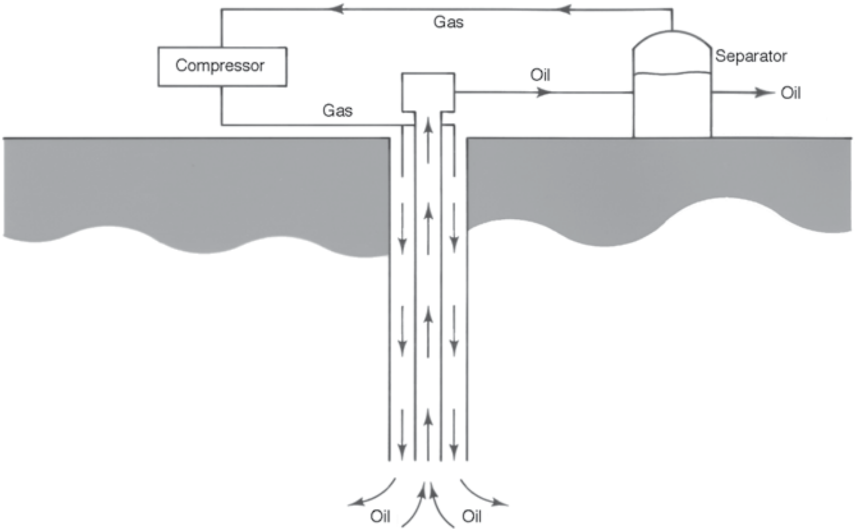

Figure 7.17 Gas lift application utilizes gas to produce oil.

Figure 7.18 Field installation of a balanced opposed, two-stage compressor in gas lift service.

Reinjection

Reinjection (Fig. 7.19) is the compressing of natural gas for injection into the gas cap of a formation to maintain field pressures and prevent salt water from moving into the oil field. This helps in the production of the oil and provides an alternative to flaring when there is no market for the gas that is being produced. When the oil reaches the surface, the gas is separated and sent to the compressor, where it is recompressed. These applications often require high pressure since field pressures may be high when the field begins producing. Typical pressure conditions are suction from 80 to 400 psig and discharge from 3000 to 4000 psig.

Distribution

This service, although not actually a gas field application, is the moving of gas after it leaves the main transmission line. Often, some compression is required to deliver the gas to industries or residences, or both. This service is relatively easy, since the gas is clean, at moderate pressures such as 200 to 400 psig, and does not require a great deal of compression. Generally, reciprocating compressors are utilized.

Transmission

Transmission is the movement of gas across country from the originating field to population centers where it is used. Generally, this involves large-diameter pipe with typical pressures from 900 to 1200 psig.

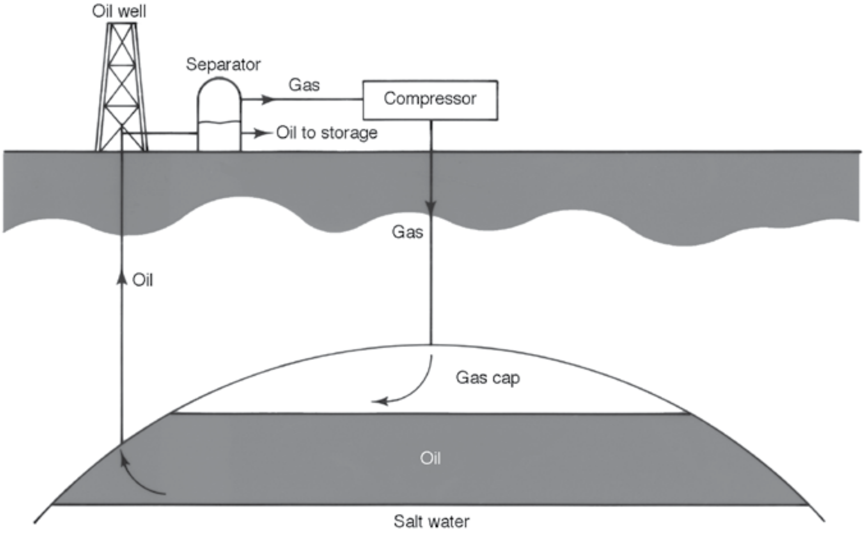

Figure 7.19 Reinjection aids in the production of oil and provides all alternative to flaring.

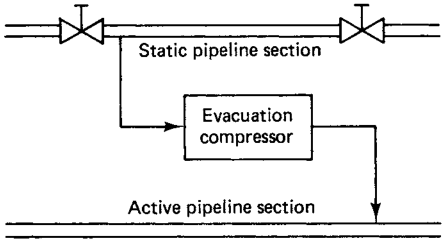

Figure 7.20 Pipeline evacuation.

Compressor stations are required at intervals to maintain pressure and flow rates. These stations must handle high volumes of gas at low ratios of compression. The horsepower requirements are large and employ either gas-turbine-driven centrifugal or integral engine-driven equipment. Efficiency becomes more important as the gas moves farther from the field toward the end user.

Pipeline evacuation evolved with the rapid rise in natural gas prices during the early 1980s. Prior to that time, it was common practice to vent the gas in a static section of pipeline to atmosphere while repairs were made to the pipeline. The static pipeline section could be anywhere from about a quarter of a mile to many miles in length. As gas prices increased, venting became economically unattractive.

Pipeline evacuation involves the transfer of gas from a static section of pipeline to an active section of pipeline (Fig. 7.20). This is accomplished by a reciprocating compressor that can handle wide variation in suction pressures while compressing against a constant discharge pressure. As gas is evacuated from the static section, the suction pressure declines, the compression ratio increases, and the compressor capacity decreases.

Applications can vary widely, but the following range of conditions can be considered typical:

	Initial Pressure	Intermediate Pressure	Final Pressure
Intake (psig)	850	450	50
Discharge (psig)	850	850	850

Packaged compressor systems specifically designed for this application feature multi-stage compressors that can maintain high driver loading throughout a wide range of compression ratios. Most units of this type are driven by natural gas engines. The entire system (compressor, engine, cooler, etc.) is generally mounted on a trailer for portability to various field locations.

Storage

Since the utility demand for natural gas is much greater during the winter months, it is necessary to store the gas produced during warm weather in reservoirs and then retrieve it to satisfy peak cold-weather requirements. The compressor must not only be able to handle filling the reservoir but also the return of the gas.

This dual service requires operating pressure flexibility and is provided best by the reciprocating compressor. Typical pressure conditions are suction from 35 to 600 psig during injection, 300 to 800 psig during withdrawal, and discharge from 600 to 4000 psig during the injection phase and 700 to 1000 psig as the gas is withdrawn from the reservoir and fed to the transmission line.

Enhanced Recovery

Since up to 60% of the discovered oil remains in place after primary and secondary recovery, further (tertiary) methods of recovery may be economically attractive. Three major categories of enhanced oil recovery are now in use.

1. ***Thermally enhanced recovery.*** These methods are generally applicable to heavy oils and require steam injection or *in situ* combustion. Steam-injection processes are either steam soaking or steam drive. In the soaking process, the steam is allowed to remain in the strata for a few days or weeks, after which the well is returned to production, with the oil having been heated to a viscosity enabling it to flow under the action of lift equipment. The process is then repeated.

   Steam drive requires a continuous injection of steam to form a steam flood front. Fluids move from the injection well to the producing well. This process does not require compression, as the steam is injected into the formation at boiler pressures.

   In situ combustion involves an underground fire flood. Compressed air to support combustion is injected down the well into the producing strata to maintain the fire. The heat makes the oil less viscous and more easily recovered. Generally, reciprocating compressors are utilized. Typical discharge pressures are from 1500 to 5000 psig.
2. ***Chemically enhanced recovery.*** Polymer flooding is a chemically augmented water flood in which small concentrations of chemicals are added to injected water to increase flood front effectiveness in displacing oil.

   Alkaline flood involves the addition of sodium chemicals. These solutions will react with constituents present in oil to form detergentlike solutions. This material reduces the ability of the formation to retain oil.

   Micellar-polymer is new and expensive but has the ability for efficient oil recovery. This system requires no compression equipment.
3. ***Gas-enhanced recovery.*** Carbon Dioxide or Nitrogen. This method involves miscible and immiscible applications of compressed CO_2 or N_2. In miscible processes, the injected gas mixes with the oil and forms a single oil-like liquid that can flow through the reservior. CO_2 is often selected for certain formations due to its ability to mix with oil and its sweep characteristics through the formation. CO_2 also reduces viscosity, density, and

surface tension, as well as distilling the light ends. It has also proved to be less expensive than natural gas in many cases.

Complete miscibility or mixing of CO_2 with crude oil depends on reservoir temperature, pressure, chemical nature, and oil density. Economics of CO_2 injection dictate that large quantities must be available within 300 or 400 miles when naturally produced or available as a result of flue gas separation from plants where CO_2 is a by-product.

The selection of N_2 or CO_2 as a medium for injection depends on the formation structure. In some reservoirs, nitrogen is utilized for pressure maintenance or flood application where CO_2 is not available. Nitrogen is often the product of an air-separation plant.

Both CO_2 and nitrogen miscible flooding generally involve significant amounts of compression by either reciprocating or centrifugal compressors. CO_2 is taken at pressures below 200 psig, compressed above 1500 psig, and put in a pipeline. At the field, it is injected into the ground at field pressure, which frequently is above 2000 psig. When the oil is recovered, the CO_2 is recycled into the field.

Several processes allow the producer to reclaim the CO_2 from the gas and reinject it into the producing formation. These processes are refrigeration, cryogenics, membrane separation, and absorption. Each of these processes has a different removal efficiency. Approximately 60% of the CO_2 can be removed from a natural gas stream by the use of the membrane system. Membrane units consist of a vessel filled with thousands of hollow fibers. The natural gas containing the CO_2 is passed over the fibers, and the natural gas, being more absorbent than the CO_2, diffuses through the membrane material to the hollow center.

Furthermore, CO_2 absorption can be accomplished through the absorption process using methyldiethanolamine (MDEA) and diethanolamine (DEA). These two processes in series with a membrane system will remove the balance of the CO_2. The MDEA process will remove approximately 38% and the DEA process another 2 percent.

The refrigeration/cryogenic process uses a cooling medium to condense the CO_2 out of the gas stream. These processes require some form of dehydration in the system to preclude freezing in the process. After the CO_2 is removed, it can be dehydrated, compressed, and reinjected into the formation.

When air-separation plants are installed and N_2 is used, compression is required to inject the N_2 into the oil field. Like CO_2, the N_2 produced can be reclaimed in a nitrogen rejection plant. A cryogenic process is used to separate the nitrogen from the associated gas stream. The cold box, which separates the N_2 from the residue gas stream, can be a separate addition to an existing gas plant, or it is possible to design an integrated NRU (nitrogen rejection unit) and NGL (natural gas liquids) processing plant. If the combined plant is selected, it is possible to design an integrated plant that optimizes the N_2 and gas liquids recovery, leaving the methane as salable gas. The biggest advantage of an integrated plant is the

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overall thermal efficiency, which is high enough to provide savings in the total energy required to operate the plant. Once the N_2 is separated, it can be recompressed and injected or vented as required by the overall production requirements.

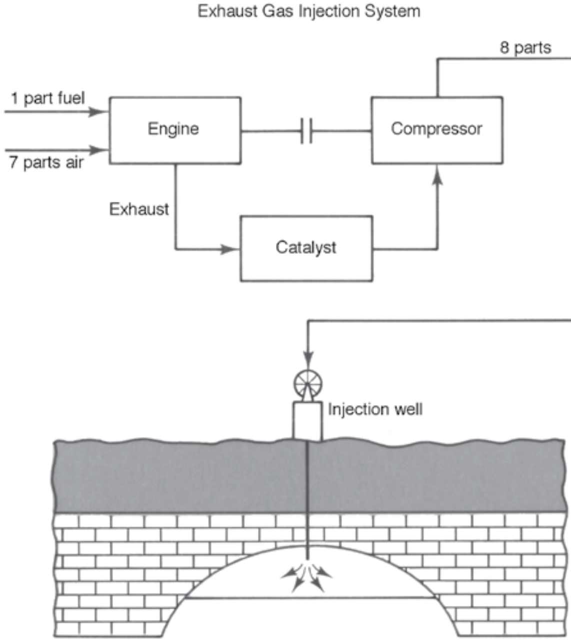

Figure 7.21 Engine exhaust can yield a noncorrosive inert gas.

Exhaust Gas. Engine exhaust gas, when properly treated by a catalyst, can yield a noncorrosive inert gas (Fig. 7.21). This treated gas acts as an economical substitute for natural gas in oil-field applications in pressure maintenance and miscible flooding. Injection gas is typically 88% N_2 and 12% CO_2 Injection pressure is dependent on the individual field. Generally, the volume of fuel gas required to injection gas is 1 to 8. The engine is used to drive a reciprocating compressor, which in turn compresses the treated exhaust gas to injection pressure.

Fuel Gas Boosting for Prime Movers

Reciprocating compressors are often used to increase the pressure of the hydrocarbon fuel gas used for operating gas engines or gas turbine drivers. Suction pressures range from 10 to 50 psig (low end in landfill gathering systems; high end in refinery or utility distribution headers) and discharge pressures range from 40 to 400 psig (low end, engines; high end, gas turbines). Horsepower requirements are generally less than 800 hp; however, systems requiring up to 1500 hp have been proposed. Control systems employing clearance pockets, cylinder end unloading, and automated bypass are required to control the gas flow required by gas engine or gas turbine generator sets.

With more emphasis placed on overall energy costs, cogeneration is quickly becoming a more important field. Cogeneration is the production of electricity while harnessing the waste heat of the prime mover. This process can provide overall thermal efficiencies of 75 to 80%. Fuel gas boosters are common accessory requirements to provide fuel gas at pressure for the prime mover.

GAS PLANTS

General Information

The term gas plant, as used in the natural gas industry, generally refers to a general class of plants designed to remove the more valuable, heavier components from the less valuable methane in a natural gas stream. These heavier components (ethane, propane, butane, and heavier gasolines) are removed by four basic means, as described next.

Absorption Process

This process uses absorption of the higher molecular weight gases into an oil medium to separate the heavier components from the gas stream (Fig. 7.22). Oil will absorb natural gas components in roughly inverse relationship to those same components in the oil. Thus, if the partial pressure of butane is reduced in an oil stream (which is then called lean oil), it will absorb a proportionately higher amount of butane from the gas stream. This oil can then be heated to drive off the gas components and thereby accomplish the separation.

Conventional Refrigeration Process

This process of separating the gas components from the gas stream uses cooling of the gas stream. The heavier components will condense first, thus allowing separation. Cooling can be done by conventional refrigeration systems in the range from +20 to -30°F, using Freon, butane, or propane refrigerant. A simplified schematic diagram of this system is shown in Fig. 7.23.

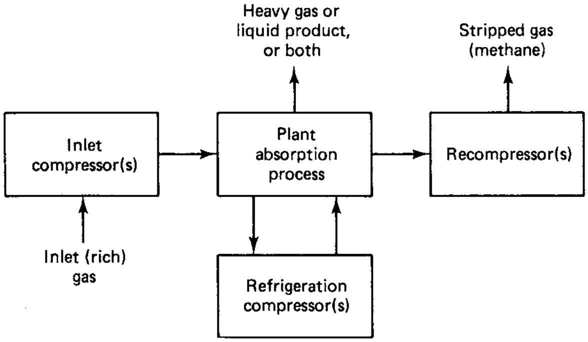

Figure 7.22 Absorbtion Process

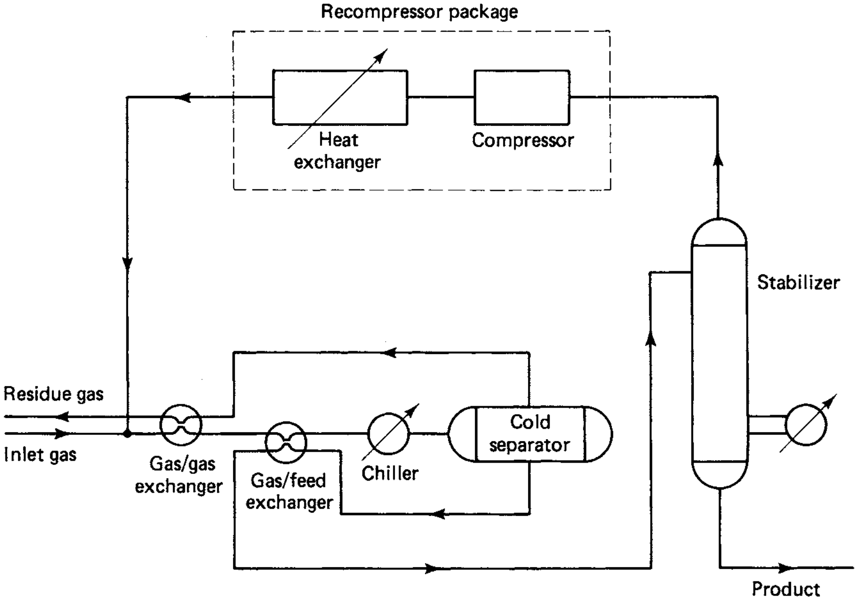

Figure 7.23 Refrigeration System

Expander Refrigeration Process

This process of separation is also a refrigeration process, and it utilizes expansion of the gas stream, as usual, to obtain the refrigeration. Utilizing a turboexpander or an expansion valve, the system achieves temperatures of -100 to -180°F. As ethane became more valuable as feed stock for plastics and chemical plants, it became economically justifiable to achieve the much colder temperatures necessary to separate this component.

The condensing temperature of ethane (C_2H_6) is -130 to 150°F. Expanding the gas stream through a valve or mechanical energy extracting device (reciprocating engine or centrifugal turbine) for recovering energy provides temperatures low enough to recover ethane. Figure 7.24 shows a typical expander plant. The majority of plants built today over 20 million scfd are expander plants. The turboexpander can be used to drive a centrifugal compressor and recover 10 to 12% of the pressure lost through the expansion process.

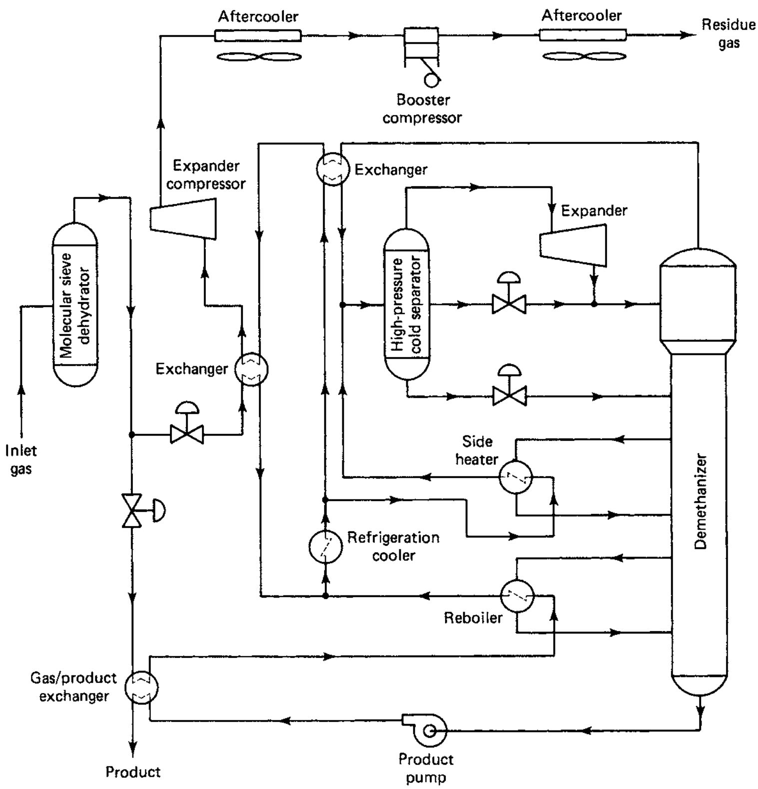

Figure 7.24 Typical Expander Plant

Membrane Separation Process

The membrane separation process is relatively new. Gas separation by membranes is based on the principle that some gases permeate much more rapidly than others due to their solubilities in a given membrane material. Use of membranes, to separate gases should be viewed as a bulk separation process, whereby a gas mixture is split into two enriched streams, each containing a portion of all the original components. Membrane separation is generally considered within the following limitations:

1. Availability of low- to medium-sized feed streams (150 million scfd to 200 million scfd).
2. Moderate concentrations of the more permeable gas in the feed stream (10 to 85 mol percent).
3. Moderate- to high-feed pressure (250 to 2000 psig).
4. Moderate-feed temperature (30 to 150°F).
5. Acceptability of moderate recoveries (85 to 95%) and product purities less than 95%.

Membrane separation can be tailored to meet specific requirements of purity and recovery and optimization of costs between separators and compressors is possible. In general, high purity means modest recovery, and low differential pressure across the membrane requires more membrane surface, and a high differential pressure requires more compression equipment. Either reciprocating or centrifugal compressors may be used depending on the pressure and the flow requirements. A simplified schematic diagram of this system is shown in Fig. 7.25.

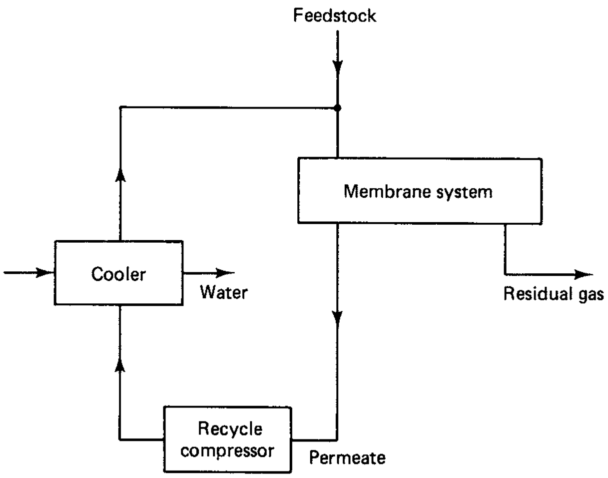

Figure 7.25 Membrane Separation System

With increasing interest in enhanced oil and gas recovery using carbon dioxide and nitrogen flooding of reservoirs, there is escalating interest and need for separation processes to separate methane from carbon dioxide, carbon dioxide from ethane and heavier liquids, and to separate carbon dioxide and hydrogen sulfide. Various companies and individuals have developed several specialized processes to achieve these separations. Several have been patented in recent years. All these processes require compressors with the specific size and types depending on the particular plant size and the process being used.

Mini-plants

Plants have become smaller over the years. Both refrigeration and absorption plants are now compact enough to be delivered on a single truck (Fig. 7.26). Modularizing these mini-plants has allowed manufacturers to make them on a mass-production basis in a factory as opposed to a job site fabrication in the field.

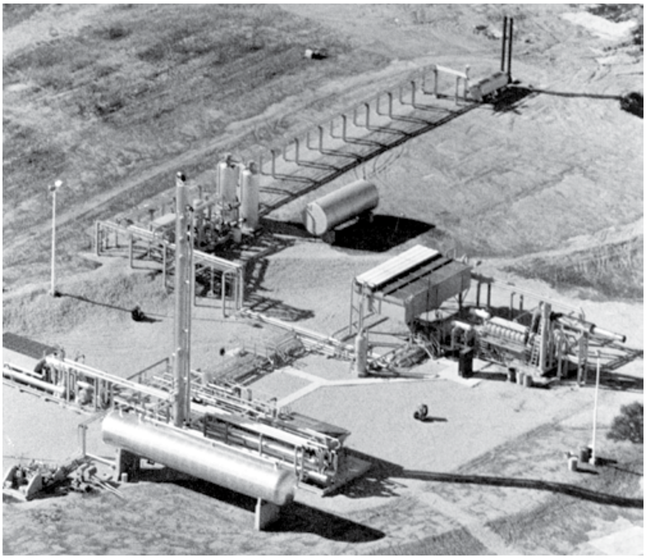

Figure 7.26 Expander Plant

Elements of the plant can be packaged separately and assembled in the field (Fig. 7.27). Reciprocating, or centrifugal compressors, or both, and turboexpanders are commonly used in these mini-plants, with the capacity and pressure conditions varying widely depending on the application requirements.

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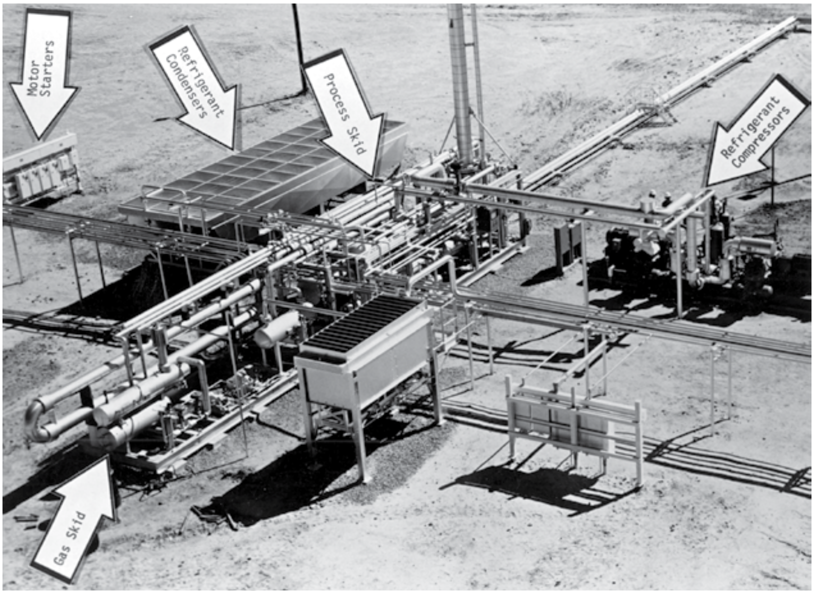

Figure 7.27 Gas Plant

Compressors in Gas Plants

Compressor requirements for a gas plant, like that in Fig. 7.28, vary widely depending on the type and size of the plant (100 to 1000 million scfd) and the makeup of the gas stream. The two primary elements that determine the type of compression used are as follows:

1. The performance flexibility required.
2. The energy balance of the plant.

These two items usually overshadow first cost, making it a distant third. Larger plants tend to use centrifugal compressors with turbines, either gas or steam, as drivers. Large-capacity and relatively stable gas conditions make the choice of centrifugal compressors practical on the basis of efficiency and installed cost.

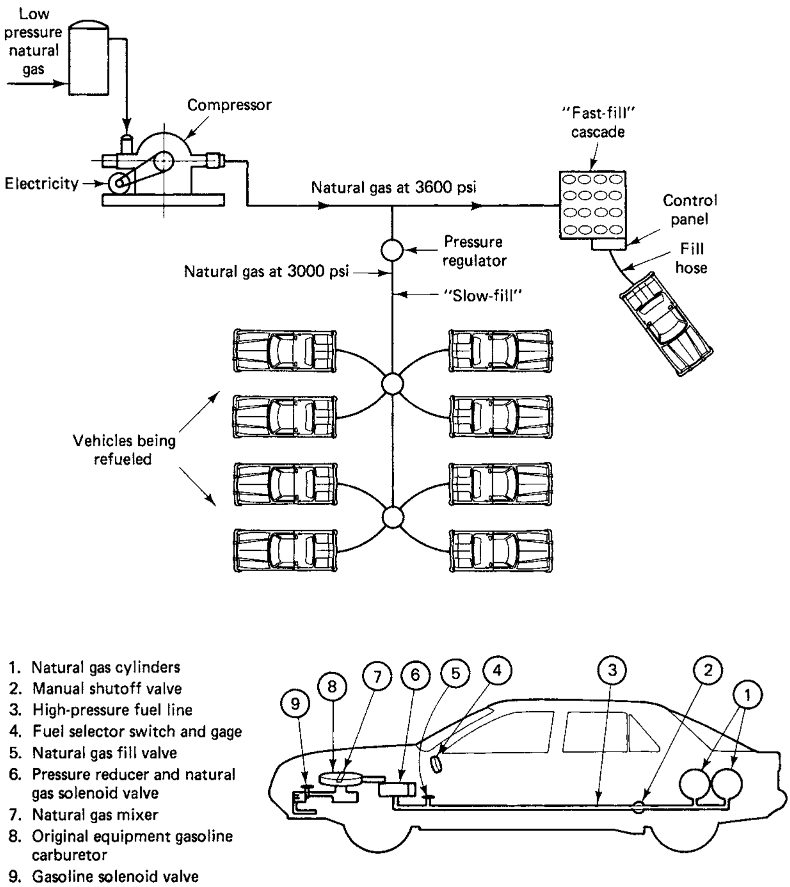

Figure 7.28 Components of a CNG combination refueling system.

As plants get smaller reciprocating compressors become more practical. Internal combustion engines powered with natural gas are the obvious choice for drivers. Recently, environmental pressures are causing the choice of electrical motors to become more prevalent. New developments in electronic controls are giving speed flexibility to electric motors, making them more acceptable for compressor drivers. Likewise, new developments in natural gas combustion technology have resulted in the availability of clean burning gas engines that meet most current environmental requirements, have relatively good fuel economy, and produce acceptable operating records.

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Inlet and Recompression Compressors

Treating gas by passing it through a process plant requires a pressure drop. This pressure drop sometimes must he recovered before the gas enters or reenters a pipeline. This pressure loss, typically 50 to 100 psig across a gas plant, is made up by an inlet compressor or a recompressor, or both, depending on the initial inlet pressure of the gas and what is done with the gas after processing.

An expander plant takes a much greater pressure drop, typically 400 to 600 psi. A good rule of thumb for the most economical design is the pressure drop through the expander will be one-half of the final discharge pressure to the pipeline. If the pressure is low going into the plant, an inlet compressor will be required to compress the gas up to about 800 to 1000 psig, which is required to economically operate the expander. A recompressor may be required after the expander to raise the gas pressure up to the pipeline level or for reinjection.

Normally, in an expander plant that discharges to a sales pipeline, a total of 5000 to 6000 horsepower will be required per 100 million scfd of inlet gas. Plants requiring less than 10,000 horsepower will normally use gas engine drivers in lieu of gas turbines unless the waste heat from the turbine can be efficiently used in the process. This is due to the better fuel efficiency of the reciprocating gas engine compared to that of the smaller gas turbines.

As regenerative cycle gas turbines become more proved and accepted, this condition may change. Due to the high horsepower often required, environmental considerations may necessitate gas turbines, electric motors, or clean-burning gas engines with systems to reduce NOX (nitrogen products from combustion), CO, and hydrocarbon pollutants. Regulations vary from state to state, and the laws and regulations covering the installations should be thoroughly researched.

Refrigeration Compressors

Refrigeration compressors are required in absorption plants to drop the lean oil temperature. Smaller mini-plants normally use commercially available Freon systems for this refrigeration. For larger gas plants, propane refrigeration is most frequently used. Small to medium (5 to 100 million scfd) plants typically use a natural-gas, engine-driven reciprocating compressor for this application. Turbines may be economically advisable if the waste heat can be used. Propane refrigeration units require no gas cooling. Scrubbers are usually part of the refrigeration plant and not located on the compressor skid. Many times in the smaller plants the refrigeration is combined on one unit.

De-ethanizer Overhead Compressors

Once the desired components are separated from the methane, either by absorption, refrigeration, or expansion, they may he further divided by condensing them in towers specifically designed for the purpose. Each is designated by the component it is to remove – de-ethanizer for ethane, depropanizer for propane – and so on. In some large plants, the residue gas from these towers may require a small reciprocating compressor to raise the pressure of the gas to move it on through the process or to sales.

The de-ethanizer overhead gas on any type of plant may require compression. This gas will typically be at 40 to 60°F and 300 to 500 psig. Many times this is chilled in the process and pumped as a liquid product.

Still Overhead Compressors

This phase of refining is most commonly done with the still overhead in an absorption plant. When the inlet gas stream has contacted the lean oil, it becomes rich with the heavy components it has absorbed. This rich oil is then heated, and the valuable heavy ends are driven off in the still. This residue is then cooled, causing further liquification. The liquid product is handled by pumps, and is sold. However, some gas will remain and must be compressed to be further processed or sold. This service typically requires 300 to 500 horsepower per 100 million scfd of inlet gas transferred to the plant and is generally a reciprocating compressor application. Pressure ratios are small, normally 10 to 50 psi pressure rise, at process pressures of 400 to 800 psig.

Vapor Recovery

Hydrocarbon liquids of all types are stored in specially designed storage tanks and transported in tank cars. During storage or transit, the liquids are constantly generating hydrocarbon vapors. It is necessary to have a vapor-recovery compression system to collect the hydrocarbon vapors that have boiled off the hydrocarbon liquids, compress these vapors to a design pressure, and then cool the vapors. During the cooling process, liquids condense, and are collected and pumped back into a storage tank or pipeline.

Storage Tanks

During the storage or handling of hydrocarbon liquids with a low vapor pressure, there is a constant need to maintain the pressure in the tank at a safe value. A vapor-recovery system is designed to prevent leakage of vapors into the atmosphere and to prevent pressure buildup that could cause structural damage to a storage tank. Care must also be taken to maintain a slight positive pressure (normally in inches of water) in the tank to prevent inward collapse of the tank or entry of air into the tank.

Tanks used for storing volatile hydrocarbon liquids are connected together through a manifold, and a vapor-recovery system is used to maintain the tank battery vapor pressure at the design pressure. These applications require compressor systems that can compress vapors with entrained liquids. Typical compressors used are the sliding-vane, liquid-ring, and reciprocating types.

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These compressor systems, which average less than 25 hp, usually have very low suction pressures and can operate at a slight vacuum. The service is normally not continuous duty. Condensation, which occurs in the compressor during frequent shutdowns, dictates the need for liquid-handling characteristics to be designed into the compressor.

Tank Car Unloading

The requirement for volatile liquid unloading is similar to the storage tank service described previously. If feasible, the vapor-recovery system is used to maintain the design pressure in the tank being charged or discharged. A balance line is connected between the storage tank and the tank car during loading or unloading. The receiving tank is always loaded from the bottom to minimize vapor formation, and the vapor-recovery system takes care of any vapors that do form.

COMPRESSED NATURAL GAS FOR VEHICLE FUEL

History

The use of methane as a vehicular fuel was first introduced in Italy in the 1930s. Other countries subsequently using this fuel were Canada, New Zealand, the United Kingdom, Holland, Iran, and Australia. At this time, however, the ready availability and low price of gasoline have delayed significant developments in the compressed natural gas (CNO) market.

Interest continued to lag, until air pollution concerns prompted the search for cleaner-burning fuels. Some public utilities in the United States converted their vehicles to liquified natural gas (LNG) in order to demonstrate methane's potential as a clean-burning, nonpolluting fuel. During the same period, developmental work on an experimental CNG system was being carried out. The CNG system proved itself to be satisfactory and made possible the conversion of several hundreds of vehicles to methane fuel during the late 1960s and early 1970s. The latter half of the 1970s saw sharp curtailment of vehicle conversion due to publicity about natural gas shortages, whether real or contrived. In the 1980s, because of many factors, natural gas has again become more abundant, and a resurgence in its use as a vehicle fuel has occurred.

Systems

Generally, CNG systems are designed in three basic types: quick fill, slow fill, and combination. Components of the combination system are seen in Fig. 7.28.

1. *Quick fill:* This system utilizes a fairly large reciprocating compressor and a storage volume that can be transferred to the vehicle's storage bottles in approximately the same amount of time required to fuel a gasoline-powered vehicle.
2. *Slow fill:* Using this system, vehicles are charged directly from the compressor and may take several hours to fill. This is an unattended operation and is shut down when the required tank pressure is achieved. Slow-fill systems find their greatest potential in home use where an overnight fill is not objectionable.
3. *Combination:* This system provides a limited quick-fill capability in conjunction with the normal slow-fill operation. Small fleets or families having several vehicles may find the combination system most advantageous.

Compression

Suction pressures may vary from 1 to 50 psig, with discharge pressures generally around 3600 psig. These conditions will probably dictate the use of four stages of compression with intercooling. Power requirements can range from 3 to 150 bhp, depending on the size of the fueling system and the suction pressure available. The compressor must be a positive-displacement type, adequately sized, and properly instrumented to give trouble-free unattended service, as shown in Fig. 7.29.

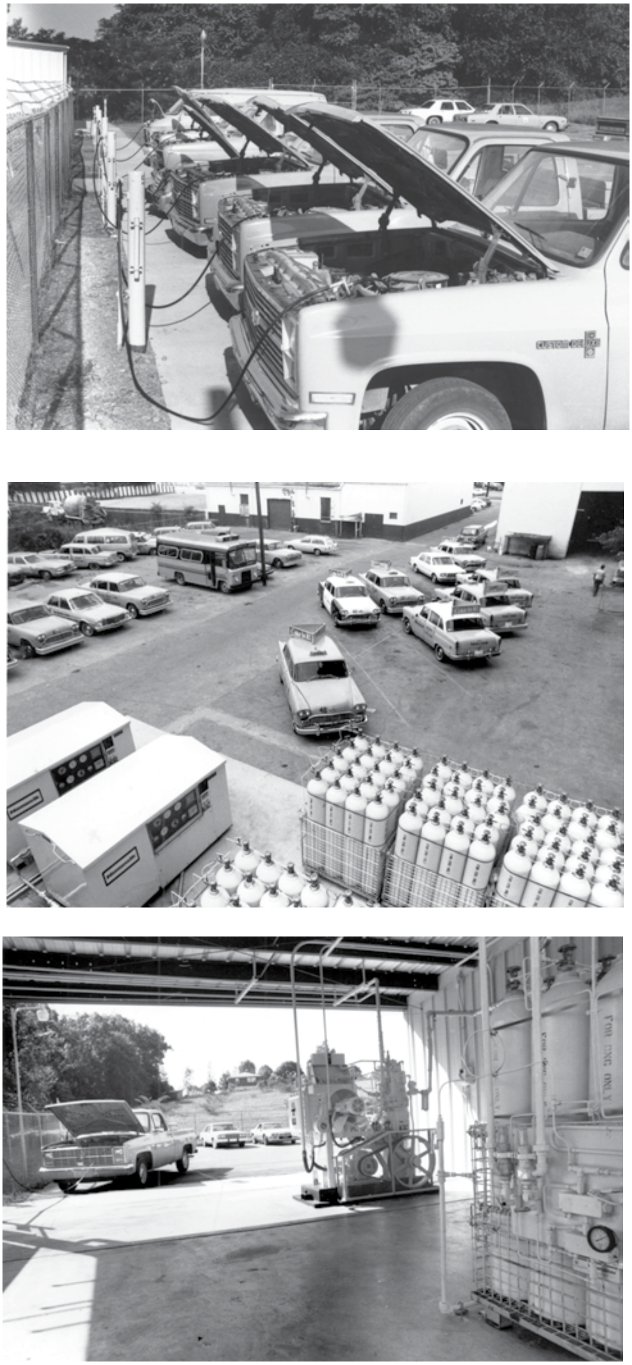

Figure 7.29 Compressed natural gas for vehicle fuel.

Operation

In operation, gas is taken from either a well or a supply line and compressed to a sufficiently high pressure to fill the storage tanks in the vehicle. The natural gas remains there in a vapor state until needed for fuel. After having been reduced in pressure and mixed properly with air, the gas is admitted to the engine's combustion chamber, where it is ignited. Engine performance drops slightly when the vehicle is tuned to CNG, but due to power reserves of most models, this had not been particularly noticeable.

Potential

As fueling centers become more widespread, the use of compressed natural gas, as shown in Fig. 7.29, is likely to increase. This will be due to its availability well into the twenty-first century, while petroleum liquid fuels will be subject to potential shortages until stability in the Middle East is achieved. Additionally, there are the advantages of reduced emissions, low cost, flexibility, longer engine life, and reduced dependence on imported products.

LANDFILL GAS APPLICATIONS

The generation of methane gas, a by-product of waste decomposition, including the anaerobic decay of organic material in municipal landfills, has been viewed in the past as somewhat of a problem. In some of the larger landfills in the United States, it was necessary to collect the gas to prevent its migration into adjacent commercial or residential properties where it could become an explosion hazard. It then had to be disposed of by incineration which was wasteful. The advent of natural gas shortages and escalating energy prices solved this problem, and today many landfill sites around the country are producing methane gas quite economically for commercial or industrial use.

A typical landfill must digest itself, or decompose, for 5 to 10 years before a commercially attractive quantity of gas is produced. To produce the gas, numerous wells, approximately 6 in. in diameter, are drilled to between 100 and 150 ft. in depth, and a gathering system is installed to collect the gas to a central location for scrubbing, dehydration, compressing, and processing. The expected life of most landfills is between 15 and 20 years.

The gas produced in a landfill can vary, but generally it is composed of about 52% methane and 46% carbon dioxide, with the remainder being nitrogen, oxygen, and miscellaneous other gases. The gas comes from the wells saturated with water at about 120°F and the heat content is around 500 Btu/ft^3.

To make the gas salable or usable, it first must be treated to remove offensive odors, water; and particulates. It can be used as low-energy fuel gas for gas engines, gas turbines, or gas-fired steam boilers and sold to a commercial on-site power generation or cogeneration project, or be further processed by molecular sieves to remove the carbon dioxide and enrich the Btu content to make the gas salable to a pipeline transmission company or utility.

Generally, reciprocating compressors are utilized in the compression of landfill gas; however, rotary positive displacement units can be used in lower discharge pressure applications.

Gas	Typical Pressures
Low Btu	
Suction (psig)	-1 to 0
Discharge (psig)	60 to 100
Enriched Btu	
Suction (psig)	-1 to 0
Discharge (psig)	300 to 600

AIR DRILLING

Rotary drilling with air results in the fastest penetration rate while cleaning the hole and removing cuttings for drilling 2000 to 10,000 ft. oil and gas wells. Unfortunately, hole instability and formation fluids allow air drilling of only about 10% of the footage in the United States, and 10,000 ft. is the normal maximum depth because of potential hole problems.

The earliest significant use of air for rotary drilling of gas wells occurred around 1956 in Pennsylvania, when several drilling companies started experimenting with various techniques and equipment. Simultaneously, other people in the Rockies, Arkansas, and West Texas found conditions adaptable to air drilling. Mining equipment and compressor manufacturers responded rapidly to supply the need for equipment adaptable to the rigors of oil field use.

Until the advent of portable, rotary-table-type rigs in the mid- 1960s, capable of drilling 3000 to 10,000 ft., most drilling was done with jackknife-type skid rigs with sufficient power at the tail shaft to run two small reciprocating air compressors. Normally, 1600 to 2400 cfm is required and pressures of up to 1000 psi were frequently required. A separate engine-driven reciprocating booster compressor was needed for almost all operations.

During the conversion of the drilling industry to diesel power, portable rotary rigs for under 10,000 ft. replaced skid rigs, thus creating a demand for new diesel-driven primary (low-pressure) reciprocating or rotary positive-displacement compressors. Drilling in the western United States and overseas evolved into requiring 2400 to 4000 scfm of air with capability of 1000 to 1500 psig discharge. This equipment is frequently rented by the operator of the well for the drilling contractor and usually is furnished by a rental or service company, complete with personnel. Air is frequently used in only the first 6000 to 10,000 ft. of a deep well, thus requiring more air and higher pressures, since larger hole sizes are required.

In the Appalachians, shallower drilling is prevalent and the wells are, for the most part, completed with air by contractors who own and operate their own air compressors. Hole diameter is generally smaller, and operating techniques are different, resulting in the normal complement of air being 1600 to 2400 scfm, capable of 500 to 600 psig pressure.

Simultaneously, high-pressure, rotary-screw compressors, with 200 to 400 psi discharge pressure have started to fill a niche in two types of applications. For shallower holes, throughout the air drilling areas of the oil fields, hydraulic, top-drive, pulldown-type rigs are becoming competitive. These rigs have on-board, high-pressure, rotary-screw compressors and can be used with an auxiliary compressor for 3000 to 4000 ft. wells. Reciprocating boosters are occasionally required. Where pressures of 1000 to 1500 psig are required, two-stage boosters must be supplied with air at 200 to 300 psig for satisfactory staging.

With the frequent moves required, averaging 5 to 10 days for portable rigs and 30 to 90 days for deep rigs, the package size will continue to dictate the type of equipment used. Road limits and location size almost mandate a unit no larger than 8 ft. wide, 8 ft. 6 in. tall, 40 ft. overall length, and 35,000 lb maximum weight. Operators are seeing increasing use of air, with improved techniques for using soap, foam, and stiff foam, all of which provide increased penetration rates over mud or fluid drilling. Portable and skid mounted rigs are discussed and illustrated in Chapter 1.

Air compressors are being used increasingly in other oil field applications, such as air notching, well servicing, and clean-out.

Positive-Displacement Gas Compressors

TYPES, ARRANGEMENTS, AND DESIGN DETAILS

Positive-displacement gas compressors are mechanical devices designed to meet a wide range of flow requirements, with inlet and discharge pressures ranging from vacuum to thousands of pounds per square inch. The various types of positive-displacement machines are defined in Chapter 2.

Process Compressors

The reciprocating type of compressor used in compressing gases has a distinctively different design from a 100 psig air compressor. To customize the compressor design for variations in pressure levels and capacity, the gas compressor cylinders can be arranged in any of the following configurations:

Type L (Fig. 7.30)
Type Y (Fig. 7.31)
Type W (Fig. 7.32)
Multiple crank throw, horizontal opposed (Fig. 7.33)

For process applications, the multiple-throw, horizontal-opposed construction provides maximum flexibility. A multiple number of stages for one or more than one service can be combined on one machine. Services requiring a large intake volume can be arranged in a multiple number of compressor cylinders. Figure 7.33 shows two first-stage cylinders for a three-stage, 8000 hp carbon dioxide compressor on enhanced recovery service.

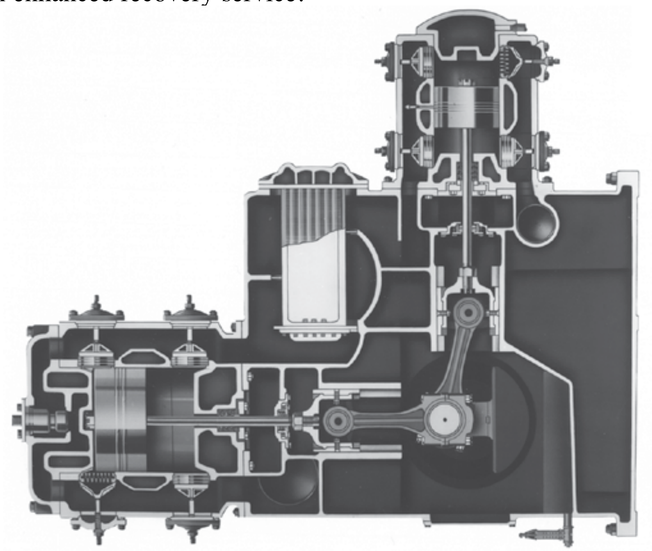

Figure 7.30 Type "L"

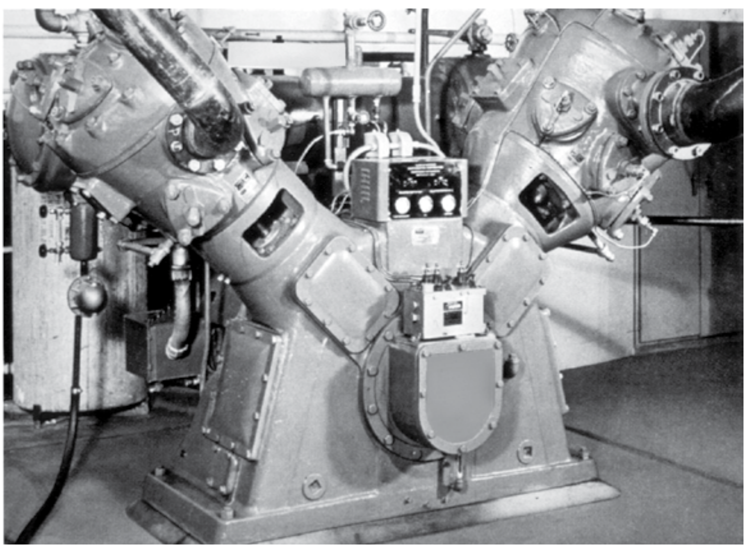

Figure 7.31 Type "Y"

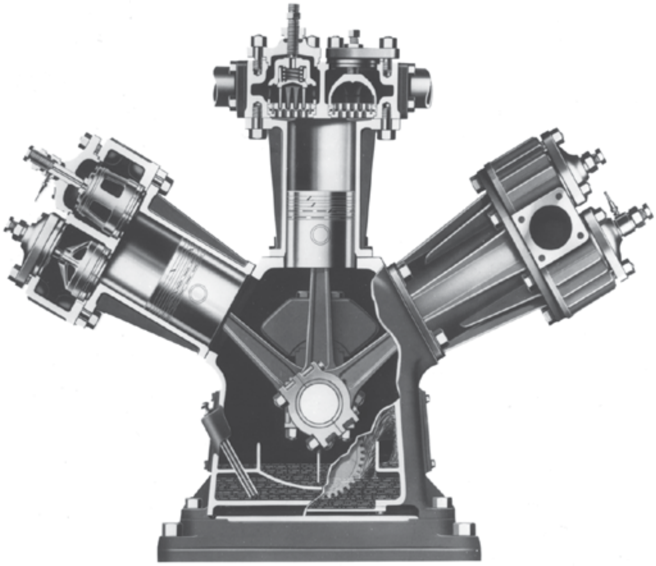

Figure 7.32 Type "W"

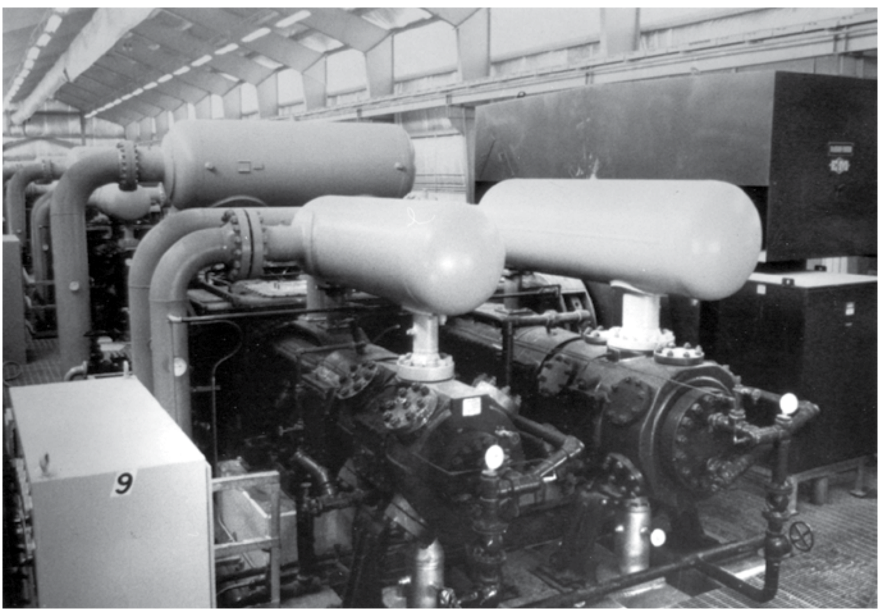

Figure 7.33 Three-stage, 6,000 hp balanced opposed carbon dioxide compressor.

The driving systems of large process reciprocating compressors are of various types. The most common is the synchronous motor. Frame arrangements of synchronous motors are as follows:

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1. Two-bearing synchronous motor: In this design, the motor manufacturer supplies the motor shaft and bearings within the motor frame.
2. Engine-type, low-speed motor (Fig. 7.34): The engine-type synchronous motor is the most frequent choice for slow-speed, high-horsepower process reciprocating compressors. The motor is furnished without shaft or bearings. The compressor supplier furnishes the motor extension shaft and outboard bearing.

The large diameter and narrow width of the synchronous motor permits mounting of the rotor on the shaft extension. This provides considerable rotating inertia, eliminating the need for, or reducing the size of, a compressor flywheel. The motor stator and outboard bearing are supported on the compressor foundation. The compressor manufacturer should furnish the motor because it is an integral part of the design and engineering of the compressor system.

Induction motor drives can be direct connected or coupled, flange mounted, V-belt, or gear reducer drive. The responsibility to design the drive system to keep current pulsation within the required limits generally is assigned to the compressor manufacturer. If a gear reducer is applied, the drive system must be designed and analyzed for torsion to prevent torque reversals and excessive peak torques at the gear teeth. Negative torque would cause gear teeth separation, resulting in noisy gears and premature wear of the gear teeth.

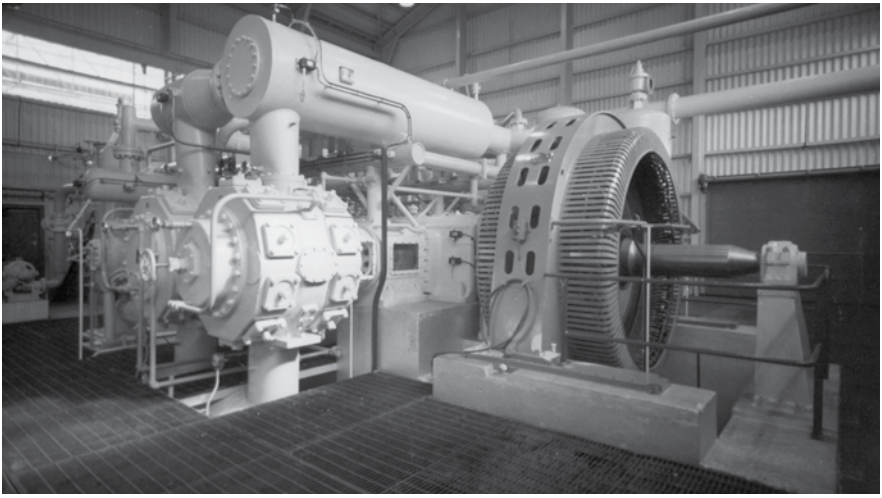

Figure 7.34 Engine type synchronous slow speed compressor driving a five-stage compressor.

Electric motors are available in many enclosures. The proper enclosure is a function of the environment where the motor will be installed. Motors can be designed with many different enclosures, such as open, drip proof, weather protected I or II, inert gas filled, and forced ventilated.

The compressor manufacturer determines the necessary additional amount of flywheel or rotor weight required. The normal maximum permitted current pulsations as specified by American Institute of Electrical Engineers (AIEE) standards is 66%, but the specific requirements should be specified by the user to suit the particular system requirements.

Torsional vibration analysis is an important design consideration in the selection of couplings and gearing for drive trains of large reciprocating compressors. The torsional reciprocating compressor analysis consists of taking the various masses of the complete shafting system of the installation and evaluating how they turn against each other. Engine and compressor manufacturers have been doing these types of analyses for over 50 years.

The objective of a reciprocating compressor analysis is to determine:

1. The various torsional natural frequencies that exist within the complete shafting system.
2. The frequencies and magnitude of fluctuating torques within the system.
3. The critical speeds of the system, where the frequency of a torque fluctuation coincides with a susceptible natural frequency (e.g., resonance) where torsional failures could occur.
4. That the critical speeds are moved outside the design operating speed range. (The system is tuned by varying the size of the masses [e.g., fly wheels, counterweights, etc.] or the rings [selection of couplings, V-belt sizes, etc.])

The following paragraphs are general guidelines for the torsional vibration analysis of complex or new combinations of reciprocating compressors or for equipment operating over a relatively broad speed range.

The natural frequencies of all modes of vibrations up to 20% above the highest known exciting frequency should be determined. Guides for exciting frequencies are as follows:

Engines and reciprocating compressors: twelve times basic speed.
Motors and generators: two times basic speed.
Turbines and other rotating equipment: two times basic speed.

There are other potential exciting frequencies (e.g., the number of gear teeth on each gear times the gear speed, or the number of turbine blades times the turbine speed); however, these frequencies are generally so high that it should be safe to ignore them.

There are exciting frequencies at which the magnitude of the exciting torques is not known or not readily determined; therefore, resonance with these frequencies should be avoided. At resonance with a responsive mode of vibration, the effect of a small force can be magnified to dangerous limits. Therefore, lacking a more complete analysis, natural frequencies should be avoided that fall within the range of 10% above the maximum operating speed to 10% below the minimum operating speed of the following equipment:

Reciprocating equipment.
Lobed or vaned rotors (up to eight lobes or vanes) – the number of lobes or vanes times the rotative speed.

A deeper investigation may reveal that the natural frequency in question may not be responsive.

Natural frequencies for the lower modes of vibrations should not fall within 10% of each other. This is to ensure that two modes of vibration do not couple and mutually excite each other.

Except in the case of synchronous-motor-driven equipment, there is no such thing as a constant-speed compressor. Whenever the speed of the driver can be varied (i.e., the speed of the engine or turbine), then the permissible speed limits for the installation must be clearly defined and the torsional analysis must ensure that the installation is torsionally safe within these limits. For compressors considered as constant-speed units for which no fixed speed limit can or will be defined, then the analysis should ensure that the installation is torsionally safe within the 10% range of the rotative speed.

The degree of analysis required from the torsional vibration aspect will vary with any project. The following factors will increase the complexity of the analysis:

1. Broadening the permissible operating speed range.
2. Broadening the differences in the rotative speed between driven and driving equipment.
3. Increasing the sources of excitation (i.e., reciprocating equipment driving reciprocating equipment).

Oil and Gas Field Compressors

Oil and gas field applications require compressor systems that are compact and can be easily moved from one location to another. These machines are similar in configuration to the process machines discussed previously. The normal drives for these compressors are coupled gas engines or electric motors. These reciprocating units are called "separables" in the oil and gas industry. For compactness, such units feature a short stroke while operating at higher rotative speeds. The piston speed is maintained typically from 750 to 1200 fpm (Fig. 7.35). They are usually assembled into a readily portable package consisting of mounting skid, driver, cooling equipment, piping and controls (Fig. 7.36).

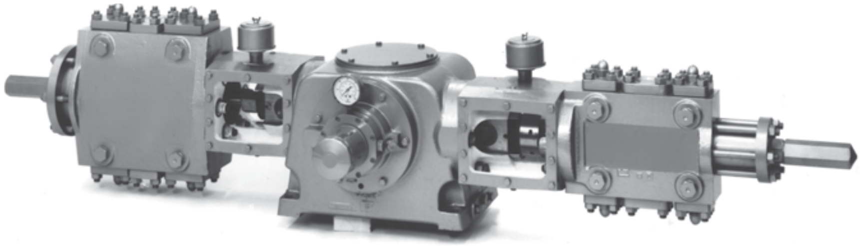

Figure 7.35 Two stage, two throw horizontal opposed, 200 hp (reciprocating gas booster compressor.

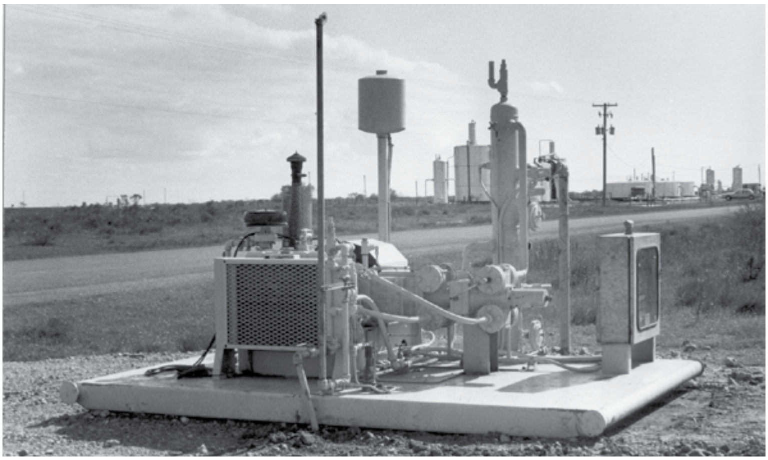

Figure 7.36 Two stage, 13 hp portable package compressing 60,000 cfd of flash gas from oil well separator into 400 psig gas gathering system.

Integral Gas Engine Compressor

Another type of reciprocating compressor is the integral gas engine compressor. The typical configuration consists of a crankcase with a horizontal crankshaft. The power cylinders deliver power through combustion of natural gas. These cylinders may be mounted vertically in-line, in a V-type vertical configuration, or horizontally, delivering the power through connecting rods to the crankshaft. Compressor cylinders are horizontally mounted to the same crankcase and crankshaft as are the power cylinders.

The integral gas engine compressor is built in sizes from 25 to 12000 horsepower. Primary oil and gas field applications are natural gas storage plants, natural gasoline, liquefied petroleum gas plants, and natural gas transmission pipe-lines.

There are two basic types of gas engines, the two-stroke cycle and four-stroke cycle, and either type can be turbocharged. The two-cycle engines require less displacement for the same rating. The differences in performance between the two-cycle and four-cycle engines are small, especially in the turbocharged models.

Rotary Positive-Displacement Gas Compressors

Sliding-Vane

The rotary sliding-vane compressor is primarily a 100 psig air compressor. Chapter 2 has further information on this device and its applications. It has been utilized quite successfully, however, on natural gas applications at lower pressures. The gas must not contain components that would deposit on the rotor, especially any sticky material such as tar. Such impurities would prevent free movement of the vanes in and out of the rotor.

Helical-screw-type Gas Compressors

The largest use of this compressor in handling gas is for industrial refrigeration. This chapter has discussions of both the reciprocating compressor and helical-screw compressor, including applications to refrigeration. Efficiency must be a consideration and must be evaluated. The helical-screw machine is different from other positive-displacement compressors since its efficiency curve is the island type. The maximum efficiency is at the designed fixed compression ratio. If, for example, the built-in compression ratio were 3:1, the adiabatic efficiency would have to be 82%.

At compression ratios of 3.6:1, 4.0:1, and 4.8:1, the approximate compression efficiencies would be 79.5%, 78.5%, and 76%, respectively. The adiabatic efficiencies of the reciprocating compressors at the same ratios are approximately 87%, 88% and 89.5%, respectively. This is not a complete evaluation between the two compressor designs, nor would the comparison be of much value for a service that operated continuously at the same compression ratio.

To compress gases other than air, certain adaptations must be made in the design. Light gases such as ammonia, helium, and hydrogen require significantly higher rotor tip speeds or oil-flooded designs. By injecting oil between the rotors, leakage of gas back to the inlet side of the compressor is reduced. A secondary advantage is that the oil removes a great deal of heat of compression. The removal of heat extends the allowable compression ratio per stage. The heat absorbed by the oil is removed in an external heat exchanger. Liquids other than oil may be used for injection in some applications.

Some designs of helical-screw compressors provide a means of varying the capacity. The device consists of a sliding control valve that changes the point at which the rotor length begins compression. This provides a stepless variation in capacity downward to about 20%.

Reciprocating Compressors: Type and Construction

Reciprocating compressors are built in a multitude of horsepower and speed ranges. The following covers the various types and construction details.

A. Frames

1. Single-throw

The single-throw is available in both the horizontal and vertical arrangements and is generally applied up to about 300 horsepower, with an upper limit of about 500 horsepower. Single cylinders or multiple tandem cylinders can be used with the single crank, which is normallybelt driven, but may be direct through reducing gears driven by slow-speed motors or by high-speed turbines or motors. The single-crank design yields unbalanced inertia forces that must be absorbed by the skid and foundation (Fig. 7.37).

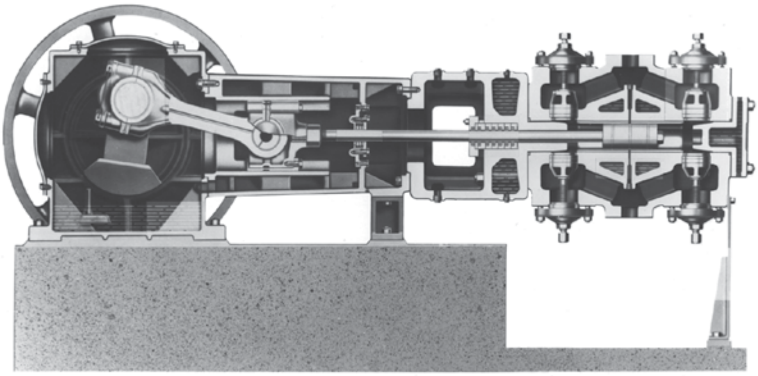

Figure 7.37 Horizontal, Single-throw Reciprocating Gas Compressor

2. Multi-throw, Horizontal, Balanced-Opposed Frame

As discussed earlier, this design is extremely flexible, lending itself to multiple cylinders arranged opposite each other on either side of the frame. For process services, several stages or even different services can be placed on the same frame. When two cylinders with equal reciprocating weights are located on opposite sides of a frame and are powered by a double-throw crankshaft with cranks set at 180°, all primary and secondary inertia forces developed mutually cancel each other, as may be seen in Fig. 7.38. Only couples are transmitted to the foundation. As many as five pairs of crank throws can be arranged on one compressor frame.

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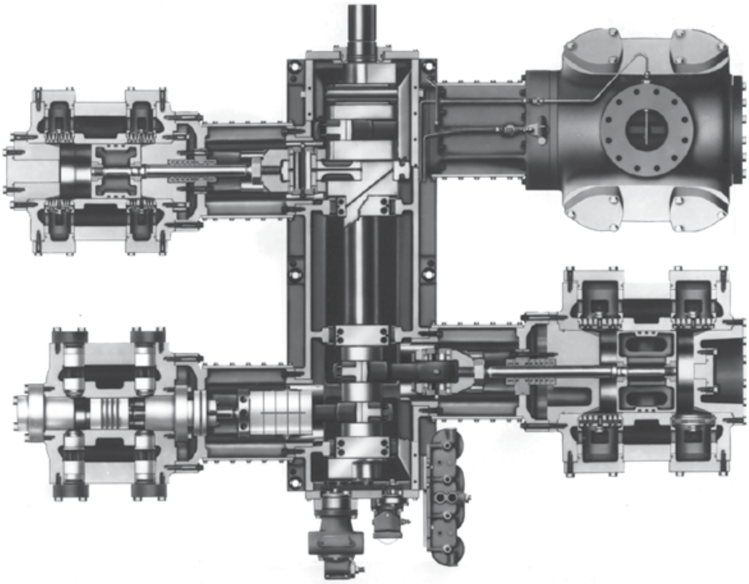

Figure 7.38 Balanced Opposed Compressor Design

Unbalanced forces cause mechanical vibrations that can result in alignment, piping, and vibration problems. These unbalanced forces areproduced by two masses, rotating and reciprocating. Rotating masses consist of the crankpin, crankpin web, and approximately two-thirds of the mass of the connecting rod. The centrifugal force produced by these masses is an exciting force and has the same intensity throughout the revolution.

Reciprocating masses consist of the piston, piston rod, crossbead,crosshead pin, and the remaining one-third of the connecting rod mass. Acceleration and deceleration of the reciprocating masses cause reciprocating forces, which act along the axis of the cylinder and exert a variable force on the crankpin. The forces resulting from the rotating and reciprocating masses are generally resolved into force systems consisting of two parts, primary and secondary forces. These are expressed in both the horizontal and vertical direction. In the case of multicrank compressors, there are also moments or couples.

The more compact oil and gas field compressors utilize the principles just described, the principal difference being that they have shorter strokes and higher rotative speeds than is the case with process compressors. To reduce inertia forces, the stroke of the piston is in the range of 3 to 8 in., compared to process electric-motor-driven compressors, which have strokes of 9 to 18 in. The advantages of the higher-speed unit are its compactness, portability, and driver flexibility

B. Cylinders

Two basic types of cylinders are used on reciprocating compressors in industry, water cooled and noncooled.

1. Water-cooled Cylinders

A water-cooled cylinder includes a waterjacket that normally is located around the bore and in both the head and crank-end cylinder heads. External passages eliminate the necessity of water passing through gasketed internal joints. Water, under pressure, is circulated through these jacketed areas:

a. To reduce thermal stresses.
b. To maintain a relatively cool cylinder bore.
c. To carry away a small amount of the heat of compression.

Another benefit of water cooling is the maintenance of a high-viscosity lubricating oil film to reduce friction and wear in the cylinder bore. This is of particular importance in corrosive gas services with large-diameter, long-stroke cylinders. The jacketcooling water should always be 10 to 20°F above the suction gas temperature to avoid condensation within the cylinder bore.

A thermosyphon cooling system for a water-cooled cylinder (Fig. 7.39) may be supplied where the gas discharge temperatures are between 190 and 210°F or the adiabatic gas temperature rise in compression is less than 150°F. Static cooling of water-cooled cylinders may be supplied when the gas discharge temperatures are less than 190°F or the adiabatic gas temperature rise in compression is less than 150°F. A mixture of equal parts glycol and water is usually employed except in extreme cold-climate outdoor applications.

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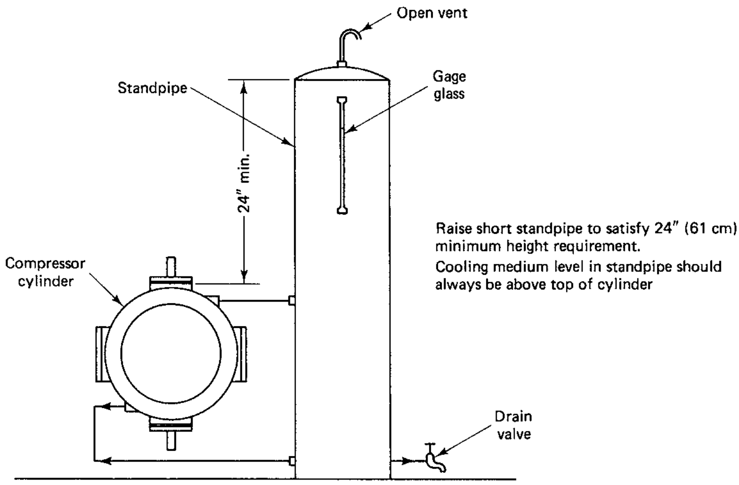

Figure 7.39 Thermo-syphon system with standpipe. Note: for use with gases having a *k* value (C_p/C_v) of 1.26 and below; with cylinder discharge temperature (adiabatic) over 210°F, up to and including 230°F.

2. *Non-cooled Cylinders*

Non-cooled compressor cylinders are used more frequently in a variety of oil and gas field applications. Perhaps the best known application is pipeline transmission, for which the compressor cylinders operate with low compression ratios and, consequently, experience only a small temperature rise, with discharge temperatures in the range of 140 to 180°F.

A noncooled cylinder (Fig. 7.40) is identical in basic design to a water-cooled cylinder except that there are no jacket-water cooling passages.

Most compressor units today are offered as packaged systems with gas engine prime movers (Fig. 7.86). Most applications for non-cooled cylinders occur where the gas inlet temperatures are 120°F and below, and the only means of cooling is to use gas engine jacket water at a temperature of 160 to 180°F. There is reasonable doubt as to the quantity of the heat rejected.

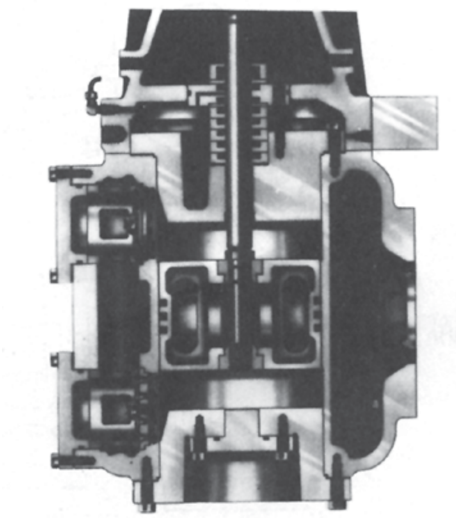

Figure 7.40 Non-cooled Cylinder

3. Compressor Cylinder Designs

Three cylinder designs are used in the process and gas industry. Horizontal cylinders with top intake and bottom discharge are preferred for saturated gases. Most cylinders can be fitted with liners or alternately must be suitable for up to 0.125-in. diameter of reboring without affecting the maximum allowable working pressure.The maximum allowable working pressure shall exceed the rated discharge pressure by at least 10% or 25 psi, whichever is greater.

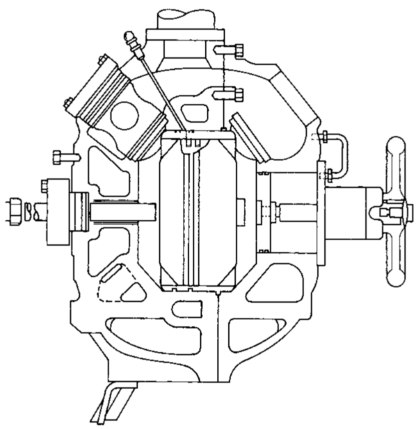

Figure 7.41 Two-piece clam-shell (gas field) low pressure cylinder.

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4. *Compressor Cylinders*

a. Low Pressure (Gas Field)

Two-piece, clam-shell-type cylinders, with liner and with valves in both heads, and three-piece cylinders with valves in the barrel are used for many applications (Figs. 7.41 and 7.42). The piston is of two-piece construction to assure accurate wall thickness and to eliminate core plugs, which might work loose during compressor operation. The piston is anodized aluminum impregnated with TFE particles to provide resistance to ring groove wear or bore scuffing.

TFE-filled pistons and packing rings should be used where the gas contains entrained liquids and traces of H_2S. The packing is vented, fill-floating, segmental-ring type with each cup containing one radial and one tangential ring, except for the last cup, which has double tangential rings to seal in both directions (Fig. 7.49).

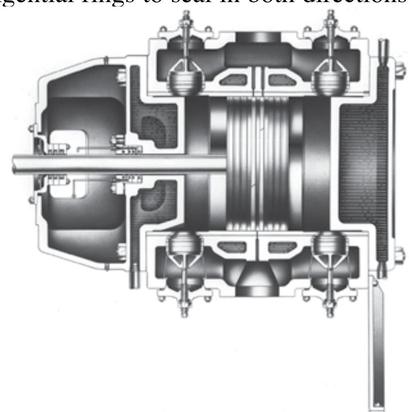

Figure 7.42 Three-piece low to medium pressure cylinder (gas field and process).

b. Low to Medium Pressure (Gas Field and Process)

The cylinder materials for a three-piece cylinder designed for medium pressure can be either gray iron, nodular iron, cast steel, or fabricated steel, depending on bore size and pressure. A general guide is as follows; it should not he used in place of regular design procedures:

	Maximum Allowable Working Pressure psig
Gray iron	1600
Nodular iron	2600
Cast steel	2600
Forged Steel	Over 2600

Pressure-containing castings of ASTM A278 for gray iron, ASTM A395 for nodular iron, and ASTM A216 for cast steel are suggested. Ring materials are similar to low-pressure designs. Pistons may be aluminum, gray or nodular iron, or steel, depending on weight and differential pressure.

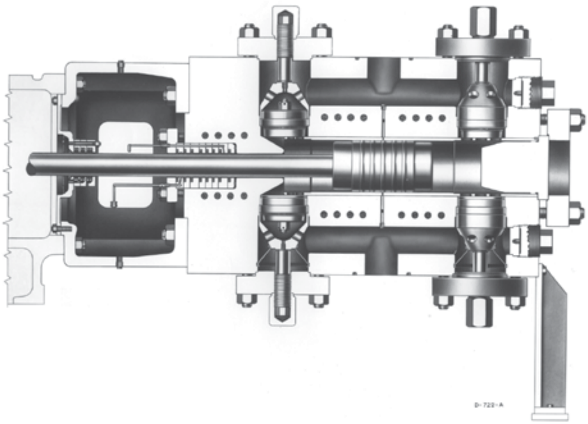

Figure 7.43 High Pressure, Forged Steel Cylinder

c. High Pressure

Forged-steel cylinders are required for pressures above approximately 2600 psig (Fig. 7.43). In Fig. 7.43, the cylinder and packing box are water cooled.

5. Compressor Cylinder Arrangements

a. Single Acting

In a single-acting compressor, gas compression takes place on only one end of the cylinder. The other end is vented to a lower pressure. Care should be taken to ensure that a crosshead pin reversal is achieved to provide proper crosshead pin bearing lubrication and long life. The load on the crosshead pin must reverse direction relative to the pin during each full crankshaft rotation.

b. Double Acting

In a double-acting compressor, gas compression takes place on both the crank end and head end of the cylinder.

c. Divided Cylinder

The outer end (head end) is the low-pressure stage of compression, and the frame end (crank end) of the cylinder provides the high-pressure stage of compression. A forged-steel cylinder may be used with separate suction and discharge passages machined into each end of the billet.

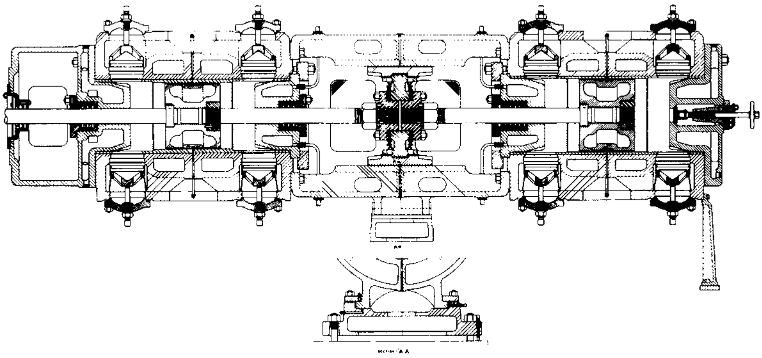

Figure 7.44 Double-acting, tandem cylinders, two-stage with high pressure cylinder adjacent to frame.

d. Tandem Cylinders

Two or more cylinders are arranged on a common piston rod for a single compressor throw. Cylinders can be double-acting (Fig. 7.44) or single-acting, depending on frame load and crosshead pin reversal design criteria (Fig. 7.45).

e. Steeple

The steeple type is defined as two or more stages utilizing a common step-type or trunk-type piston. Normally, the low-pressure stage is double-acting, while the subsequent stages are single-acting. This arrangement will permit up to five stages on a single compressor throw.

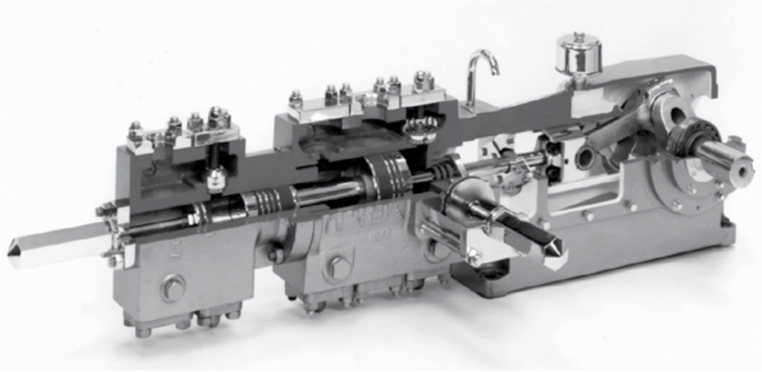

Figure 7.45 Non-cooled tandem cylinders, two-stage, single-acting, low-pressure stage, adjacent to frame.

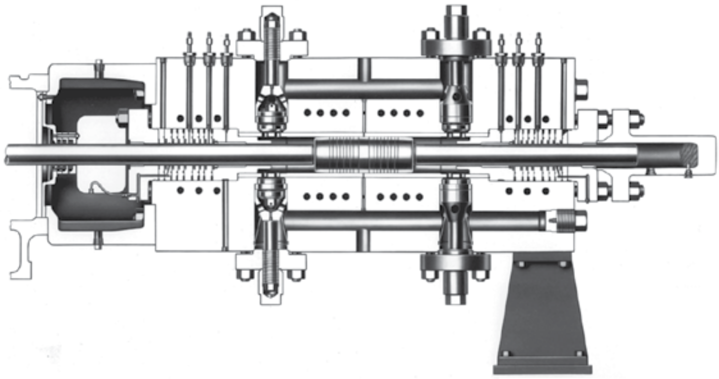

Figure 7.46 High-pressure double-acting tailrod cylinder. The forged steel cylinder above uses a tailrod with forged steel guard for higher pressures.

f. Double-acting Tail Rod

Both cylinder ends compress gas and the piston rod is extended into the outer end of the cylinder, thus providing equal flow to both ends. This arrangement is utilized where a crosshead pin reversal problem might exist or as a means to reduce frame load on a conventional double-acting cylinder arrangement (Fig. 7.46).

g. High-pressure Plunger

High-pressure plunger cylinders are designed for pressures up to 60,000 psig and are primarily used in the petrochemical industry. The forged-steel barrel is made of aircraft-quality forged steel and the plungers are usually nitrided steel or tungsten carbide materials (Fig. 7.47). The plunger reciprocates through a cooled packing; piston rings are not used.

h. Non-lubricated Cylinders

The elimination of hydrocarbon lubricants in compressor cylinders has long been a requirement in many process applications where contamination or possible combustion is not tolerable. The strongest example would be in compressing oxygen. The oil and oxygen would ignite. In the case of oxygen, no hydrocarbon compound can be allowed to come in contact with the gas. In other cases, it may be that lubricating oil will discolor the final product or contaminate the process. With the advent of self-lubricating materials such as TFE for piston rings, piston wear bands, packings, and special valves, the applications of non-lubricated compressors has increased and become widespread.

Compressor cylinder designs may be defined as:

(1) Fully lubricated
(2) Minimum lubricated, or mini-lube
(3) Non-lubricated

(1) A fully lubricated cylinder is supplied with one or more points of lubrication in the bore, where oil is introduced to provide a protective film. This oil is injected under pressure from an auxiliary pump.

(2) A mini-lube cylinder has no injection of oil to the cylinder bore. The only lubrication is that which is carried from the lubricated pressure packing.

(3) Non-lubricated cylinders operate truly without the benefit of any lubricant. They depend solely on non-metallic piston rings, riders, and special valves. Extra distance pieces or compartments may be provided, as required, along with oil slingers to preclude absolutely any possibility of oil traveling along the piston rod into the cylinder. Slightly higher power requirements may be expected with non-lubricated compressors.

i. *Labyrinth-type Cylinders*

A labyrinth-type piston design can be used for non-lubricated, dry gas compressor service. The compressor cylinder or cylinders are located vertically above the crankshaft. The piston uses TFE-filled guide rings and utilizes a captive-ring design principle. The captive ring principle restrains the piston ring within the piston so that the resulting radial force from the pressure difference over the piston ring is no longer able to press the ring against the cylinder wall, thus eliminating the main cause of wear. The clearance between the cylinder wall and the piston ring is approximately 1 to 2 mils, thus reducing gas leakage between the high- and low-pressure side during compression.

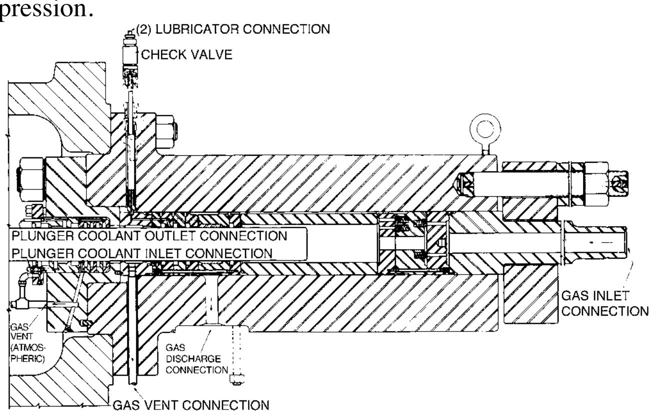

Figure 7.47 High Pressure Single-acting Plunger Cylinder

The piston tail rod runs in a TFE-filled bushing above the piston, which, assisted by the captive rings, ensures an extremely stable piston. (Fig. 7.51).

6. Cylinder Components

a. Valves

Valve designs utilizing moving, flexing, metallic strips or channels are employed on low-pressure, dry, noncorrosive applications. Circular-plate-type valves are most often used for high-pressure and corrosive services.

Figure 7.48 Valves

The selection of valve materials for operation in various gases depends on several factors. The series designation for stainless-steel strips or plates depends on H_2S content and lubrication requirements. Nonmetallic valve plate materials are sometimes used for these difficult services. Valve spring materials include stainless steels, cadmium-plated steel alloys, Inconel, and Hastelloy.

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Valve seats and guards (Fig. 7.48) are gray or nodular iron for low-pressure applications and steel for high-differential-pressure service.

b. *Piston Rod Packing*

Most reciprocating compressors are double acting. To seal the frame end of the cylinder where the piston rod travels through the cylinder head, piston rod packing is used.

Piston rod packing is furnished in a great many materials, depending on the service and application. Typical packing ring materials are non-metallic, such as filled TFE and bronze; packing cases are made of iron, steel or bronze. Typical packing ring configurations include tangential and radial sealing rings; the number of sets of packing rings (Fig. 7.49), is a function of cylinder pressure.

For critical high-pressure service such as hydrogen or non-lubricated applications, the packing rings may he cooled. Water or oil is pumped through the packing case jackets, thus cooling the packing rings to ensure longer packing life.

c. *Piston Design*

The piston design for reciprocating compressors is critical to the operation of the compressor. There are many designs used, depending on application and service. Pistons are made of gray or nodular iron, aluminum, steel, and fabricated steel. One-, two-, and three-piece piston designs (Fig. 7.50) are utilized depending on application criteria.

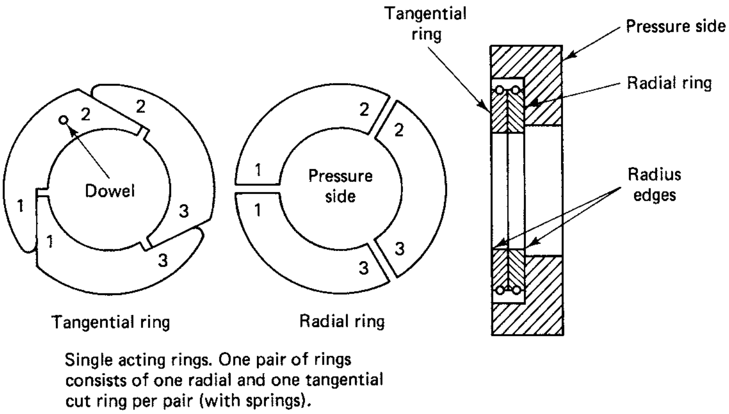


Figure 7.49 Piston Rod Packing

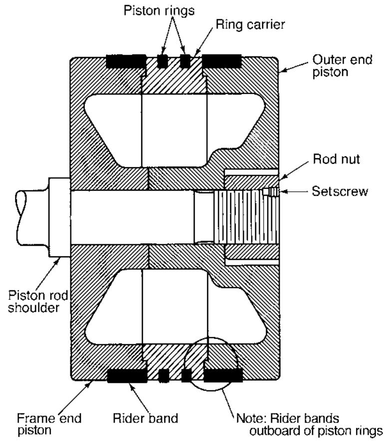


Figure 7.50 Typical Three-piece Piston Design

For applications using large-diameter pistons or critical services where lubrication may be difficult, or for non-lubricated services, a rider or bearing surface is frequently provided to assist in break-in or to assure long life. The rider band is usually a filled Teflon-type material.

d. Piston Rods

A surface hardness of RC 50 minimum is preferred on piston rods in the area that passes through the packing. Special alloys and rod coatings are often applied for sour gas service. Threads may be either cut or rolled, depending on root stress design load criteria.

e. Piston Rings

Many types of piston rings are used, depending upon cylinder pressure, temperature, and bore size. Piston and rider rings are generally made of filled TFE materials, metallic materials, or combinations of these.

f. Distance Pieces

Since reciprocating compressors are used to handle many gases that are flammable or toxic, many types of distance pieces are used to ensure that these gases do not get into the compressor frame and that lubricating oil from the compressor frame is not mixed with cylinder lubrication. In addition, distance pieces are usually provided between the frame and the cylinder to allow maximum accessibility to the piston rod packing and the frame-end wiper packing.

Distance pieces with open-end compartments allow non-hazardous gas to be vented to the atmosphere. If the gas being compressed is hazardous, the distance piece has gas tight covers and is frequently vented or purged with inert gas to prevent contamination of the atmosphere and the crankcase oil by the compressed gas.

Water-cooled or oil-cooled packing is sometimes required in high-pressure service. A variety of distance pieces and configurations is available so that the user may choose exactly the right cylinder-distance system for this application. The following configurations are available (Fig. 7.51):

Type A[1]: Short, close-coupled, single compartment for lubricated service when oil carryover is not objectionable and where it is desired to keep overall width to a minimum for ease of transportation.

Type A: Standard, short, single-compartment, for lubricated service where oil carryover is not objectionable.

Type B: Long, single compartment, for non-lubricated and lubricated service. To prevent crankcase oil from entering the cylinder. No part of the piston rod shall alternately enter the crankcase and the cylinder pressure packing. Normally, an oil slinger is attached to the rod to ensure that oil does not travel along the rod.

Type C: Long, two-compartment distance piece for flammable, hazardous, or toxic gases. No part of the piston rod shall alternately enter the wiper packing, intermediate seal packing, and the cylinder pressure packing. The compartment next to the cylinder can be purged with an inert gas to prevent the hazardous gas from entering the crankcase. Normally, an oil slinger is attached to the rod to ensure oil does not travel along the rod. Slingers may be used in either or both compartments.

Type D: Short, two-compartment distance piece for flammable, hazardous, or toxic gases. No part of the piston rod shall alternately enter the wiper packing and the intermediate seal packing. The compartment next to the cylinder can be purged.

7. Capacity Control

Several different control devices are available to vary compressor flow rate to match system demands.

a. Clearance Pockets

Adding clearance volume to a cylinder end changes the suction valve opening as it is related to the position of the crankshaft. Clearance pockets can be used to reduce compressor flow and horse power or to permit operation at alternate pressure and flow conditions. They are generally added to the head end of the cylinder, as seen in Fig. 7.52.

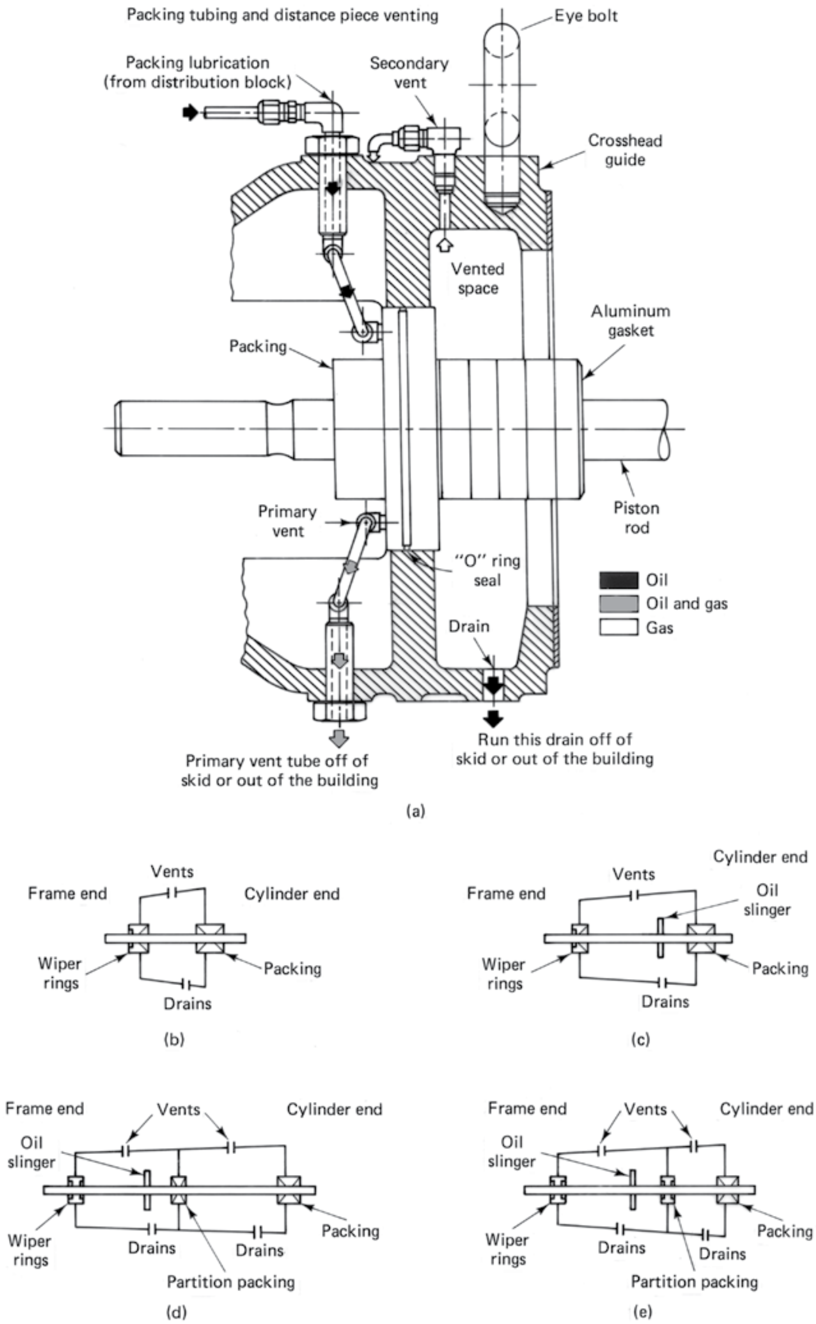


Figure 7.51 Several distance piece configurations.

(1) Fixed-volume clearance pocket: reduces compressor capacity a definite amount or a single step. Manual or pneumatic actuation is available.

(2) Variable-volume clearance pocket: provides capacity reduction in an infinite number of steps over a given reduction range. It is normally for manual actuation only; however, automatic operation may be provided but is generally complicated and expensive.

b. Split-valve Yokes

Clearance can be added to a cylinder by elevating the compressor valve from its cylinder seat. This requires a two-piece (split) valve yoke that permits the valve either to sit on the cylinder seat (minimum clearance) or to be sandwiched between the parts of the two-piece yoke (added clearance). The compressor must be shut down to make these physical changes (Fig. 7.53).

c. Clearance Rings

A clearance ring, equivalent to a thick gasket, can be inserted between the cylinder barrel and head to add clearance volume. These modifications are generally limited to the outboard cylinder head, as inboard modifications affect the piping. The compressor must be shut down to make this physical change.

d. End Unloading

For start-up and capacity control, unloading devices or valve removal can be utilized to vent a cylinder end continually to suction pressure. In general, end unloading is not used for inlet pressure over 2500 psi. If one cylinder end is unloaded, the flow and horsepower are reduced by approximately 50%.

(1) *Depressors* (Fig. 7.54): End unloading can be accomplished by depressing (holding open) suction valve sealing elements that vent the cylinder end to suction pressure. Unloading devices are required over all suction valves and are available for manual or pneumatic actuation.

Figure 7.51 Several distance piece configurations

a. Close coupled single-compartment distance piece.
b. Standard, short single-compartment distance piece. (May be integral with crosshead guide or distance piece).
c. Long, single-compartment distance piece.
d. Long, two-compartment distance piece of sufficient length for oil slinger if required. Partition pecking may be lubricated in each compartment.
e. Short, two-compartment distance piece with partition packing and oil slinger if required. Partition packing may be lubricated in frame-end compartment.

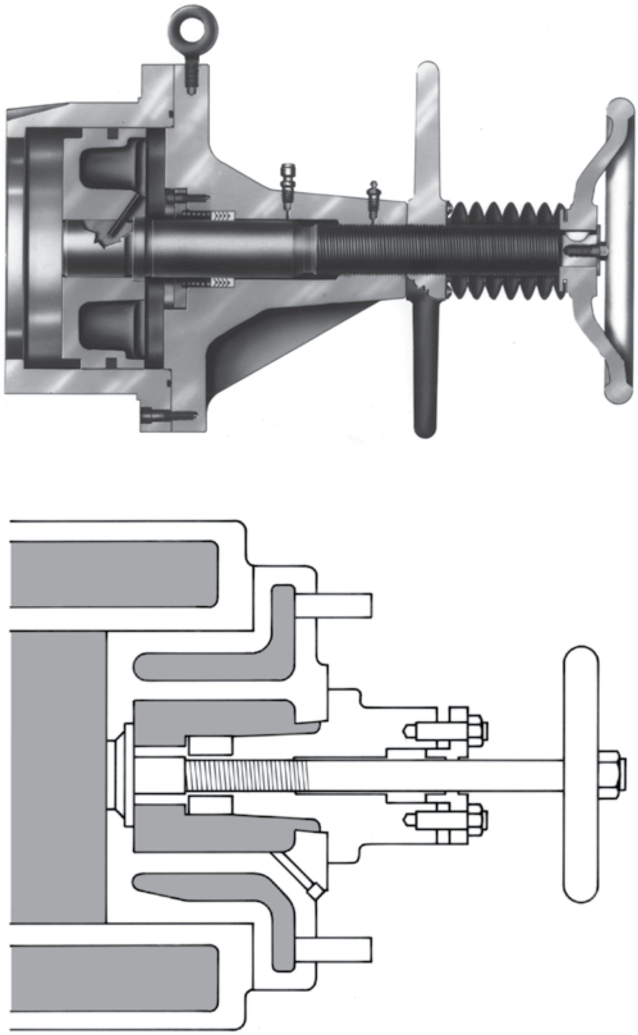


Figure 7.52 Manual fixed volume clearance pocket and manual variable clearance pocket.

(2) *Plug* (Fig. 7.54): A cylinder end may be unloaded by venting directly to the suction gas port through a separate opening either in the cylinder barrel or in the center of a circular plate type suction valve. This device does not interrupt normal intake valve operation and is available for manual or pneumatic actuation.

e. Gas Bypass

Changes in process flow demands can be compensated for by recirculating gas around the compressor. A portion of the discharge gas is redirected (bypassed) through a heat exchanger and then expanded back to suction pressure, where it is mixed with the normal suction gas stream. The other capacity control devices listed previously achieve flow reduction with an almost proportionate reduction in horsepower. With gas bypass systems, the horsepower requirement remains constant because the compressor throughput is not reduced.

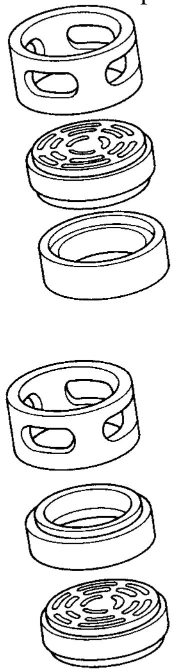
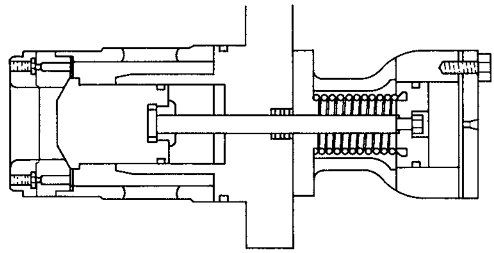


Figure 7.53 Split valve yoke, maximum and minimum clearance.

f. Speed Control

Capacity is almost proportional to the compressor's rotating speed. Special attention to torsional natural frequencies is mandatory when designing variable-speed Systems. Special flywheels, quill shafts, and couplings are often required. Oversized or auxiliary frame oil pumps may be necessary when speed reduction is used.

(1) *Engine drivers:* Speed reductions of 50% are possible.
(2) *Turbine and gear:* Speed reductions of 25% are commonly used.
(3) *Electric motors:* Speed reductions of up to 50% are possible with ac variable-speed controllers and DC. Initial capital costs, stiff electrical system grids, and motor efficiency versus power correction factor limit the practicality of AC systems.



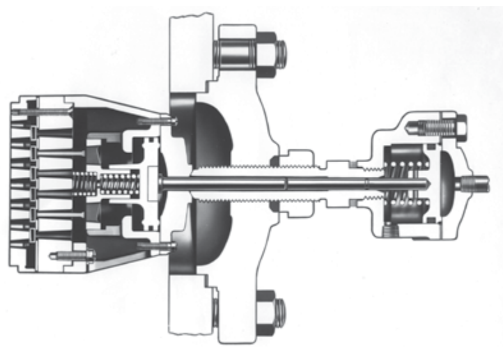


Figure 7.54 Suction valve depressor and plug unloader.

8. Sour Gas

Sour gas can be defined as gas having substantial amounts of hydrogen sulfide (H_2S) (above 1/2 mol percent). Sulfides result in a highly corrosive condition requiring special selection of cylinder materials. The most critical material selection is any part that is made of steel. Any steel in an H_2S atmosphere is susceptible to stress corrosion cracking. Such factors as yield strength, hardness, design stress, and temperature all influence the performance of these parts in H_2S atmosphere. Various experiences and testing have developed these guidelines.

The selection of component material is best done by the compressor manufacturer so that the material will suit the gas being handled and will be compatible with other wearing parts within the cylinder.

- Materials with yield strengths of 60,000 to 90,000 psi are acceptable.
- All test results indicate that hardness levels should not exceed RC22. The softer the material, the less susceptible it is to corrosion cracking and embrittlement.
- Design temperature must be kept low to prevent acceleration of corrosion fatigue.

a. Compressor Cylinder Material Selection for Sour Gas Cylinder Body:

Gray iron and nodular iron cylinders have been successfully used on many sour gas applications with both low and high H_2S concentrations. When steel cylinders are required, castings or forged-billet-type cylinders are applied. Special attention must be given to yield strength, carbon percentage and hardness.

Liner Material: Centrifugally cast gray iron liners have been successfully used with a hardness level of 217 BHN, minimum. A higher hardness level, which exhibits good antiwear properties, is preferred. All these applications are lubricated machines, and the lubricant film on the liner prevents corrosion. In some cases, Ni-resist has been used for anticorrosion reasons.

Piston: Cast iron and aluminum have been successfully used.

Rider Rings: Filled TFE has proved to be the best in this environment. Split or band types have been used successfully.

Piston Rings: Filled TFE gives the most consistent results.

Piston Rods: A number of materials that have been used with various degrees of success include the following:

(1) Stainless steel, No.410, with a core hardness of RC22 to RC28, induction hardened in the packing travel to RC40.
(2) SAE 8620 or AISI 4140 with a maximum core hardness less than RC22, with a chrome-plated or tungsten carbide overlay in the packing travel area.
(3) Carpenter Custom 450 with an RC40 hardness and 17-4 PH with a maximum hardness of RC3 have been used in sour gas applications.

Distance Piece: Must be closed, with proper venting.

Packing Rings: Filled TFE packing rings with metallic backup rings have proved to be most successful.

Packing Cases: Gray and nodular iron have been used up to 1500-psig discharge pressure. In excess of 1500 psig, AISI 4140 material has been used successfully. Packing cases must be cooled at pressures in excess of 1500 psig when filled TFE rings are used.

Valves: The use of steel valve components must comply with the design criteria outlined previously. Gray or nodular steel seats and guards are used for medium differential pressure applications. The use of inert, nonmetallic valve plates (thermoplastic) is well suited for sour gas service.

9. Material Requirements for Special Applications

Table 7.1 gives typical materials for a few special gas compressor applications.

Table 7.1 Typical Compressor Cylinder Material Usage

Compressor Application	Hydrogen	Carbon Dioxide	Hydrogen Sulfide (sour gas)	Oxygen	Low-Temperature Applications below -60°F	Chlorides
Cylinder and heads	Gray iron Nodular iron Steel	Gray iron Nodular iron Steel	Gray iron Nodular iron Steel	Gray iron Nodular iron	Stainless steel Iron alloy	Gray iron Nodular iron
Liners	Gray iron Ni-resist	Iron Ni-resist	Iron Ni-resist	No liner	Gray iron Ni-resist	Gray iron
Piston	Gray iron Aluminum Steel	Gray iron Aluminum Steel	Gray iron Aluminum	Gray iron Stainless steel Monel	Stainless steel Iron alloy	Gray iron Stainless steel
Piston rods	Steel Stainless steel	Steel Stainless steel	Stainless steel	K-monel	Stainless steel	Stainless steel
Piston rod nuts	Steel Stainless steel	Steel Stainless steel	Stainless steel	Everdur	Stainless steel	Stainless steel
Valve seats and guards	Gray iron Nodular iron Steel Stainless steel	Gray iron Nodular iron Stainless steel	Gray iron Nodular iron Stainless steel	Gray iron Nodular iron Stainless steel	Stainless steel Iron alloy	Gray iron Nodular iron
Valve plates	Stainless steel Nonmetallic	Stainless steel Nonmetallic	Stainless steel Nonmetallic	Stainless steel Nonmetallic	Stainless steel	K-monel Nonmetallic
Valve springs	Steel Stainless steel Chrome-Vanadium	Steel Stainless steel Chrome-Vanadium	Inconel Stainless steel Chrome-Vanadium	Stainless steel Chrome-Vanadium	Inconel Chrome-Vanadium	K-monel Chrome-Vanadium
Packing cases	Gray iron Steel Stainless steel	Gray iron Steel Stainless steel	Gray iron Stainless steel	Bronze Stainless steel	Stainless steel	Gray iron Stainless steel
Packing rings	Nonmetallic[a] Metallic (bronze)	Nonmetallic[a] Metallic (bronze)	Nonmetallic[a]	Nonmetallic[a]	Nonmetallic[a]	Nonmetallic[a]
Piston rings	Nonmetallic[a] Metallic (bronze)	Nonmetallic[a] Metallic (bronze)	Nonmetallic[a]	Nonmetallic[a]	Nonmetallic[a]	Nonmetallic[a]
Rider rings	Nonmetallic[a] Metallic (bronze)	Nonmetallic[a] Metallic (bronze)	Nonmetallic[a]	Nonmetallic[a]	Nonmetallic[a]	Nonmetallic[a]
Valve seats	Soft steel	Copper Soft steel	Soft steel	Copper	Steel Steel	Monel
Valve cover gasket	O-ring	O-ring	O-ring	O-ring	O-ring	O-ring
Studs	Steel	Steel	Steel (4140/B714 stainless steel)	Steel	Stainless steel	
Type distance piece	A	A, B	C, D	D	C, D	C, D
or	A	AB	CD	D	C	CD

[a] Typical filler material includes fiberglass, carbon, and bronze; appropriate selection depends on the service.

COMPRESSOR PERFORMANCE

The following section on compressor performance has been reprinted with permission from the *Gas Processors Suppliers Association Engineering Data Book, Eleventh Edition,* with minor editorial changes including the elimination of references to sections that are not included. A few footnotes have been added, as well as a few cross references to other sections of the *Compressed Air and Gas Handbook.* Figure numbers have been changed to the system used in this handbook.

Depending on application, compressors are manufactured as positive-displacement, dynamic, or thermal type (Fig. 7.56). Positive displacement types fall into two basic categories: reciprocating and rotary.

The reciprocating compressor consists of one or more cylinders each with a piston or plunger that moves back and forth, displacing a positive volume with each stroke.

The diaphragm compressor uses a hydraulically pulsed flexible diaphragm to displace the gas.

Rotary compressors cover lobe-type, screw-type, vane-type, and liquid ring type, each having a casing with one or more rotating elements that either mesh with each other, such as lobes or screws, or that displace a fixed volume with each rotation.

The dynamic types include radial-flow (centrifugal), axial-flow and mixed-flow machines. They are rotary continuous-flow compressors in which the rotating element (impeller or bladed rotor) accelerates the gas as it passes through the element, converting the velocity head into static pressure, partially in the rotating element and partially in stationary diffusers or blades.

Ejectors are “thermal” compressors that use a high-velocity gas or stream jet to entrain the inflowing gas, then convert the velocity of the mixture to pressure in a diffuser.

Figure 7.57 covers normal range of operation for compressors of the commercially available types. Figure 7.58 summarizes the difference between reciprocating and centrifugal compressors.

ACFM	=	actual cubic feet per minute (i.e. at process conditions)
A_p	=	cross sectional area of piston, sq in.
A_r	=	cross sectional area of piston rod, sq in.
Bhp	=	brake or shaft horsepower
C	=	cylinder clearance as a per cent of cylinder volume
C_p	=	specific heat at constant pressure, Btu/(lb · °F)
C_v	=	specific heat at constant volume, Btu/(lb · °F)
D	=	cylinder inside diameter, in.
d	=	piston rod diameter, in.
F	=	an allowance for interstage pressure drop, eq. 7.4
Ghp	=	gas horsepower, actual compression horsepower, excluding mechanical losses, bhp
H	=	head, ft · lb/lb
h	=	enthalpy, Btu/lb
ICFM	=	inlet cubic feet per minute, usually at suction conditions
k	=	isentropic exponent, C_p/C_v
MC_p	=	molal specific heat at constant pressure, Btu/(lb mol · °F)
MC_v	=	molar specific heat at constant volume, Btu/(lb mol · °F)
MW	=	molecular weight
mm cfd	=	million cubic ft/day
N	=	speed, rpm
N_m	=	molal flow, mols/min
n	=	polytropic exponent or number of mols
P	=	pressure, psia
P_c	=	critical pressure, psia
PD	=	piston displacement, ft^3/min
P_L	=	pressure base used in the contract or regulation, psia
pP_c	=	pseudo critical pressure, psia
P_R	=	reduced pressure, P/P_c
pT_c	=	pseudo critical temperature, °R
P_V	=	partial pressure of contained moisture, psia
p	=	pressure, lb/ft^2
Q	=	inlet capacity (ICFM)
R	=	universal gas constant
	=	$10.73 \dfrac{psia \cdot ft^3}{lb\,mol \cdot °R}$
	=	$1545 \dfrac{(lb/ft^2) \cdot ft^3}{lb\,mol \cdot °R}$ or $\dfrac{ft \cdot lb}{lb\,mol \cdot °R}$
	=	$1.986 \dfrac{Btu}{lb\,mol \cdot °R}$
r	=	compression ratio, P_2/P_1
s	=	entropy, Btu/(lb · °F) or number of wheels
SCFM	=	standard cubic feet per minute measured at 14.7 psia at 60 °R
stroke	=	length of piston movement, in.
T	=	absolute temperature, °R
T_c	=	critical temperature, °R
T_R	=	reduced temperature, T/T_c
t	=	temperature, °F
V	=	specific volume, ft^3/lb
VE	=	volumetric efficiency, per cent
W	=	work, ft · lb
w	=	weight flow, lb/min
X	=	temperature rise factor
y	=	mol fraction
Z	=	compressibility factor*
Z_{avg}	=	average compressibility factor = $\dfrac{Z_s + Z_d}{2}$ *
η	=	efficiency, expressed as a decimal
Subscripts		
d	=	discharge
is	=	isentropic process
p	=	polytropic process
S	=	standard conditions, usually 14.7 psia, 60°F
s	=	suction
t	=	total or overall
1	=	inlet conditions
2	=	outlet conditions
L	=	standard conditions used for calculation or contract

*To avoid possible confusion, many authorities refer to Z as supercompressibility, reserving compressibility to mean departures from liquid, or incompressible behavior. For further discussion of this topic see Chapter 8 under the heading *Superconductivity.*

Figure 7.55 Nomenclature

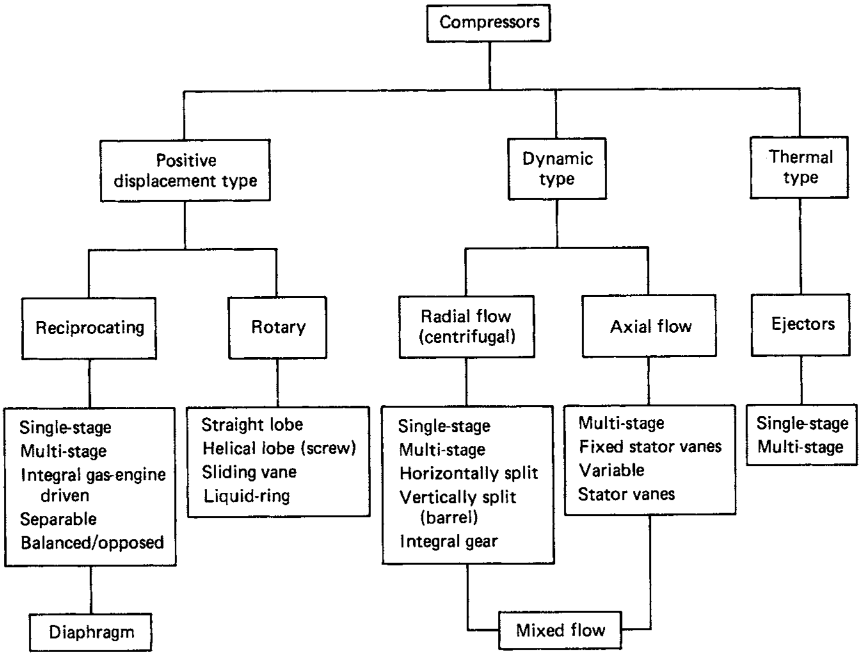


Figure 7.56 Types of Compressors

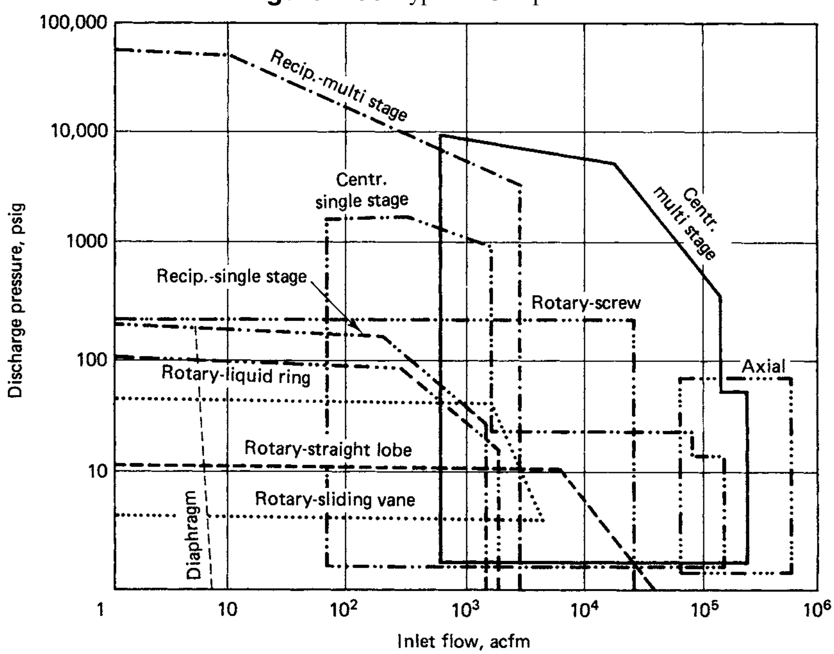


Figure 7.57 Compressor Coverage Chart

The advantages of a centrifugal compressor over a reciprocating machine are:

a. Lower installed first cost where pressure and volume conditions are favorable.
b. Lower maintenance expense.
c. Greater continuity of service and dependability.
d. Less operating attention required.
e. Greater volume capacity per unit of plot area.
f. Adaptability to high-speed low-maintenance-cost drivers.

The advantages of a reciprocating compressor over a centrifugal machine are:

a. Greater flexibility in capacity and pressure range.
b. Higher compressor efficiency and lower power cost.
c. Capability of delivering higher pressures.
d. Capability of handling smaller volumes.
e. Less sensitive to changes in gas composition and density.

Figure 7.58 Comparison of reciprocating and centrifugal compressors.

RECIPROCATING COMPRESSORS

Reciprocating compressor ratings vary from fractional to more than 20,000 hp per unit. Pressures range from low vacuum at suction to 30,000 psi and higher at discharge for special process compressors.

Reciprocating compressors are furnished either single-stage or multi-stage. The number of stages is determined by the overall compression ratio. The compression ratio per stage (and valve life) is generally limited by the discharge temperature and usually does not exceed 4, although small-sized units (intermittent duty) are furnished with a compression ratio as high as 8.

Gas cylinders are generally lubricated, although a non-lubricated design is available when warranted, for example, for nitrogen, oxygen, or instrument air.

On multi-stage machines, intercoolers may be provided between stages. These are heat exchangers which remove the heat of compression from the gas and reduce its temperature to approximately the temperature existing at the compressor intake. Such cooling reduces the actual volume of gas going to the high-pressure cylinders, reduces the horsepower required for compression, and keeps the temperature within safe operating limits.

Reciprocating compressors should be supplied with clean gas as they cannot satisfactorily handle liquids and solid particles that may be entrained in the gas. Liquids and solid particles tend to destroy cylinder lubrication and cause excessive wear. Liquids are noncompressible and their presence could rupture the compressor cylinder or cause other major damage.

Performance Calculations

The engineer in the field is frequently required to:

1. Determine the approximate horsepower required to compress a certain volume of gas at some intake conditions to a given discharge pressure.
2. Estimate the capacity of an existing compressor under specified suction and discharge conditions.

The following text outlines procedures for making these calculations from the standpoint of quick estimates and also presents more detailed calculations. For specific information on a given compressor consult the manufacturer of that unit.

For a compression process the enthalpy change is the best way of evaluating the work of compression. If a P-h diagram is available (as for propane refrigeration systems), the work of compression would always be evaluated by the enthalpy change of the gas in going from suction to discharge conditions. Years ago the capability of easily generating P-h diagrams for natural gases did not exist. The result was that many ways of estimating the enthalpy change were developed. They were used as a crutch and not because they were the best way to evaluate compression horsepower requirements.

Today the engineer does have available, in many cases, the capability to generate that part of the P-h diagram he requires for compression purposes. This is done using equations of state on a computer. This still would be the best way to evaluate the compression horsepower. The other equations are used only if access to a good equation of state is not available.

Some practical references continue to treat reciprocating and centrifugal machines as being different so far as estimation of horsepower requirements is concerned. This treatment reflects industry practice. The only difference in the horsepower evaluation is the efficiency of the machine. Otherwise, the basic thermodynamic equations are the same for all compression.

The reciprocating compressor horsepower calculations presented are based on charts. However, they may equally well be calculated using the equations in the centrifugal compressor section, particularly Eqs. (7.25) through (7.43). This also includes the mechanical losses in Eqs. (7.37) and (7.67).

There are two ways in which the thermodynamic calculations for compression can be carried out by assuming:

1. Isentropic reversible path: a process during which there is no heat added to or removed from the system and the entropy remains constant, pv^k = constant.
2. Polytropic path: a process in which heat transfer or changes in gas characteristics during compression are considered pv^n = constant.

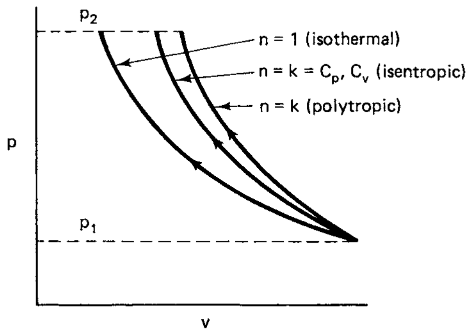


Figure 7.59 Compression curves.

Figure 7.59 shows a plot of pressure versus volume for each value of exponent n. The work, W performed in a flow process from p_1 to p_2 along any polytropic curve (Fig. 7.62) is:

$$W = \int_1^2 V \cdot dp = \int_{p1}^{p2} V \cdot dp \qquad (7.1)$$

The amount of work required is dependent upon the polytropic curve involved and increases with increasing values of n. The path requiring the least amount of input work is n = 1, which is equivalent to isothermal compression, a process during which there is no change in temperature. For isentropic compression n = k = ratio of specific heat at constant pressure to that at constant volume.

It is usually impractical to build sufficient heat-transfer equipment into the design of most compressors to carry away the bulk of the heat of compression. Most machines tend to operate along a polytropic path which approaches the isentropic. Most compressor calculations are therefore based on an efficiency applied to account for true behavior.

A compression process following the middle curve in Fig. 7.59 has been widely referred to in industry as "adiabatic." However, many compression processes of practical importance are approximately adiabatic. The term adiabatic does not adequately describe this process, since it only implies no heat transfer. The ideal process also follows a path of constant entropy and should be called "isentropic," as will be done subsequently in this chapter.

Equation (7.3), which applies to all ideal gases, can be used to calculate k.

$$MC_p - MC_v = R = 1.986 \text{ Btu/(lb mol} \cdot \text{°F)} \qquad (7.2)$$

By rearrangement and substitution, we obtain

$$k = \frac{C_p}{C_v} = \frac{MC_p}{MC_v} = \frac{MC_p}{MC_p - 1.986} \qquad (7.3)$$

To calculate k for a gas we need only know the constant pressure molal heat capacity (MC_p) for the gas. Figure 7.60 gives values of molecular weight and ideal-gas heat capacity (i.e., at 1 atm) for various gases. The heat capacity varies considerably with temperature.

Gas	Chemical formula	Mol wt	Temperature 0°F	50°F	60°F	100°F	150°F	200°F	250°F	300°F
Methane	CH_4	16.043	8.23	8.42	8.46	8.65	8.95	9.28	9.64	10.01
Ethyne (Acetylene)	C_2H_2	26.038	9.68	10.22	10.33	10.71	11.15	11.55	11.90	12.22
Ethene (Ethylene)	C_2H_4	28.054	9.33	10.02	10.16	10.72	11.41	12.09	12.76	13.41
Ethane	C_2H_6	30.070	11.44	12.17	12.32	12.95	13.78	14.63	15.49	16.34
Propane (Propylene)	C_3H_6	42.081	13.63	14.69	14.90	15.75	16.80	17.85	18.88	19.89
Propane	C_3H_8	44.097	15.65	16.88	17.13	18.17	19.52	20.89	22.25	23.56
1-Butene (Butylene)	C_4H_8	56.108	17.96	19.59	19.91	21.18	22.74	24.26	25.73	27.16
cis-2-Butene	C_4H_8	56.108	16.54	18.04	18.34	19.54	21.04	22.53	24.01	25.47
trans-2-Butene	C_4H_8	56.108	18.84	20.23	20.50	21.61	23.00	24.37	25.73	27.07
iso-Butane	C_4H_{10}	58.123	20.40	22.15	22.51	23.95	25.77	27.59	29.39	31.11
n-Butane	C_4H_{10}	58.123	20.80	22.38	22.72	24.08	25.81	27.55	29.23	30.90
iso-Pentane	C_5H_{12}	72.150	24.94	27.17	27.61	29.42	31.66	33.87	36.03	38.14
n-Pentane	C_5H_{12}	72.150	25.64	27.61	28.02	29.71	31.86	33.99	36.08	38.13
Benzene	C_6H_6	78.114	16.41	18.41	18.78	20.46	22.45	24.46	26.34	28.15
n-Hexane	C_6H_{14}	86.177	30.17	32.78	33.30	35.37	37.93	40.45	42.94	45.36
n-Heptane	C_7H_{16}	100.204	34.96	38.00	38.61	41.01	44.00	46.94	49.81	52.61
Ammonia	NH_3	17.0305	8.52	8.52	8.52	8.52	8.52	8.53	8.53	8.53
Air		28.9625	6.94	6.95	6.95	6.96	6.97	6.99	7.01	7.03
Water	H_2O	18.0153	7.98	8.00	8.01	8.03	8.07	8.12	8.17	8.23
Oxygen	O_2	31.9988	6.97	6.99	7.00	7.03	7.07	7.12	7.17	7.23
Nitrogen	N_2	28.0134	6.95	6.95	6.95	6.96	6.96	6.97	6.98	7.00
Hydrogen	H_2	2.0159	6.78	6.86	6.87	6.91	6.94	6.95	6.97	6.98
Hydrogen sulfide	H_2S	34.08	8.00	8.09	8.11	8.18	8.27	8.36	8.46	8.55
Carbon monoxide	CO	28.010	6.95	6.96	6.96	6.96	6.97	6.99	7.01	7.03
Carbon dioxide	CO_2	44.010	8.38	8.70	8.76	9.00	9.29	9.56	9.81	10.05

Exceptions: Air, Keenan and Keyes, Thermodynamic Properties of Air, Wiley, 3rd Printing 1947. Ammonia, Edw. R. Grabl, Thermodynamic Properties of Ammonia at High Temperatures and Pressures, Petr. Processing, April 1953. Hydrogen Sulfide, J. R. West, Chem. Eng. Progress, 44, 287, 1948.

Figure 7.60 Molal Heat Capacity MC_p (ideal-gas state), Btu (lb mol · °R) (1) Data source selected values of properties of Hydrocarbons, API Research Project 44.

Since the temperature of the gas increases as it passes from suction to discharge in the compressor, k is normally determined at the average of suction and discharge temperatures.

For a multicomponent gas the mol weighted average value of molal heat capacity must be determined at average cylinder temperature. A sample calculation is shown in Fig. 7.61.

The calculation of pP_p and pT_c in Fig. 7.61 permits calculation of the reduced pressure $P_R = P/pP_c$ mix and reduced temperature $T_R = T/pT_c$ mix. The compressibility* Z at T and P can then be determined using the charts in Chapter 8.

If only the molecular weight of the gas is known and not its composition, an approximate value for k can be determined from the curves in Fig. 7.62.

Estimating Compressor Horsepower

Equation (7.4) is useful for obtaining a quick and reasonable estimate for compressor horsepower. It was developed for large slow-speed (300 to 450 rpm) compressors handling gases with a specific gravity of 0.65 and having stage compression ratios above 2.5.

Caution: Compressor manufacturers generally rate their machines based on a standard condition of 14.4 psia rather than the more common gas industry value of 14.7 psia.

Due to higher valve losses the horsepower requirement for high-speed compressors (1000 rpm range, and some up to 1800 rpm) can be as much as 20% higher, although this is a very arbitrary value. Some compressor designs do not merit a higher horsepower allowance and the manufacturers should be consulted for specific applications.

$$\text{Brake horsepower} = (22)\,(\text{ratio/stage})(\text{no. of stages})(\text{mm cfd})(F) \quad (7.4)$$

where mm cfd compressor capacity referred to 14.4 psia and intake temperature

F = 1.0 for single-stage compression
1.08 for two-stage compression
1.10 for three-stage compression

Equation (7.4) will also provide a rough estimate of horsepower for lower compression ratios and/or gases with a higher specific gravity, but it will tend to be on the high side. To allow for the tendency use a multiplication factor of 20 instead of 22 for gases with a specific gravity in the 0.8 to 1.0 range; likewise, use a factor in the range of 16 to 18 for compression ratios between 1 and 2.0.

*[To avoid possible confusion, many authorities refer to Z as *supercompressibility*, reserving *compressibility* to mean departures from liquid, or incompressible behavior. For a further discussion of this topic, see Chapter 8 under the heading *Supercompressibility*.]

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Example gas mixture		Determination of mixture mol weight		Determination of MC_p, Molal heat capacity		Determination of pseudo critical pressure, pP_c, and temperature, pT_c			
Component name	Mol fraction y	Individual Component Mol weight MW	y · MW	Individual Component MC_p @ 150°F*	y · MC_p @ 150°F	Component critical pressure P_c psia	y · P_c	Component critical temperature T_c°R	y · T_c
methane	0.9216	16.04	14.782	8.95	8.248	666	615.6	343	316.1
ethane	0.0488	30.07	1.467	13.78	0.672	707	34.6	550	26.8
propane	0.0185	44.10	0.816	19.52	0.361	616	11.4	666	12.3
i-butane	0.0039	58.12	0.227	25.77	0.101	528	2.1	734	2.9
n-butane	0.0055	58.12	0.320	25.81	0.142	551	3.0	765	4.2
i-pentane	0.0017	72.15	0.123	31.66	0.054	490	0.8	829	1.4
Total =	1.0000	MW =	17.735	MC_p =	9.578	pP_c =	667.5	pT_c =	363.7
$MC_v = MC_p - 1.986 = 7.592$ $k = MC_p/MC_v = 9.578/7.592 = 1.26$									
* For values of MC_p other than @ 150°F, refer to Fig. 7.60									

Figure 7.61 Calculation of k.

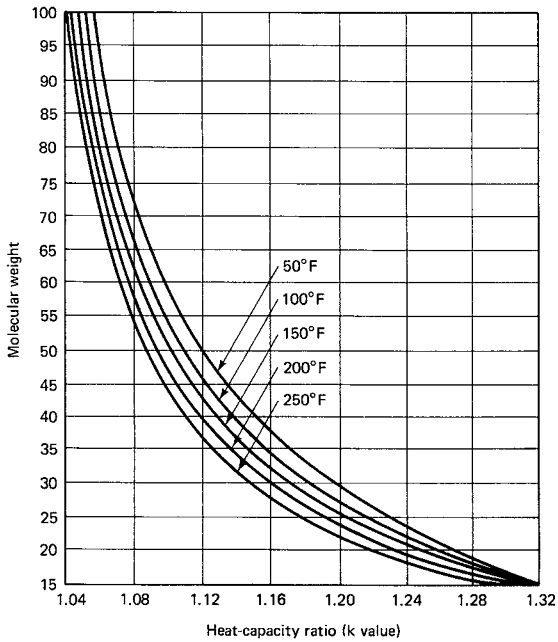


Figure 7.62 Approximate heat-capacity ratios of hydrocarbon gases.

Curves are available that permit easy estimation of approximate compression-horsepower requirements. Figure 7.63 is typical of these curves.

Example 7-1

Compress 2 mm cfd of gas at 14.4 psia and intake temperature through a compression ratio of 9 in a two-stage compressor. What will be the horsepower?

Solution steps

$$\text{Ratio per stage} = \sqrt{9} = 3$$

From Eq (7.4) we find the brake horsepower to be

$$(22)\ (3)\ (2)\ (2)\ (1.08) = 285 \text{ Bhp}$$

From Fig. 7.63, using a *k* of 1.15, we find the horsepower requirement to be 136 Bhp/mm cfd or 272 Bhp. For a *k* of 1.4, the horsepower requirement would be 147 Bhp/mm cfd or 294 total horsepower.

The two procedures give reasonable agreement, particularly considering the simplifying assumptions necessary in reducing compressor horsepower calculations to such a simple procedure.

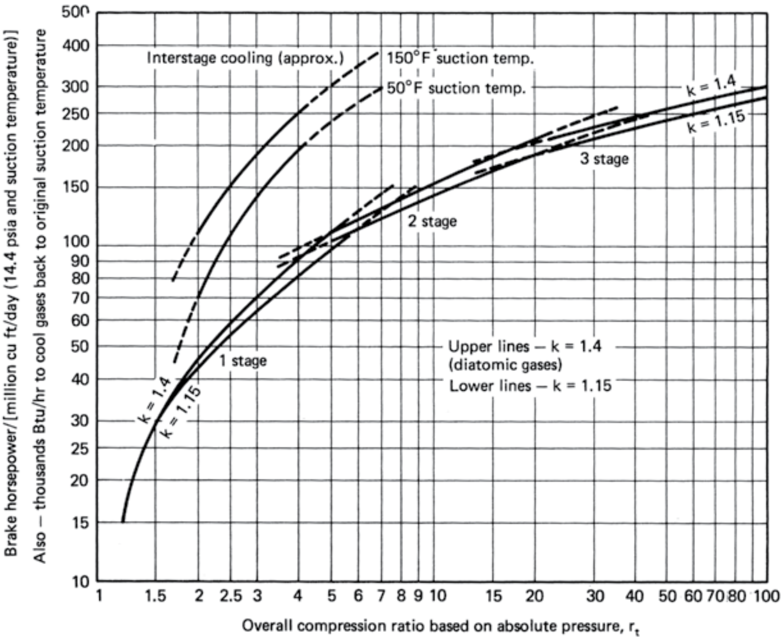


Figure 7.63 Approximate horsepower required to compress gases.

Detailed Calculations

There are many variables which enter into the precise calculation of compressor performance. Generalized data as given in this section are based upon the averaging of many criteria. The results obtained from these calculations, therefore, must be considered as close approximations to true compressor performance.

Capacity

Most gases encountered in industrial compression do not exactly follow the ideal gas equation of state but differ in varying degrees. The degree in which any gas varies from the ideal is expressed by a supercompressibility factor, Z, which modifies the ideal gas equation:

$$PV = nRT$$

to

$$PV = nZRT$$

Supercompressibility factors can be determined from charts in Chapter 8 using the pP_R and pT_R of the gas mixture. For pure components such as propane, compressibility factors can be determined from the P-H diagrams, although the user would be better advised to determine the compression horsepower directly using the P-H diagram.

For the purpose of performance calculations, compressor capacity is expressed as the actual volumetric quantity of gas at the inlet to each stage of compression on a per minute basis (ICFM).

From scfm,

$$Q = \text{scfm}\left(\frac{14.7}{520}\right)\left(\frac{T_1 Z_1}{P_1 Z_L}\right) \tag{7.7}$$

From weight flow (w, lb/min)

$$Q = \frac{10.73}{\text{MW}}\left(\frac{wT_1 Z_1}{P_1 Z_L}\right) \tag{7.8}$$

From weight flow (w, lb/min)

$$Q = \left(\frac{379.5 \cdot 14.7}{520}\right)\left(\frac{N_m T_1 Z_1}{P_1 Z_L}\right) \tag{7.9}$$

From these equations inlet volume to any stage may be calculated by using the inlet pressure P_1 and temperature T_1. Moisture, when in the form of a gas, should be handled just as any other component in the gas.

In a reciprocating compressor, effective capacity may be calculated as the piston displacement (generally in cfm) multiplied by the volumetric efficiency.

The piston displacement is equal to the net piston area multiplied by the length of piston sweep in a given period of time. This displacement may be expressed as follows:

For a single-acting piston compressing on the outer end only,

$$\text{PD} = \frac{(\text{stroke})(N)(D^2)\pi}{(4)(1{,}728)} \qquad (7.10)$$
$$= 4.55\ (10^{-4})(\text{stroke})(N)(D^2)$$

For a single-acting piston compressing on the crank end only,

$$\text{PD} = \frac{(\text{stroke})(N)(D^2 - d^2)\pi}{(4)(1{,}728)} \qquad (7.11)$$
$$= 4.55\ (10^{-4})(\text{stroke})(N)(D^2 - d^2)$$

For a double-acting piston (other than tail rod type),

$$\text{PD} = \frac{(\text{stroke})(N)(2D^2 - d^2)\pi}{(4)(1{,}728)} \qquad (7.12)$$
$$= 4.55\ (10^{-4})(\text{stroke})(N)(2D^2 - d^2)$$

Volumetric Efficiency

In a reciprocating compressor the piston does not travel completely to the end of the cylinder at the end of the discharge stroke. Some clearance volume is necessary and it includes the space between the end of the piston and the cylinder head when the piston is at the end of its stroke. It also includes the volume in the valve ports, the volume in the suction valve guards and the volume around the discharge valve seats.

Clearance volume is usually expressed as a percentage of piston displacement and referred to as percent clearance, or cylinder clearance, C.

$$C = \frac{\text{clearance volume, in3}}{\text{piston displacement, in3}}(100) \qquad (7.13)$$

For double-acting cylinders, the percent clearance is based on the total clearance volume for both the head end and the crank end of a cylinder. These two clearance volumes are not the same due to the presence of the piston rod in the crank end of the cylinder. Sometimes additional clearance volume (external) is intentionally added to reduce cylinder capacity.

The term volumetric efficiency refers to the actual volume capacity of a cylinder compared to the piston displacement. Without a clearance volume for the gas to expand and delay the opening of the suction valve(s), the cylinder could deliver its entire piston displacement as gas capacity. The effect of the gas contained in the clearance volume on the pumping capacity or a cylinder can be represented by

$$VE = 100 - r - C\left[\frac{Z_s}{Z_d}(r^{l/k}) - 1\right] \qquad (7.13)$$

where:

r	=	P_d/P_s (pure ratio, not expressed as a percentage)
P_d	=	cylinder discharge pressure, psia
P_s	=	cylinder suction pressure
Z_d	=	supercompressibility factor at discharge conditions
Z_s	=	supercompressibility factor at suction conditions

Volumetric efficiencies as determined by Eq. (7.14) are theoretical in that they do not account for suction and discharge valve losses. The suction and discharge valves are actually spring-loaded check valves that permit flow in one direction only. The springs require a small differential pressure to open. For this reason, the pressure within the cylinder at the end of the suction stroke is lower than the line suction pressure and, likewise, the pressure at the end of the discharge stroke is higher than line discharge pressure.

One method for accounting for suction and discharge valve losses is to reduce the volumetric efficiency by an arbitrary amount, typically 4 percent, thus modifying Eq. (7.14) as follows:

$$VE = 96 - r - C\left[\frac{Z_s}{Z_d}(r^{l/k}) - 1\right] \qquad (7.15)$$

When a non-lubricated compressor is used, the volumetric efficiency should be corrected by subtracting an additional 5 percent for slippage of gas. This is a capacity correction only and, as a first approximation, would not be considered when calculating compressor horsepower. The energy of compression is used by the gas even though the gas Slips by the rings and is not discharged from the cylinder.

If the compressor is in propane, or similar heavy gas service, an additional 4 percent should be subtracted from the volumetric efficiency. These deductions for non-lubricated and propane performance are both approximate and, if both apply, cumulative.

Figure 7.64 provides the solution to the function $r^{l/k}$. Values for compression ratios not shown may be obtained by interpolation. The closest *k* value column may be safely used without a second interpolation.

Volumetric efficiencies for high-speed separable compressors in the past have tended to be slightly lower than estimated from Eq. (7.14). Recent information suggests that this modification is not necessary for all models of high-speed compressors.

In evaluating efficiency, horsepower, volumetric efficiency, and so on, the user should consider past experience with different speeds and models. Larger valve area for a given swept volume will generally lead to higher compression efficiencies.

Equivalent Capacity

The net capacity for a compressor, in cubic feet per day at 14.4 psia and suction temperature, may be calculated by Eq. (7.16a), which is shown in dimensioned form:

$$\text{mm cfd} = \frac{PD\,\dfrac{\text{ft}^3}{\text{min}}\cdot 1440\,\dfrac{\text{min}}{d}\cdot\dfrac{VE\%}{100}\cdot P_s\,\dfrac{\text{lb}}{\text{in.}^2}\cdot 10^{-6}\,\dfrac{\text{mmft}^3}{\text{ft}^3}\cdot Z_{14.4}}{14.4\,\dfrac{\text{lb}}{\text{in.}^2}\cdot Z_s} \qquad (7.16\text{a})$$

which can be simplified to Eq. (7.16b) when $Z_{14.4}$ is assumed to equal 1.0.

$$\text{mm cfd} = \frac{PD\cdot VE\cdot P_s\cdot 10^{-6}}{Z_s} \qquad (7.16\text{b})$$

For example, a compressor with 200 cfm piston displacement, a volumetric efficiency of 80%, a suction pressure of 75 psia, and suction compressibility of 0.9 would have a capacity of 1.33 mm cfd at 14.4 psia and suction temperature. If compressibility is not used as a divisor in calculating cfm, then the statement "not corrected for compressibility" should be added.

Compression Ratio	k, isentropic exponent C_p/C_v 1.10	1.14	1.18	1.22	1.26	1.30	1.34	1.38	1.42
1.2	1.180	1.173	1.167	1.161	1.156	1.151	1.146	1.141	1.137
1.4	1.358	1.343	1.330	1.318	1.306	1.295	1.285	1.276	1.267
1.6	1.533	1.510	1.489	1.470	1.452	1.436	1.420	1.406	1.392
1.8	1.706	1.675	1.646	1.619	1.594	1.572	1.551	1.531	1.513
2.0	1.878	1.837	1.799	1.765	1.733	1.704	1.677	1.652	1.629
2.2	2.048	1.997	1.951	1.908	1.870	1.834	1.801	1.771	1.742
2.4	2.216	2.155	2.100	2.050	2.003	1.961	1.922	1.886	1.852
2.6	2.384	2.312	2.247	2.188	2.135	2.086	2.040	1.999	1.960
2.8	2.550	2.467	2.393	2.326	2.264	2.208	2.156	2.109	2.065
3.0	2.715	2.621	2.537	2.461	2.391	2.328	2.270	2.217	2.168
3.2	2.879	2.774	2.680	2.595	2.517	2.447	2.382	2.323	2.269
3.4	3.042	2.926	2.821	2.727	2.641	2.563	2.492	2.427	2.367
3.6	3.204	3.076	2.961	2.857	2.764	2.679	2.601	2.530	2.465
3.8	3.366	3.225	3.100	2.987	2.885	2.792	2.708	2.631	2.560
4.0	3.526	3.374	3.238	3.115	3.005	2.905	2.814	2.731	2.655
4.2	3.686	3.521	3.374	3.242	3.124	3.016	2.918	2.829	2.747
4.4	3.846	3.668	3.510	3.368	3.241	3.126	3.021	2.926	2.839
4.6	4.004	3.814	3.645	3.493	3.357	3.235	3.123	3.022	2.929
4.8	4.162	3.959	3.779	3.617	3.473	3.342	3.224	3.116	3.018
5.0	4.319	4.103	3.912	3.740	3.587	3.449	3.324	3.210	3.106
5.2	4.476	4.427	4.044	3.863	3.700	3.554	3.422	3.303	3.193
5.4	4.632	4.390	4.175	3.984	3.813	3.659	3.520	3.394	3.279
5.6	4.788	4.532	4.306	4.105	3.925	3.763	3.617	3.485	3.364
5.8	4.943	4.674	4.436	4.224	4.035	3.866	3.713	3.574	3.448
6.0	5.098	4.815	4.565	4.343	4.146	3.968	3.808	3.663	3.532
6.2	5.252	4.955	4.694	4.462	4.255	4.069	3.902	3.751	3.614
6.4	5.406	5.095	4.822	4.579	4.363	4.170	3.996	3.839	3.696
6.6	5.560	5.235	4.949	4.696	4.471	4.270	4.089	3.925	3.777
6.8	5.713	5.374	5.076	4.813	4.578	4.369	4.181	4.011	3.857
7.0	5.865	5.512	5.202	4.928	4.685	4.468	4.272	4.096	3.937

Figure 7.64 Values of $r^{1/k}$.

In many instances the gas sales contract or regulation will specify some other measurement standard for gas volume. To convert volumes calculated using Eq. (7.16) (i.e., at 14.4 psia and suction temperature) to a P_L and T_L basis, Eq. (7.17) would be used:

$$\text{mm scfd at } P_L, T_L = (\text{mm cfd from Eq. 7.16}) \left(\frac{14.4}{P_L}\right)\left(\frac{T_L}{T_s}\right)\left(\frac{Z_L}{Z_s}\right) \quad (7.17)$$

Discharge Temperature

The temperature of the gas discharged from the cylinder can be estimated from Eq. (7.18), which is commonly used but not recommended. (*Note:* The temperatures are in absolute units, °R or K.) Equations (7.31) and (7.32) give better results.

$$T_d = T_s\left(r^{(k-1)/k}\right) \quad (7.18)$$

Figure 7.65 is a nomograph that can be used to solve Eq. (7.18). The discharge temperature determined from either Eq. (7.18) or Fig. 7.65 is the theoretical value. While it neglects heat from friction, irreversibility effects, etc., and may be somewhat low, the values obtained from this equation will be reasonable field estimates.

Rod Loading

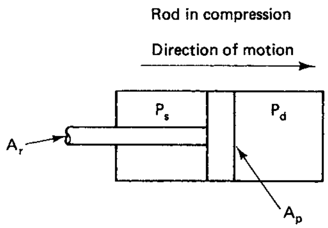
Each compressor frame has definite limitations as to maximum speed and load-carrying capacity. The load-carrying capacity of a compressor frame involves two primary considerations: horsepower and rod loading.

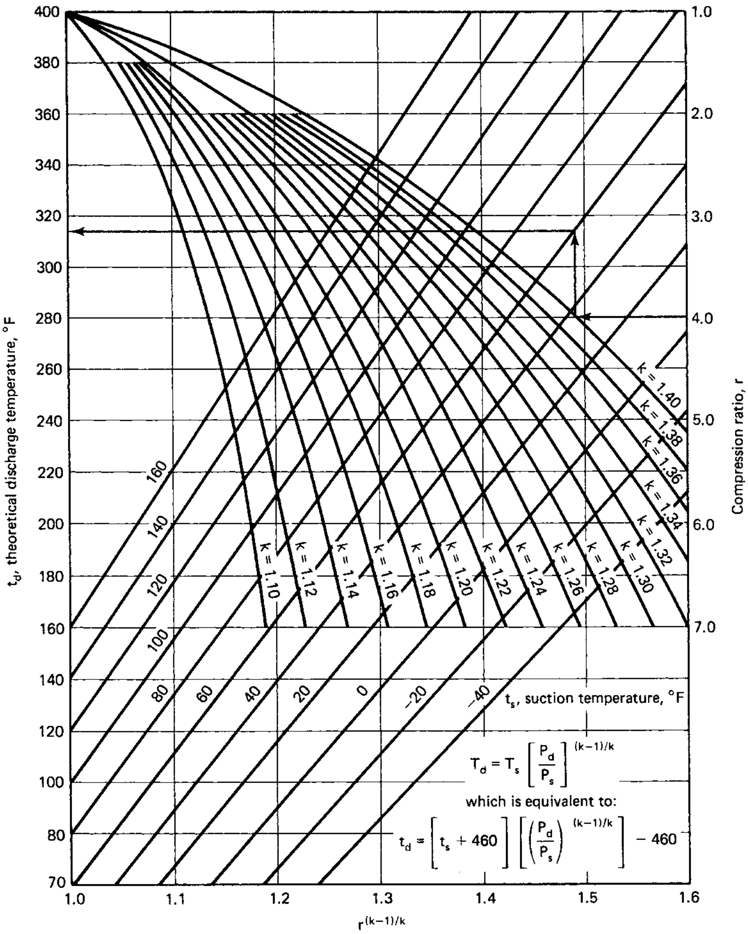
The horsepower rating of a compressor frame is the measure of the ability of the supporting structure and crankshaft to withstand torque (turning moment) and the ability of the bearings to dissipate frictional heat. Rod loads are established to limit the static and inertial loads on the crankshaft, connecting rod, frame, piston rod, bolting, and projected bearing surfaces.

Good design dictates a reversal of rod loading during each stroke. Nonreversal of the loading results in failure to allow bearing surfaces to part and permit entrance of sufficient lubricant. The result will be premature hearing wear or failure.

Rod loadings may be calculated by the use of Eqs. (7.19) and (7.20).

Load in compression = $P_d A_p - P_s(A_p - A_r) = (P_d - P_s)A_p + P_s A_r \quad (7.19)$

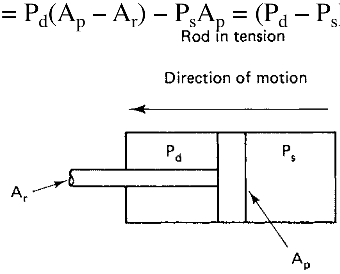




Note: Pressure drop across inlet and discharge valves is assumed to be nil. Allowance should be made for a higher-than-indicated compression ratio if this is not the case.

Figure 7.65 Theoretical discharge temperatures, single-state compression. Read r to k to t_s to t_d.

$$\text{Load in tension} = P_d(A_p - A_r) - P_sA_p = (P_d - P_s)\,A_p - P_dA_r \qquad (7.20)$$



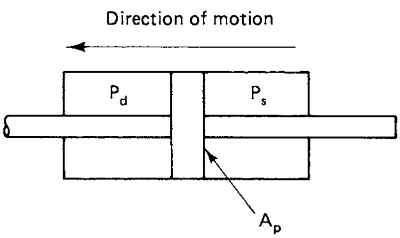
Using Eqs. (7.19) and (7.20), a plus value for the load in both compression and tension indicates a reversal of loads based on gas pressure only. Inertial effects will tend to increase the degree of reversal.

The true rod loads would be those calculated using internal cylinder pressures after allowance for valve losses. Normally the operator will know only line pressures and because of this manufacturers generally rate their compressors based on line-pressure calculations.

A further refinement in the rod-loading calculation would be to include inertial forces. While the manufacturer will consider inertial forces when rating compressors, useful data on this point are seldom available in the field. Except in special cases, inertial forces are ignored.

A tail-rod cylinder would require consideration of rod cross-section area on both sides of the piston instead of on only one side of the piston, as in Eqs. (7.19) and (7.20).

Tail-rod



Horsepower

Detailed compressor horsepower calculations can be made through the use of Figs. 7.66 and 7.67. For ease of calculations, these figures provide net horsepower, including mechanical efficiency and gas losses. Figures 7.68 and 7.69 are included for modifying the horsepower numbers for special conditions.

Proper use of these charts should provide the user with reasonably correct horsepower requirements that are comparable to those calculated by the compressor manufacturer. For more detailed design, the engineer should consult a compressor manufacturer.

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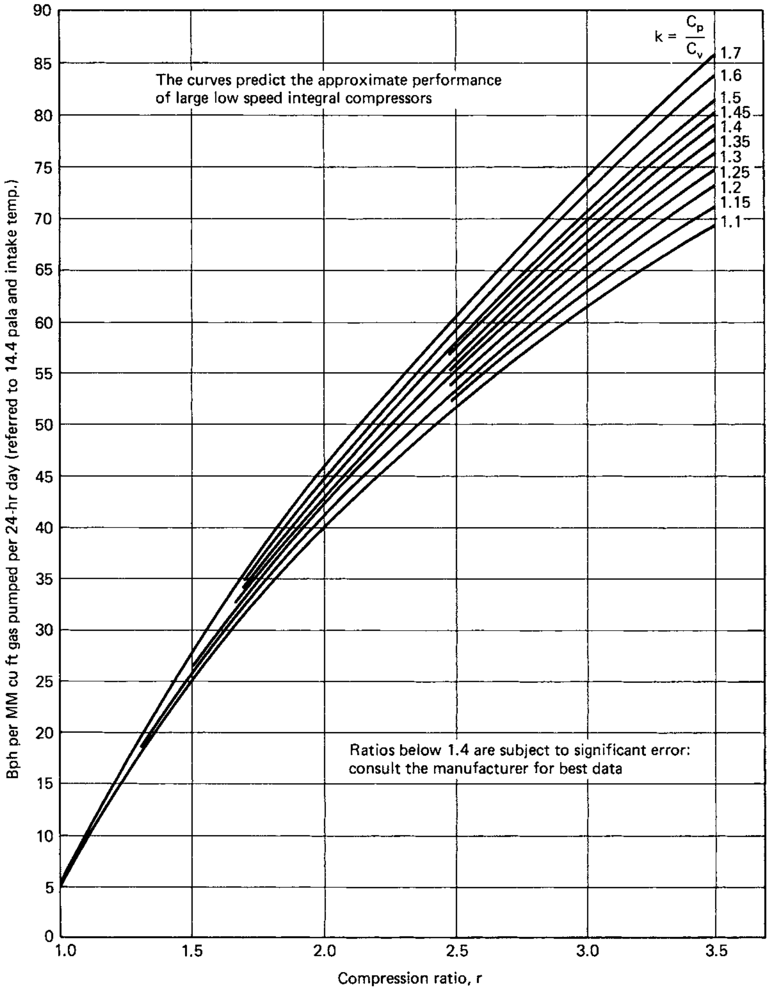

Figure 7.66 Bhp per million curve, mechanical efficiency-95%. Gas velocity through valve-3000 ft./min (API equation).

Volumes to be handled in each stage must be corrected to the actual temperature at the inlet to that stage. Note that moisture content corrections can also be important at low pressure and/or high temperature.

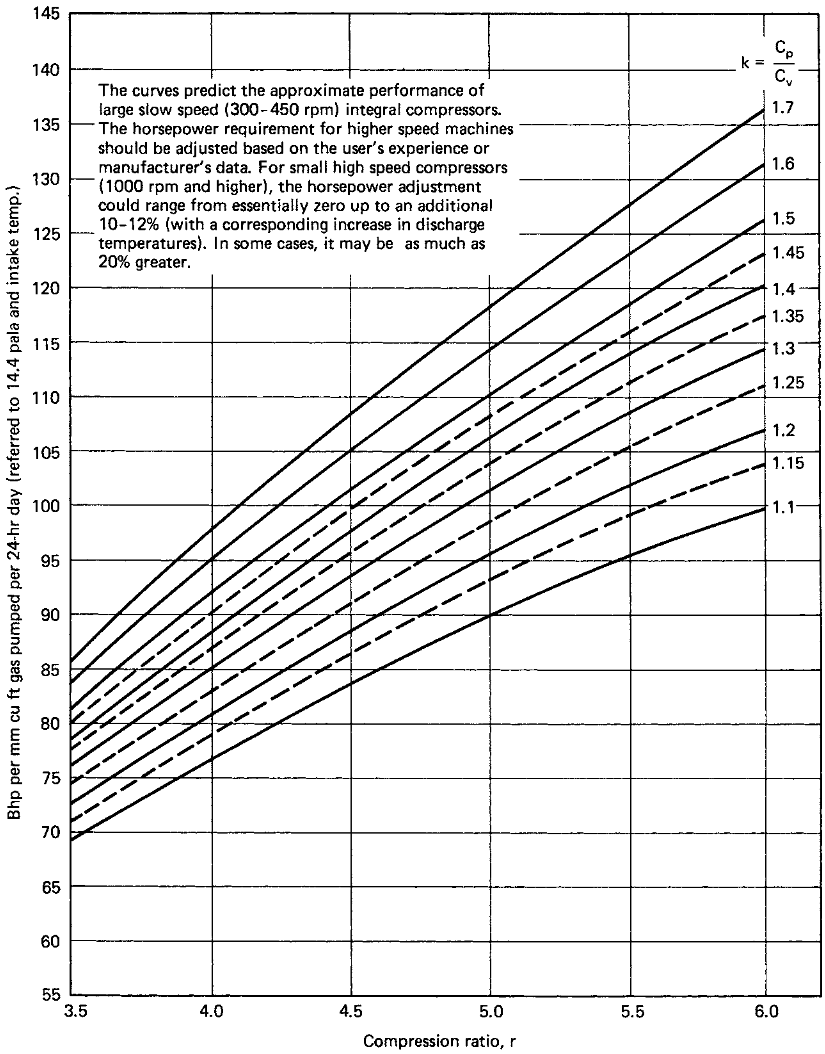

Figure 7.67 Bhp per million curve, mechanical efficiency-95%. Gas velocity through valve – 3000 ft./min (API equation).

When intercoolers are used, allowance must be made for interstage pressure drop. Interstage pressures may be estimated by:

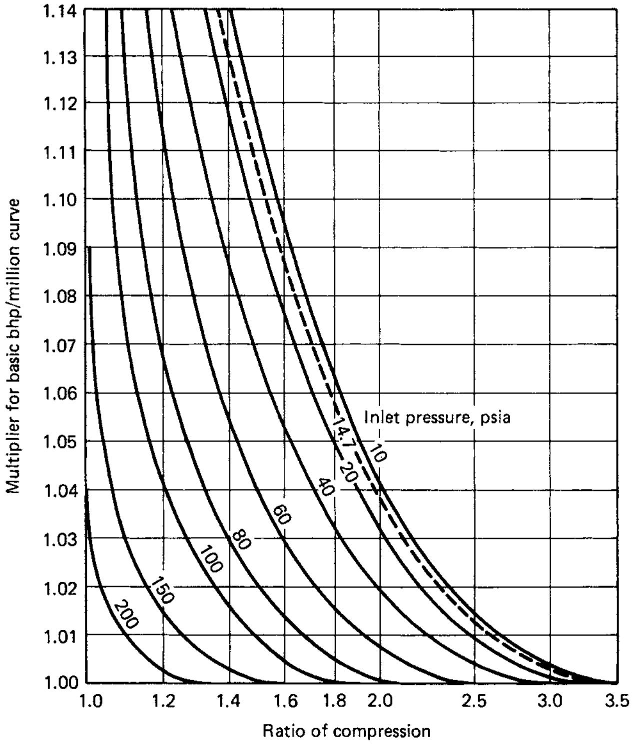

Figure 7.68 Correction factor for low intake pressure.

1. Obtaining the overall compression ratio, r_t.
2. Obtaining the calculated ratio per stage, r, by taking the s root of r_t, where s is the number of compression stages.
3. Multiplying r by the absolute intake pressure of the stage being considered.

This procedure gives the absolute discharge pressure of this stage and the theoretical absolute intake pressure to the next stage. The next stage intake pressure can be corrected for intercooler pressure drop by reducing the pressure by 3 to 5 psi. This can be significant in low pressure stages.

Horsepower for compression is calculated by using Figs. 7.66 and 7.67 and Eq. (7.21).

$$\text{Bhp} - (\text{Bhp/mmcfd})\left(\frac{P_L}{14.4}\right)\left(\frac{T_s}{T_L}\right)(Z_{avg})\,(\text{mmcfd}) \qquad (7.21)$$

Bhp/mmcfd is read from Figs. 7.66 and 7.67, which use a pressure base of 14.4 psia.

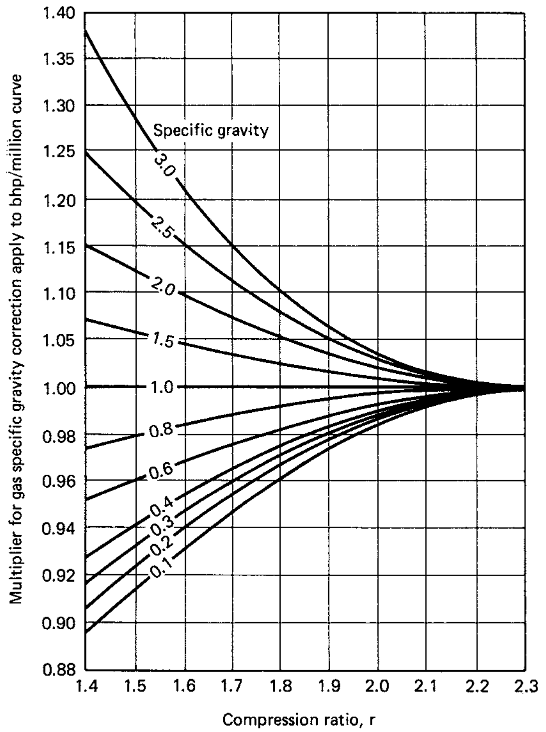

Figure 7.69 Correction factor for specific gravity.

Figures 7.66 and 7.67 are for standard valved cylinders. Caution should be used in applying conventional cylinders to low compression-ratio pipeline compressors. For low ratio pipeline compressors a high clearance type cylinder permits valve designs with higher efficiency. The compressor manufacturer should be consulted for bhp curves on this type cylinder.

Figure 7.68 provides a correction for intake pressure. The correction factor, as read from the curve, is used as a multiplier in the right hand side of Eq. (7.21) to obtain the corrected brake horsepower.

Figure 7.69 provides a correction factor for gas specific gravity. The correction factor is used as a multiplier in the right-hand side of Eq. (7.21) to obtain the corrected horsepower.

Data presented in Figs. 7.66 and 7.67 are for slow-speed integral compressors rather than the high-speed separable compressors. To adjust the horsepower for the high-speed unit, the values obtained from Figs. 7.66 and 7.67 may be increased by the following percentages:

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Gas Specific Gravity	Per Cent Horsepower Increase for High-speed Units
0.5 – 0.8	4
0.9	5
1.0	6
1.1	8
1.5 and propane refrigeration units	10

Because of variations by different manufacturers in specifying valve velocities for high speed as opposed to slow speed compressors, a given unit may differ from the horsepower corrections shown. Experience with compressors from a specific manufacturer will serve to guide the user and give confidence in utilization of the correction factors shown. For applications which are outside typical ranges discussed here, compressor manufacturers should be consulted.

Example 7.2

Compress 2 mm scfd of gas measured at 14.65 psia and 60°F. Intake pressure is 100 psia, and intake temperature is 100°F. Discharge pressure is 900 psia. The gas has a specific gravity of 0.80. What is the required horsepower?

1. Compression ratio is

$$\frac{900 \text{ psia}}{100 \text{ psia}} = 9$$

This would be a two-stage compressor; therefore, the ratio per stage is $\sqrt{9}$ or 3.

2. 100 psia x 3 =300 psia (first-stage discharge pressure)
   300 psia – 5 = 295 psia (suction to second stage)

where the 5 psi represents the pressure drop between first stage discharge and second stage suction.

$$\frac{900 \text{ psia}}{295 \text{ psia}} = 3.05 \text{ (compression ratio for second stage)}$$

It may be desirable to recalculate the interstage pressure to balance the ratios. For this sample problem, however, the first ratios determined will be used.

3. From Fig. 7.62, a gas with specific gravity of 0.8 at 150°F would have an approximate k of 1.21. For most compression applications, the 150°F curve will be adequate. This should be checked after determining the average cylinder temperature.
4. Discharge temperature for the first stage may be obtained by using Fig. 7.65 or solving Eq. (7.18). For a compression ratio of 3, discharge temperature = approximately 220°F. Average cylinder temperature = 160°F.

5. In the same manner, discharge temperature for the second stage (with r = 3.05 and assuming interstage cooling to 120°F) equals approximately 244°F. Average cylinder temperature = 182°F.
6. From the physical properties section in Chapter 8, estimate the compressibility* factors at suction and discharge pressure and temperature of each stage.

* To avoid possible confusion, many authorities refer to Z as supercompressibility, reserving compressibility to mean departures from liquid, or incompressible behavior. For further discussion of this topic see Chapter 8 under the heading *Superconductivity*.

1st stage:	Z_s	=	0.98
	Z_d	=	0.97
	Z_{avg}	=	0.975
2nd stage:	Z_s	=	0.94
	Z_d	=	0.92
	Z_{avg}	=	0.93

7. From Fig. 7.66 Bhp/mmcfd at 3 compression ratio and a k of 1.21 is 63.5 (first stage). From Fig. 7.66, Bhp/mmcfd at 3.05 compression ratio and a k of 1.21 is 64.5 (second stage).
8. There are no corrections to be applied from Fig. 7.68 or 7.69, as all factors read unity.
9. Substituting in Eq. (7.21),

1st stage: $$\text{Bhp/mmscfd} = 63.5\left(\frac{14.65}{14.4}\right)\left(\frac{560}{520}\right)0.975 = 67.8$$

$$\text{Bhp for 1}^{\text{st}}\text{ stage} = 2\text{mm scfd} \times 67.8 = 135.6$$

2nd stage: $$\text{Bhp/mmscfd} = 64.5\left(\frac{14.65}{14.4}\right)\left(\frac{580}{520}\right)0.93 = 68.1$$

$$\text{Bhp for 2nd stage} = (2\text{ mm scfd})\,(68.1) = 136.2$$

$$\text{Total Bhp for this application} = 135.6 + 136.2 = 271.8$$

Note that in Example 7.1 the same conditions result in compression power of 285 Bhp, which is in close agreement.

Limits to Compression Ratio Per Stage

The ratio of compression permissible in one stage is usually limited by the discharge temperature or by rod loading, particularly in the first stage.

When handling gases containing oxygen, which could support combustion, there is a possibility of fire and explosion because of the oil vapors present.

To reduce carbonization of the oil and the danger of fires, the safe operating limit may be considered to be approximately 300°F. Where no oxygen is present in the gas stream, temperatures of 350°F may be considered as the maximum, even though mechanical or process requirements usually dictate a lower figure.

Packing life may be significantly shortened by the dual requirement to seal both high-pressure and high-temperature gases. For this reason, at higher discharge pressures, a temperature closer to 250° or 275°F may be the practical limit.

In summary, and for most field applications, the use of 300°F maximum would be a good average. Recognition of the above variables is, however, still useful.

Economic considerations are also involved because a high ratio of compression will mean a low volumetric efficiency and require a larger cylinder to produce the same capacity. For this reason a high rod loading may result and require a heavier and more expensive frame.

Where multi-stage operation is involved, equal ratios of compression per stage are used (plus an allowance for piping and cooler losses if necessary) unless otherwise required by process design. For two stages of compression the ratio per stage would approximately equal the square root of the total compression ratio; for three stages, the cube root; etc. In practice, especially in high-pressure work, decreasing the compression ratio in the higher stages to reduce excessive rod loading may prove to be advantageous.

Cylinder Design

Depending on the size of the machine and the number of stages, reciprocating compressors are furnished with cylinders fitted with either single- or double-acting pistons; see examples in Figs. 7.70 through 7.72.

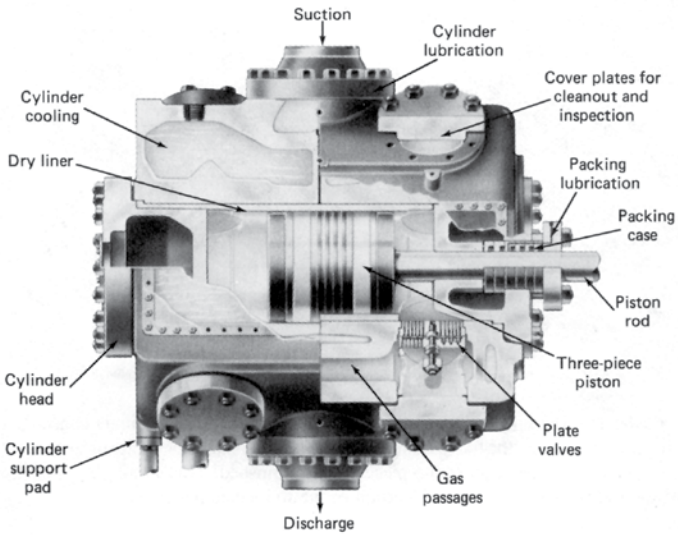

Figure 7.70 Low pressure cylinder with double-acting piston.

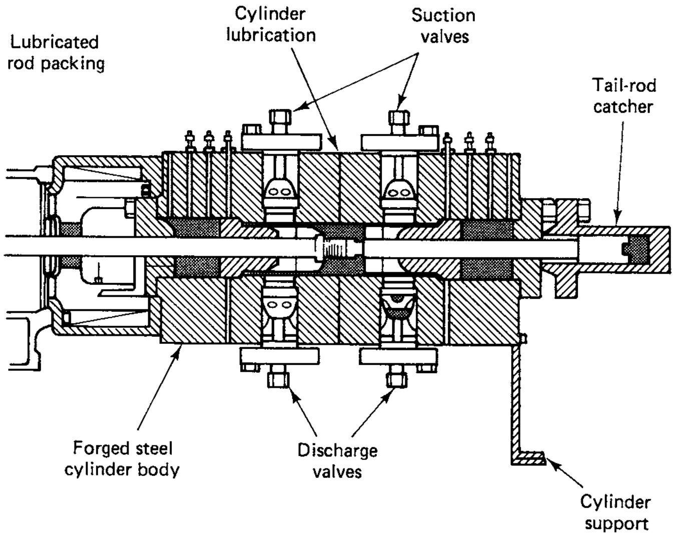

Figure 7.71 High pressure cylinder with double-acting piston and tail-rod.

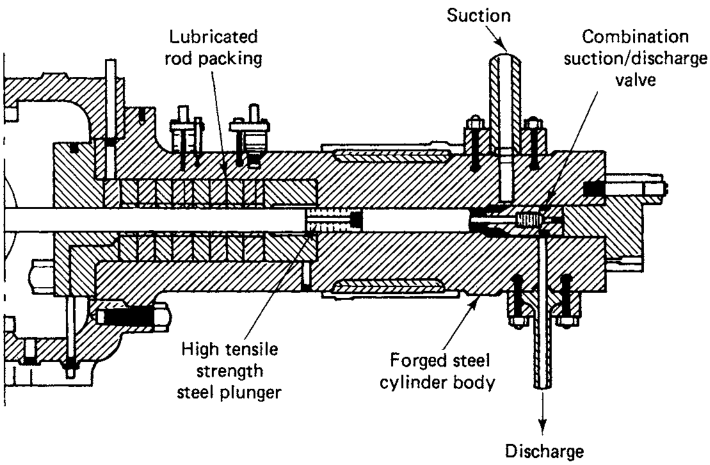

Figure 7.72 Single-acting plunger cylinder designed for 15,000 psig discharge.

In the same units, double-acting pistons are commonly used in the first stages and often single-acting in the higher stages of compression.

Cylinder materials are normally selected for strength; however, thermal shock, mechanical shock, or corrosion resistance may also be a determining factor. The following table shows discharge pressure limits generally used in the gas industry for cylinder material selection.

Cylinder Material	Discharge Pressure (psig)
Cast iron	Up to 1200
Nodular iron	About 1500
Cast steel	1200 to 2500
Forged steel	Above 2500

API standard 618 recommends 1000 psig as the maximum pressure for both cast iron and nodular iron.

Cylinders are designed both as a solid body (no liner) and with liners. Cylinder liners are inserted into the cylinder body to either form or line the pressure wall. There are two types. The wet liner forms the pressure wall as well as the inside wall of the water jacket. The dry type lines the cylinder wall and is not required to add strength.

Standard cylinder liners are cast iron. If cylinders are required to have special corrosion or wear resistance, other materials or special alloys may be needed.

Most compressors use oils to lubricate the cylinder with a mechanical, force-feed lubricator having one or more feeds to each cylinder.

The non-lubricated compressor has found wide application where it is desirable or essential to compress air or gas without contaminating it with lubricating oil.

For such cases a number of manufacturers furnish a “non-lubricated” cylinder (Fig. 7.73). The piston these cylinders is equipped with piston rings of graphitic carbon or plastic as well as pads or rings of the same material to maintain the proper clearance between the piston and cylinder. Plastic packing of a type that requires no lubricant is used in the stuffing box. Although oil-wiper rings are used on the piston rod where it leaves the compressor frame, minute quantities of oil might conceivably enter the Cylinder on the rod. Where even such small amounts of oil are objectionable, an extended cylinder connecting piece can be furnished. This simply lengthens the piston rod so that no lubricated portion of the rod enters the cylinder.

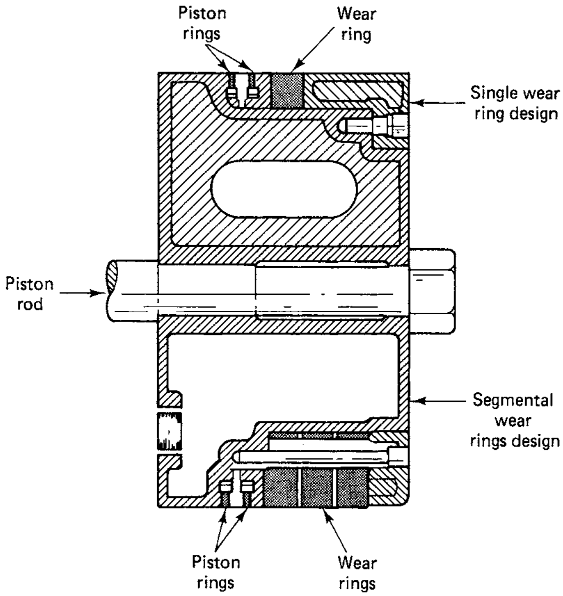

Figure 7.73 Piston equipped with Teflon® piston and wear rings for a single-acting nonlubricated cylinder.

A small amount of gas leaking through the packing can be objectionable. Special distance pieces are furnished between the cylinder and frame, which may be either single-compartment or double-compartment. These may be furnished gastight and vented back to the suction, or may be filled with a sealing gas or fluid and held under a slight pressure, or simply vented.

Compressor valves for non-lubricated service operate in an environment that has no lubricant in the gas or in the cylinder. Therefore, the selection of valve materials is important to prevent excessive wear

Piston rod packing universally used in non-lubricated compressors is of the full-floating mechanical type, consisting of a case containing pairs of either carbon or plastic (TFE) rings of conventional design.

When handling oxygen and other gases such as nitrogen and helium, it is absolutely necessary that all traces of hydrocarbons in cylinders be removed. With oxygen, this is required for safety; with other gases, to prevent system contamination.

High-pressure compressors with discharge pressures from 5,000 to 30,000 psi usually require special design and a complete knowledge of the characteristics of the gas.

As a rule, inlet and discharge gas pipe connections on the cylinder are fitted with flanges of the same rating for the following reasons:

1. Practicality and uniformity of casting and machining.
2. Hydrostatic test, usually at 150% design pressure.
3. Suction pulsation bottles are usually designed for the same pressure as the discharge bottle (often by federal, state, or local government regulation).

Reciprocating Compressor Control Devices

Output of compressors must be controlled (regulated) to match system demand. In many installations some means of controlling the output of the compressor is necessary. Often constant flow is required despite variations in discharge pressure, and the control device must operate to maintain a constant compressor capacity. Compressor capacity, speed, or pressure may be varied in accordance with the requirements. The nature of the control device will depend on the regulating variable, whether pressure, flow, temperature, or some other variable, and on type of compressor driver.

Unloading at startup. Practically all reciprocating compressors must be unloaded to some degree before starting so that the driver torque available during acceleration is not exceeded. Both manual and automatic compressor startup unloading is used. Common methods of unloading include discharge venting, discharge to suction bypass, and holding open the inlet valves using valve lifters.

Capacity control. The most common requirement is regulation of capacity. Many capacity controls, or unloading devices, as they are usually termed, are actuated by the pressure on the discharge side of the compressor. A falling pressure indicates that gas is being used faster than it is being compressed and that more gas is required. A rising pressure indicates that more gas is being compressed than is being used and that less gas is required.

A common method of controlling the capacity of a compressor is to vary the speed. This method is applicable to steam-driven compressors and to units driven by internal-combustion engines. In these cases the regulator actuates the steam-admission or fuel-admission valve on the compressor driver to control the speed.

Electric-motor-driven compressors usually operate at constant speed, and other methods of controlling the capacity are necessary. On reciprocating compressors up to about 100 hp, two types of control are usually available. These are automatic start-and-stop control and constant-speed control.

Automatic start-and-stop control, as its name implies, stops or starts the compressor by means of a pressure-actuated switch as the gas demand varies. It should be used only when the demand for gas will be intermittent.

Constant-speed control permits the compressor to operate at full speed continuously, loaded part of the time and fully or partially unloaded at other times. Two methods of unloading the compressor with this type of control are in common use: inlet-valve unloaders and clearance unloaders. Inlet-valve unloaders (Fig. 7.74) operate to hold the compressor inlet valves open and thereby prevent compression. Clearance unloaders (Fig. 7.75) consist of pockets or small reservoirs which are opened when unloading is desired. The gas is compressed into them on the compression stroke and expands into the cylinder on the return stroke, reducing the intake of additional gas.

Motor-driven reciprocating compressors above 100 hp in size are usually equipped with a step control. This is in reality a variation of constant-speed control in which unloading is accomplished in a series of steps, varying from full load down to no load.

Five-step control (full load, three-quarter load, one-half load, one-quarter load, and no load) is accomplished by means of clearance pockets. On some makes of machines inlet-valve and clearance control unloading are used in combination.

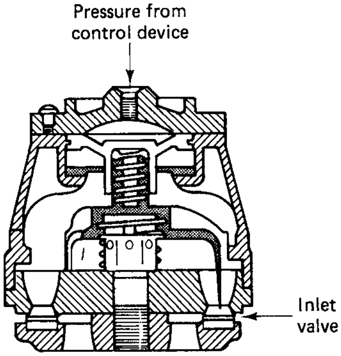

Figure 7.74 Inlet Valve Unloader

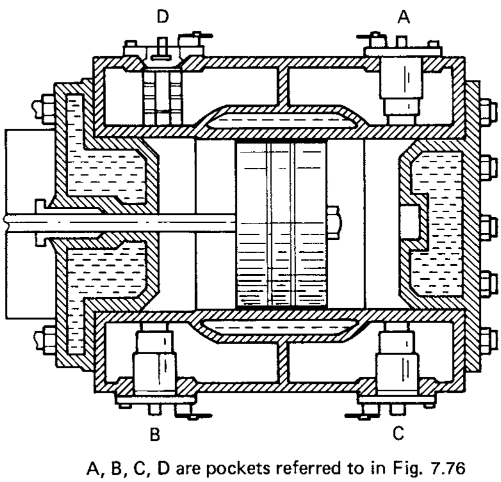

Figure 7.75 Pneumatic valves controlling four fixed pockets in compressor for five-step control.

A common practice in the natural gas industry is to prepare a single set of curves for a given machine unless there are side loads or it is a multiservice machine.

Figure 7.76 shows indicator cards that demonstrate the unloading operation for a double acting cylinder at three capacity points. The letters adjacent to the low-pressure diagrams represent the unloading influence of the respective and accumulative effect of the various pockets as identified in Fig. 7.75. Full-load, one-half and no-load capacity is obtained by holding corresponding suction valves open or adding sufficient clearance to produce a zero volumetric efficiency. No-capacity operation includes holding all suction valves open.

Figure 7.77 shows an alternative representation of compressor unloading operation with a step-control using fixed volume clearance pockets. The curve illustrates the relationship between compressor capacity and driver capacity for a varying compressor suction pressure at a constant discharge pressure and constant speed. The driver can be a gas engine or electric motor.

The purpose of this curve is to determine what steps of unloading are required to prevent the driver and piston rods from serious overloading. All lines are plotted for a single stage compressor.

The driver capacity line indicates the maximum allowable capacity for a given horsepower. The cylinder capacity lines represent the range of pressures calculated with all possible combinations of pockets open and cylinder unloading, as necessary, to cover the capacity of the driver.

Starting at the end (line 0-0) with full cylinder capacity, the line is traced until it crosses the driver capacity line at which point it is dropped to the next largest cylinder capacity and follows it until it crosses the driver line, etc. This will produce a "sawtooth" effect, hence the name "sawtooth" curve. The number of teeth depends on the number of combinations of pockets (opened or closed) required for unloading.

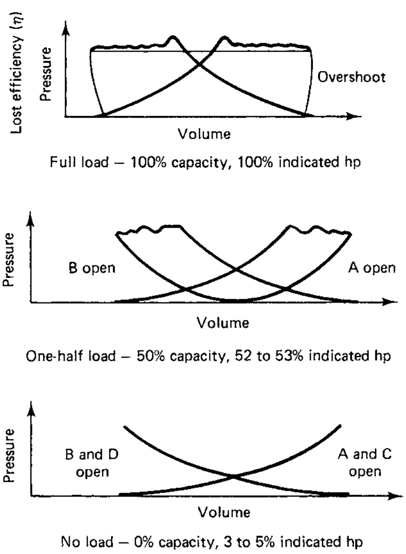

Figure 7.76 Indicator diagram for three load points of operation.

The same method is followed for multi-stage units. For each additional stage another "sawtooth" curve must be constructed (i.e., for a two-stage application); two curves are required to attain the final results.

Although control devices are often automatically operated, manual operation is satisfactory for many services. Where manual operation is provided, it often consists of a valve, or valves, to open and close clearance pockets. In some cases, a movable cylinder head is provided for variable clearance in the cylinder (Fig. 7.78).

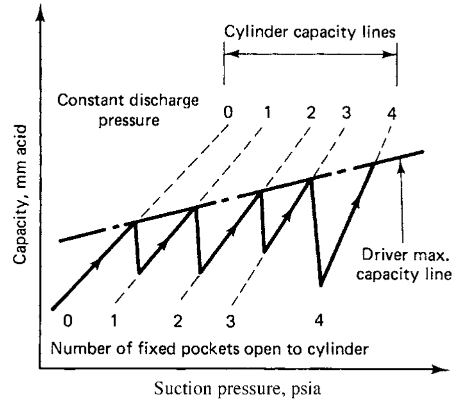

Figure 7.77 "Sawtooth" curve for unloading operation.

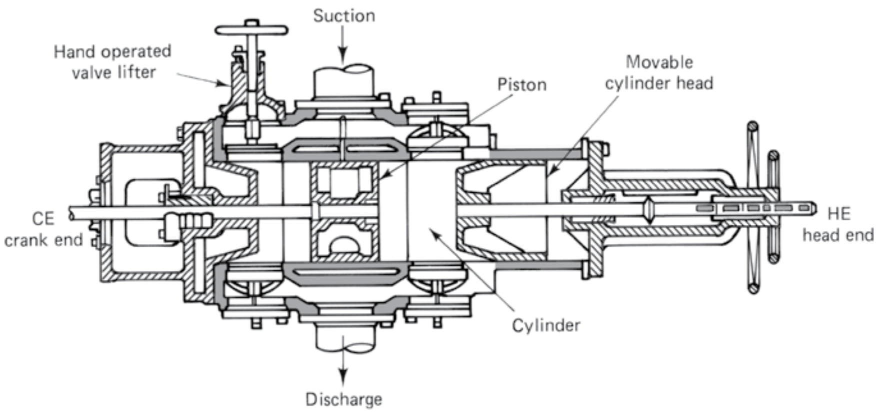

Figure 7.78 Sectional view of a cylinder equipped with a hand-operated valve lifter and variable-volume clearance.

Gas Pulsation Control

Pulsation is inherent in reciprocating compressors because suction and discharge valves are open during only part of the stroke. Pulsation must be damped (controlled) in order to:

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1. Provide smooth flow of gas to and from the compressor.
2. Prevent overloading or underloading of the compressors.
3. Reduce overall vibration.

There are several types of pulsation chambers. The simplest one is a volume bottle, or a surge drum, which is a pressure vessel, unbaffled internally and mounted on or very near a cylinder inlet or outlet. A manifold joining the inlet and discharge connections of cylinders operating in parallel can also serve as a volume bottle. Performance of volume bottles is not normally guaranteed without an analysis of the piping system from the compressor to the first process vessel.

Volume bottles are sized empirically to provide an adequate volume to absorb most of the pulsation. Several industry methods were tried in an effort to produce a reasonable rule-of-thumb for their sizing. Figure 7.79 may be used for approximate bottle sizing.

Example 7.3

Indicated suction pressure = 600 psia
Indicated discharge pressure = 1400 psia
Cylinder bore = 6 in.

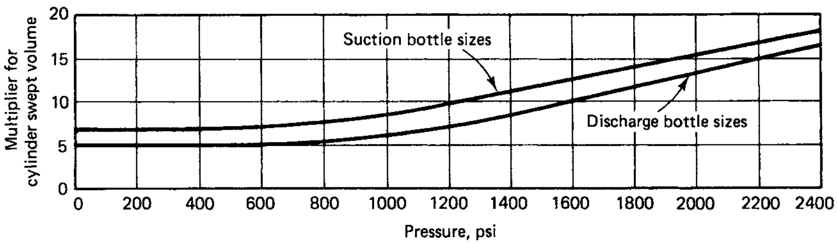

Figure 7.79 Approximate bottle sizing chart.

Cylinder stroke = 15 in.
Swept volume = π (6^2/4) (15) = 424 in.^{3}

From the chart, at 600 psia inlet pressure, the suction bottle multiplier Is approximately 7.5. Suction-bottle volume = (7.5) (424) = 3.180 in.^{3}.

At 1,400 psi discharge pressure, the discharge bottle multiplier is approximately 8.5. Discharge-bottle volume = (8.5) (424) = 3600 in.^{3}.

Note: When more than one cylinder is connected to a bottle, the sum of the individual swept volumes is the size required for the common bottle.

For more accurate sizing, compressor manufacturers can be consulted. Organizations which provide designs and/or equipment for gas-pulsation control are also available.

Having determined the necessary volume of the bottle, the proportioning of diameter and length to provide this volume requires some ingenuity and judgement. It is desirable that manifolds be as short and of as large diameter as is consistent with pressure conditions, space limitations, and appearance.

A good general rule is to make the manifold diameter one and a half times the inside diameter of the largest cylinder connected to it, but this is not always practicable, particularly where large cylinders are involved.

Inside diameter of pipe must be used in figuring manifolds. This is particularly important in high-pressure work and in small sizes where wall cross section may be a considerable part of the inside cross section. Minimum manifold length is determined from cylinder center distances and connecting pipe diameters. Some additions must be made to the minimum thus determined to allow for saddle reinforcements and for welding of caps.

It is customary to close the ends of manifolds with welding caps, which add both volume and length. Figure 7.80 gives approximate volume and length of standard caps.

	Standard weight, Schedule 40		Extra strong, Schedule 80		Double Extra strong	
Pipe size	Volume, cu in.	Length, in.	Volume, cu in.	Length, in.	Volume, cu in.	Length, in.
4”	24.2	2 1/2	20.0	2 1/2	15	3
6”	77.3	3 1/2	65.7	3 1/2	48	4
8”	148.5	4 11/16	122.3	4 11/16	120	5
10”	295.6	5 3/4	264.4	8 3/4		
12”	517.0	6 7/8	475.0	6 7/8		
14”	684.6	7 13/16	640.0	7 13/16		
16”	967.6	9	911.0	9		
18”	1432.6	10 1/16	1363.0	10 1/16		
20”	2026.4	11 1/4	1938.0	11 1/4		
24”	3451.0	13 7/16	3313.0	13 7/16		

Figure 7.80 Welding Caps

Pulsation Dampers (Snubbers)

A pulsation damper is an internally baffled device. The design of the pulsation damping equipment is based on acoustical analog evaluation which takes into account the specified operating speed range, conditions of unloading, and variations in gas composition.

Analog evaluation is accomplished with an active analog that simulates the entire compressor, pulsation dampers, piping and equipment system and considers dynamic interactions among these elements.

Pulsation dampers also should be mounted as close as possible to the cylinder; and in large volume units, nozzles should be located near the center of the chamber to reduce unbalanced forces.

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Pulsation dampers are typically guaranteed for a maximum residual peak-to-peak pulsation pressure of 2 percent of average absolute pressure at the point of connection to the piping system, and pressure drop through the equipment of not more than 1 percent of the absolute pressure. This applies at design condition and not necessarily for other operating pressures and flows. A detailed discussion of recommended design approaches for pulsation suppression devices is presented in API Standard 618, Reciprocating Compressors for General Refinery Services.

As pressure vessels, all pulsation chambers (volume bottles and dampers) are generally built to Section VIII of ASME Code and suitable for applicable cylinder relief valve set pressure.

Suction pulsation chambers are often designed for the same pressure as the discharge units, or for a minimum of two-thirds of the design discharge pressure.

Troubleshooting

Minor troubles can normally be expected at various times during routine operation of the compressor. These troubles are most often traced to dirt, liquid, and maladjustment, or to operating personnel being unfamiliar with functions of the various machine parts and systems. Difficulties of this type can usually be corrected by cleaning, proper adjustment, elimination of an adverse condition, or quick replacement of a relatively minor part.

TROUBLE	PROBABLE CAUSE(S)
COMPRESSOR WILL NOT START	1. Power supply failure. 2. Switchgear or starting panel. 3. Low oil pressure shut down switch. 4. Control panel.
MOTOR WILL NOT SYNCHRONIZE	1. Low voltage. 2. Excessive starting torque. 3. Incorrect power factor. 4. Excitation voltage failure.
LOW OIL PRESSURE	1. Oil pump failure. 2. Oil foaming from counterweights striking oil surface. 3. Cold oil. 4. Dirty oil filter. 5. Interior frame oil leaks. 6. Excessive leakage at bearing shim tabs and or bearings. 7. Improper low oil pressure switch setting. 8. Low gear oil pump by-pass/relief valve setting. 9. Defective pressure gage. 10. Plugged oil sump strainer. 11. Defective oil relief valve.
NOISE IN CYLINDER	1. Loose piston. 2. Piston hitting outer head or frame end of cylinder. 3. Loose crosshead lock nut. 4. Broken or leaking valve(s). 5. Worn or broken piston rings or expanders. 6. Valve improperly seated damaged seat gasket. 7. Free air unloader plunger chattering.
EXCESSIVE PACKING LEAKAGE	1. Worn packing rings. 2. Improper lube oil and or insufficient lube rate (blue rings). 3. Dirt in packing. 4. Excessive rate of pressure increase. 5. Packing rings assembled incorrectly. 6. Improper ring side or end gap clearance. 7. Plugged packing vent system. 8. Scored piston rod. 9. Excessive piston rod run-out.

TROUBLE	PROBABLE CAUSE(S)
PACKING OVER-HEATING	1. Lubrication failure. 2. Improper lube oil and/ or insufficient lube rate. 3. Insufficient cooling.
EXCESSIVE CARBON ON VALVES	1. Excessive lube oil. 2. Improper lube oil (too light, high carbon residue). 3. Oil carryover from inlet system or previous stage. 4. Broken or leaking valves causing high temperature. 5. Excessive temperature due to high pressure ratio across cylinders.
RELIEF VALVE POPPING	1. Faulty relief valve. 2. Leaking suction valves or rings on next higher stage. 3. Obstruction (foreign material, rags), blind or valve closed in discharge line.
HIGH DISCHARGE TEMPERATURE	1. Excessive ratio on cylinder due to leaking inlet valves or rings on next higher stage. 2. Fouled intercooler piping. 3. Leaking discharge valves or piston rings. 4. High inlet temperature. 5. Fouled water jackets on cylinder. 6. Improper lube oil and or lube rate.
FRAME KNOCKS	1. Loose crosshead pin, pin caps or crosshead shoes. 2. Loose worn main, crankpin or crosshead bearings. 3. Low oil pressure. 4. Cold oil. 5. Incorrect oil. 6. Knock is actually from cylinder end.
CRANKSHAFT OIL SEAL LEAKS	1. Faulty seal installation. 2. Clogged drain hole.
PISTON ROD OIL SCRAPER LEAKS	1. Worn scraper rings. 2. Scrapers incorrectly assembled. 3. Worn scored rod. 4. Improper fit of rings to rod/side clearance.

Figure 7.81 Probable causes of reciprocating compressor trouble.

Major trouble can usually be traced to long periods of operation with unsuitable coolant or lubrication, careless operation and routine maintenance, or the use of the machine on a service for which it was not intended.

A defective inlet valve can generally be found by feeling the valve cover. It will be much warmer than normal. Discharge valve leakage is not as easy to detect since the discharge is always hot. Experienced operators of water-cooled units can usually tell by feel if a particular valve is leaking. The best indication of discharge valve trouble is the discharge temperature. This will rise, sometimes rapidly, when a valve is in poor condition or breaks. This is one very good reason for keeping a record of the discharge temperature from each cylinder.

Recording of the interstage pressure on multi-stage units is valuable because any variation, when operating at a given load point, indicates trouble in one or the other of the two stages. If the pressure drops, the trouble is in the low-pressure cylinder. If it rises, the problem is in the high-pressure cylinder.

Troubleshooting is largely a matter of elimination based on a thorough knowledge of the interrelated functions of the various parts and the effects of adverse conditions. A complete list of possible troubles with their causes and corrections is impractical, but a list of the more frequently encountered troubles and their causes is offered as a guide in Figure 7.81.

SYSTEM COMPONENTS

The process and oil and gas industries have basic API (American Petroleum Institute) specifications that apply to the applications and design of the compressors and system components. In the process industry, API-618 provides reciprocating compressor design criteria. API-11P is the general purchase specification relating to packaged reciprocating compressors for oil and gas production services. Each of these specifications addresses system components like the compressor, driver, instrumentation, pulsation control, coolers, and so on. It is common for the API specifications to refer to other detailed specifications, such as TEMA for coolers, ANSI for piping, and NEMA for motors. ASME B19.3 covers safety standards for process compressors.

Instrumentation

The typical oil field compressor has a basic protection and monitoring control system. Sensors monitor oil and water temperatures, process pressures and temperatures, vibration levels of major components, plus oil and water pressures. Control systems have a first-out function to indicate to the operator the cause of equipment shutdown. On engine-driven compressors, the controls are powered by the ignition system or are pneumatic, using field-gas or instrument air.

Figure 7.82 shows a pneumatic control panel for an oil field compressor.

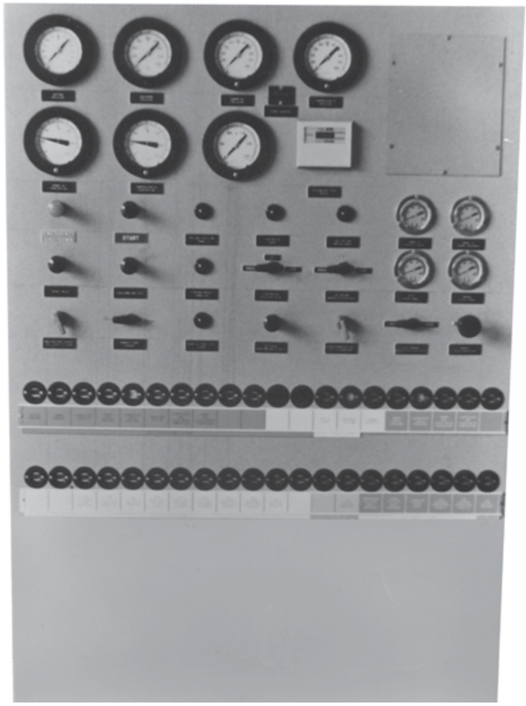

Figure 7.82 Pneumatic Control Panel

Electric-motor-driven compressors usually use electric power for the controls; however, pneumatic controls are sometimes used. In nonhazardous locations, electric controls are generally used; however, pneumatic controls are sometimes used. In hazardous locations, pneumatic controls are generally used.

Control Systems on process machinery are often much more complex. It is common to utilize all logic control systems that will handle full start-up and loading procedures of the machine, plus automatic capacity adjustment. The increased use of microprocessors provides another dimension to instrumentation. Microprocessors can perform calculations with the recorded data, and more precise process control is possible.

Gas Piping

Process gas piping for compressor applications is generally designed and fabricated in accordance with ANSI B-31.3. Compressor piping should be carefully arranged to avoid strains that may cause machinery misalignment. Misalignment is a frequent cause of vibration, and most misalignment is directly traceable to piping strains. On packaged gas field compressors, piping runs are normally short enough that pipe strain due to thermal expansion generally does not create a problem.

Piping must be sized to minimize pressure drops. On oil field applications with 0.65-specific gravity natural gas, compressor packages typically use a design velocity limitation of 2500 to 3000 ft./min. Package compressors normally utilize pipe clamps to support and hold the piping in place.

On process machines, which normally involve longer pipe runs, care must be taken to allow for reduction in strain on the machine due to (1) the dead weight of the piping itself, or (2) expansion or contraction of the pipe as it undergoes temperature change. Prevention of excessive dead loads on the equipment flanges is normally handled with flexible-type supports. Strain due to expansion or contraction is controlled by means of a bellows-type expansion joint on low-pressure applications. High-pressure installations utilize an expansion loop or pipe bend adjacent to the flange of the compressor to control pipe strain.

Pulsation Control

Since the flow into and out of a reciprocating compressor is cyclic, pressure pulses are formed in the suction and discharge gas piping. Pulsations are also directly related to rotative speed, compressor stroke, and gas passage volume. In some cases, high rpm, short stroke volume bottles can be eliminated entirely. However, if pulsations are not minimized and controlled, severe vibration or compressor valve breakage, or both, can occur, resulting in a hazardous operating condition. Additionally, the efficiency of the compressor may be affected.

One method of pulsation control uses volume bottles (with no internal parts) mounted on both the inlet and discharge of each cylinder. Bottles should be mounted as close to the cylinder as possible. Figure 7.83 shows typical pulsation bottles mounted on a compressor cylinder. Historically, a rule of thumb formerly used, hoping to achieve a design peak-to-peak pulsation level of about 5 percent for a 0.65-specific gravity natural gas at less than 1500 psi, was to make the suction bottle equal to 12 times the swept volume of the cylinder and the discharge bottle 10 times the swept volume. It should be pointed out that this is not a very sure method and does not assure avoidance of resonant conditions and high peak-to-peak pulsations. For critical services, or where it is preferred to supply smaller bottles to achieve a given degree of pulsation control, vessels with internal baffles and choke tubes are furnished. Analog and digital modeling techniques are available to simulate pulsation levels throughout a system. These simulations are done regularly in the process industry to provide vessel and piping designs for reliable, long-term compressor and piping system operation. Typical pulsation levels with an acoustic analysis are 2 percent peak-to-peak or less. Generally, it is recommended to do a mechanical and acoustical study for each installation.

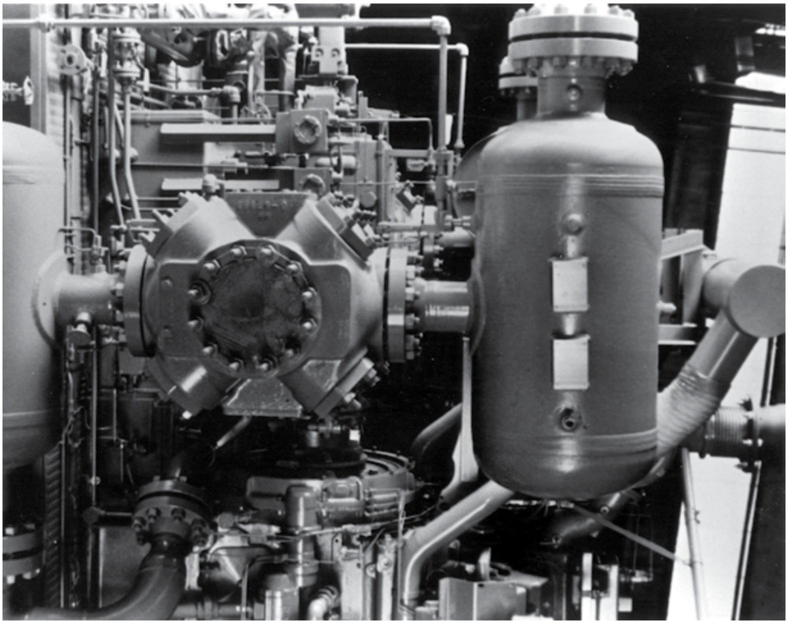

Figure 7.83 Pulsation bottles mounted on a cylinder.

Pulsation bottles are considered unfired pressure vessels and are generally designed and built to ASME Code Section VIII. It is a good design practice to use reinforcing pads on all nozzle connections for pulsation bottles.

Analog analysis. The entire piping and compressor system can be studied by an electronic analog device. The analog simulates the flow of gas through piping as if it were alternating current. The piping system is simulated by an equivalent electrical system. The voltage drop through the system due to electrical resistance is correlated to the pressure drop through piping and heat exchangers. Electrical resistors are used to simulate flow friction through an orifice, the orifice being represented as a linear device with the correct pressure drop at the average flow rate. The compressor system, including pulsation bottles, coolers, and gas piping on the machine and the customer's piping from the machine to the first major vessel of the user's system, is scanned by the analog. If large pulsations are detected, the piping system is modified on the analog hoard, simulating physical changes to the system that would correct the gas flow pattern of the actual system.

The cathode tube gives an image of the pressure-volume card inside the compressor cylinder. The waveform through any section of the piping can he viewed. If acoustical resonance in the piping system is at the same frequency as a major harmonic of compressor operating speed, or if the vessels are too small, pulsation amplitude may be excessive. Pulsation levels can be reduced by changing the size and configuration of the vessel, adding orifices to the piping, or modifying the length and diameter of pipe in the system.

The pressure-volume diagram of each compressor cylinder can be viewed. The card will be distorted if the piping to and from the compressor cylinder causes high pulsation levels. It is very possible that the waveform is such that it adversely affects the opening or closing of the valve(s). Whether this distortion is detrimental can only he determined by calculating the effect of the pulsations on the valve operation. If distortion occurs in the actual compressor, the valves may be slammed closed, resulting in failure of valve plates or valve springs, or both. Pulsations can develop dynamic wave action, which could cause supercharging or starvation of gas to the cylinder. Capacity may he increased or decreased accordingly. Part load operation of the compressor should be scrutinized by the analog method.

Digital analysis. A digital analysis can be used to provide piping system analysis similar to that of the analog method. Two methods are commonly used; each has advantages and disadvantages relative to the other and to the analog simulation.

The analog and impedance digital methods treat the system as a series of pipes connected together. The flow in each pipe is assumed to be one dimensional. In a digital analysis using the impedance method, the response of each pipe to the frequency of interest is calculated. By combining these, the response of the complete system is obtained. When this system is stimulated by pulsations of the frequency and amplitude produced by the cylinder, the pulsations in the piping system are obtained. This is a linear analysis, and the various harmonics of interest are calculated separately. The assumptions made are the same as those made in an analog study. The advantages of this method over the analog are that (1) specialized equipment is not needed and the size or number of jobs that can he analyzed is not limited by the special equipment available, and (2) the time taken to set up a digital calculation is less than that required to build an analog of the piping system. The advantage of the analog is that once the analog is set up on the board, runs representing variable-speed or unloading conditions can he made instantaneously, whereas separate runs are required with a digital analysis.

The assumption that all effects are linear, which is inherent to the analog and impedance methods, precludes the accurate simulation of losses through orifices, compressor valves, and other nonlinear components.

In the time-domain method of digital analysis, a small time step (representing a few degrees of crankshaft rotation) is chosen and the solution built up by successively calculating pressures and temperatures in the cylinder and piping at each time step through the cycle. This method is nonlinear and can accurately simulate conditions in the cylinder, the pressure drop, and the dynamics of the compressor valves, orifices, pipe friction, and other nonlinear losses. It does, however, require significant amounts of computer time and is usually best used for a check of the final design, including investigation of the effect of the pulsations on the valve dynamics after the design has been developed using the impedance or analog methods.

It is also possible to combine an impedance analysis of piping with a time-domain solution for the cylinders and valves. It is then necessary to iterate to obtain the interaction between the two.

Mechanical analysis. The acoustic analysis of the piping system determines the predominant gas pulsation frequencies present in the system. The mechanical analysis is completed to assure that these frequencies do not excite the system's mechanical natural frequencies to cause excessive piping vibration. A review of the type and location of supports is included with the mechanical analysis.

Separation

Reciprocating compressor valves are not very tolerant of liquids in the gas stream, so proper separation of liquids from the gas stream is essential to ensure successful operation. The two most common types of scrubbers used in the oil and gas industry for compressors are a stainless-steel mesh pad or vanes. Figure 7.84 shows a typical oil field compressor scrubber. The principle of both designs is the same; they use a velocity reduction and a sharp change in direction to drop out large liquid droplets and particulate contamination. The gas then makes repeated changes in flow direction as it passes through a 4 to 6 in. mesh pad or through vanes, depending on the design. This change of direction causes entrained liquid to coalesce and drop out of the gas stream. The stainless mesh pad has the advantage of being cheaper than the vanes and has a wider capacity operating range. However, the efficiency of the vane-type scrubber is higher and a smaller diameter scrubber can be designed. The vanes are especially attractive for larger separators.

These same scrubbers are used in the process industry along with many specialized separators for individual processes. When there are particulate contaminants, as well as liquid to be separated from the gas stream, multi-stage coalescing filter-separators are used. These have replaceable or cleanable filter elements, or both, and provide a high degree of protection.

Centrifugal-action separator elements are often used for process applications. Locating the separator element as part of the suction pulsation vessel provides additional protection close to the cylinder. This ensures that most entrained liquids are removed before the gas enters the cylinder. Scrubbers are considered unfired pressure vessels and are usually designed to ASME Code Section VIII. It is also recommended that reinforcing pads be used in all major nozzle connections for scrubbers.

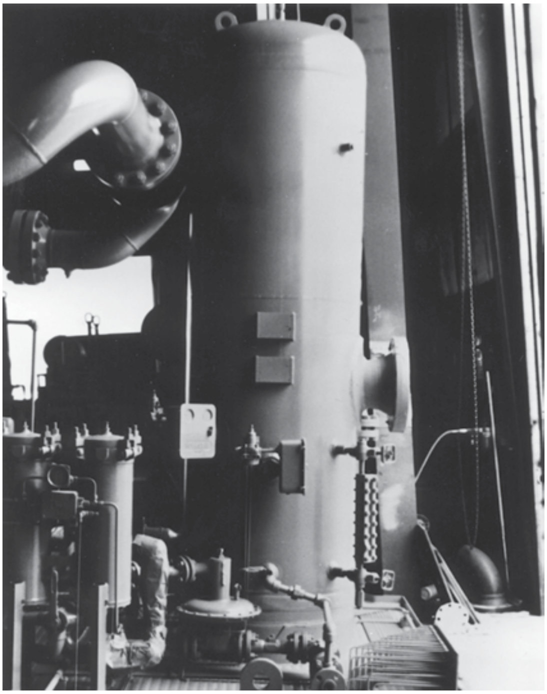

Figure 7.84 Oil Field Gas Scrubber

Cooling

During the compression of gas, heat is generated. For compression ratios greater than 3, intercooling of the gas stream may be required. Cooling of the gas stream has two objectives: (1) to reduce the gas temperature and volume, and (2) to condense and remove water vapor or other condensable constituents. Both of these objectives tend to reduce power requirements. Cooling after the last stage of compression (aftercooling) does not reduce power, but is done to promote safety and to liquefy condensables that might otherwise deposit under undesirable conditions.

Water cooling is required to cool auxiliary support systems such as engine jacket water or lube oil and cylinder jacket water, when required. Water also can be used as a primary cooling medium for compression.

The two main types of coolers used are air cooler and water coolers.

Air-cooled designs. Air coolers are of fin-tube construction with the compressed gas passing through the tube. A cooler may consist of a single cooler section or several sectionalized assemblies. Air coolers are most common in the oil field where cooling water is scarce or of poor quality. Air coolers transfer the heat from the compressed gas or water to the atmosphere. On engine-driven packages, the cooler fan would typically be driven from the engine, and the engine cooling would be in a closed system using a radiator section in the cooler to discharge the engine's heat to the atmosphere. Figure 7.85 shows an air-cooled cooler used for oil field applications.

Figure 7.85 Air-cooled heat exchanger during transport to job site.

The two major types of air-cooled heat exchangers are induced draft and forced draft. Generally, the induced draft design is less expensive to fabricate; however, it does require more fan horsepower to provide the same amount of cooling. Project economics will dictate which design is applicable.

Water-cooled designs. When cooling water of good quality is available, shell and tube coolers are more economical and also much quieter. There are again two basic designs of water-cooled heat exchangers: (1) gas in the tubes, water in the shell, and (2) gas in the shell, water in the tubes. There are advantages and disadvantages to both designs. Proper selection for a particular application depends on cooler size, pressure limitations, and relative cleanliness of the gas and water.

Process compressors usually utilize shell and tube coolers as they are normally installed in process plants or refineries. API-618 is a reference for the major specification that would apply to cooler designs for process compressors.

Utility Piping

Utility piping consists of piping for oil, water, vents, drains, fuel, or gas engine supply. In general, care must be taken in design and fabrication of utility piping to ensure the following:

1. Cleanliness.

2. Ease of operation and maintenance (i.e., drain and fill valves on all systems should be placed for easy access by the operator). High points on water lines should have vent valves, and low points should have drain valves. Break flanges, unions, or isolation valves should be placed around common maintenance items like auxiliary water pumps.
3. Line sizes should be adequate to minimize pressure drops in water and oil systems.

Package

Skid. Process compressors are normally not skid mounted. In oil field applications, however, they are normally skid mounted packages. The driver, compressor, cooler, and all the necessary items are mounted and aligned on a skid. Thus, a properly designed and fabricated skid is of utmost importance. Figure 7.86 shows a typical oil field compressor skid. The skid must be designed to withstand the loading and handling of the equipment during the move from the fabricating facility to the job site. It must also withstand moves from one job site to another, as packaged compressors are frequently relocated when operating conditions change. It must be large enough to support all the auxiliary equipment and still provide adequate working space around the machine for maintenance and operation. At the same time, it must be compact enough to allow shipment to a job site without disassembly. On smaller units, less than 800 hp, skids are often concrete filled. This minimizes the foundation requirements, and in some cases the units can be installed on a firm, level underbase such as packed gravel.

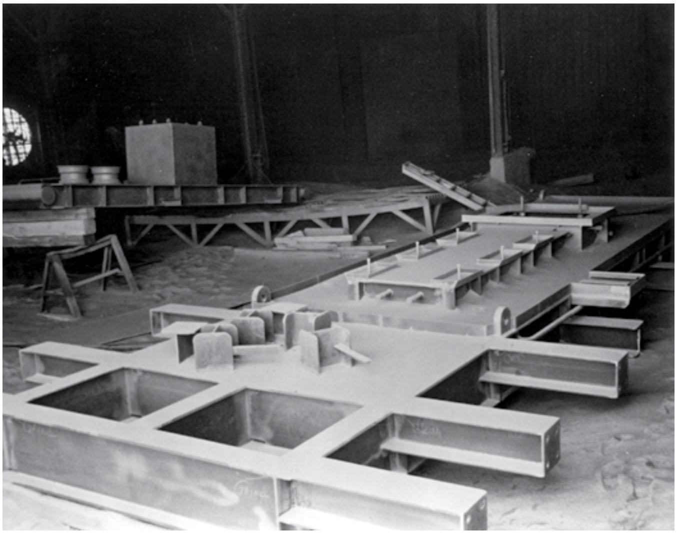

Figure 7.86 Separable Oil Field Compressor Skid

Transportation costs will increase due to additional weight. Proper alignment of the engine and compressor on the skid is such a key fabricating consideration that most compressor packagers optically align these components on the assembly floor prior to final shimming or grouting of the engine or compressor to the skid. Extra time spent during assembly is paid back several fold by easier, quicker installation in the field.

Assembly. During the design and assembly of a compressor package, care must be taken to ensure that safety, ease of operation, and maintenance are given high priority. Inspection openings must he clear of obstructions so they can be properly utilized. Instrumentation should be placed and oriented so it is clearly visible to the operator standing at ground level. There must be sufficient space on the package so that the operator has normal access for maintenance on items such as lubricator pumps, separator dump valves, compressors, engines, and other routine maintenance items.

The package must be designed to provide maximum operator safety. All belts, couplings, and flywheels should be totally enclosed by guards that meet the necessary standards. Provisions should be made to pipe relief valves to a safe location (often to the top of the cooler). All shutdown instrumentation should be calibrated and tested. If the unit is to be unused or stored for a period of time, provision must be made for protection of exposed surfaces, instruments, and anything else that could be damaged by the elements.

Valving

All compressors require process valving to operate them safely. This valving can be furnished either as part of the package or as part of the installation. The following is the minimum valving recommended.

1. *Suction block valve:* A suction block valve is used to isolate the machine from the supply when maintenance is necessary.
2. *Relief valves:* Relief valves should be furnished in the suction line, in each interstage before the cooler, or in the discharge before the cooler. Relief valves are required by ASME, Section VIII, which is a minimum standard to which most pressure vessels are designed and built. Rupture discs are sometimes furnished in lieu of relief valves. The relief valve must be set to protect the component with the minimum design pressure, and it must be large enough to relieve the maximum predicted capacity.
3. *Discharge check valve:* A discharge check valve may be installed immediately downstream from the compressor unit. This protects the unit from being exposed to full line pressure when the machine is shut down.
4. *Discharge block valve:* A discharge block valve is installed downstream from the discharge check valve and provides positive isolation of the machine for maintenance or repair.

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5. *Blowdown valve:* A blowdown valve should be installed preferably between the discharge check and discharge block valves. It is used to depressurize the machine for maintenance purposes.
6. *Bypass valve:* The bypass valve takes gas from compressor discharge upstream of the check valve and allows it to be recycled back to the suction of the machine downstream of the suction block valve. This valve can be manual or automatic. It also is used to unload the machine during start-up.

After the machine is up to idle or operating speed, the bypass valve can be slowly closed to load up the unit. This method of loading will increase the life of the engine driver.

The bypass valve also can be used for capacity-control purposes to maintain a minimum suction pressure or maximum discharge pressure. When used in this manner, it is necessary to use gas downstream of the aftercooler to prevent overheating due to the recycling of hot gas.

In a full logic control system, suction, bypass, blowdown, and discharge valves can all be automated to open and close as required. Figure 7.87 shows a schematic diagram of a standard process valve system for compressor installation.

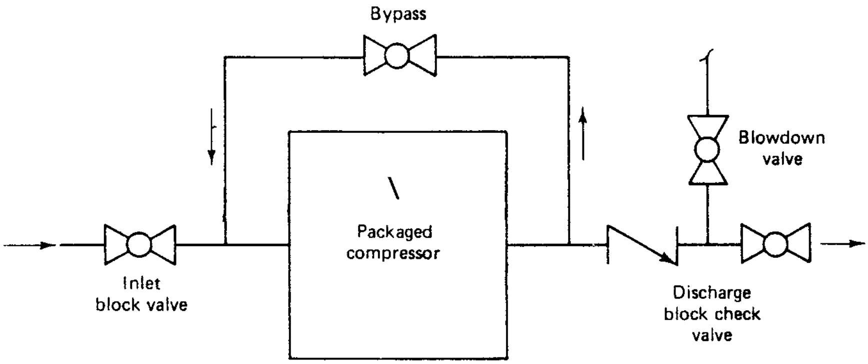

Figure 7.87 Schematic of bypass valve system. Only cooled gas generally is bypassed back to suction. Low percentage flow bypass or short duration bypassing may not require coding.

PRIME MOVERS

The prime mover or driver of a compressor is the main power source that provides the energy to drive the compressor. The driver, through the coupling or other connection, must provide the power to start the compressor, accelerate it to full speed, and keep the unit operating under any design condition of capacity and power.

In the selection of a driver for a positive-displacement compressor, two important factors should he taken into consideration. A complete technical analysis should be done, considering the application, in addition to a total economic analysis.

The technical conditions should include the following items:

1. Application and service requirements of the compressor.
2. Available power sources.
3. Compatibility of driver and compressor.

Commercial consideration should include the following cost items:

1. Driver first cost.
2. Fuel consumption or power cost.
3. Maintenance cost.
4. Installation cost (especially when comparing natural gas versus an electric motor).
5. Environmental considerations.

It is normally to the user's advantage for the compressor supplier to furnish both the driver and accessories so that total responsibility for a properly designed system is assumed by one source.

Natural Gas Engines

Gas engines are normally divided into two general categories related to speed. These categories are slow-speed engines (0 to 600 rpm) and medium-speed engines (600 to 2100 rpm).

Slow speed engines are commonly used in integral gas engine compressors. An integral engine utilizes a common crankshaft to drive both the compressor and power cylinders. Integral machines are further divided into two horsepower groups: small horsepower, 25 to 800 hp, and large horsepower, 800 to 7000 hp.

Small horsepower integral engines are generally used for oil field services such as gas gathering, gas injection, and small gas processing plants, and in some sizes, 25 to 100 hp, are available at up to 1400 rpm. Large horsepower units are used in process plants, main line gas transmission, gas injection, and large gas plants.

Figure 7.88 shows a typical main line gas transmission integral engine installation. Figure 7.89 is an integral gas engine packaged for offshore gas lift.

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Figure 7.88 Gas transmission integral engine compressor.

Medium-speed gas engines (600 to 2100 rpm) are generally used to drive separable oil field compressors. Horsepower sizes range from 5 to 3600 hp. The smaller horsepower driver, 5 to 400 hp, is generally medium speed, 1400 to 1800 rpm, and can be direct connected to a compressor or used as a V-belt driver. Occasionally, a gear speed reducer is used.

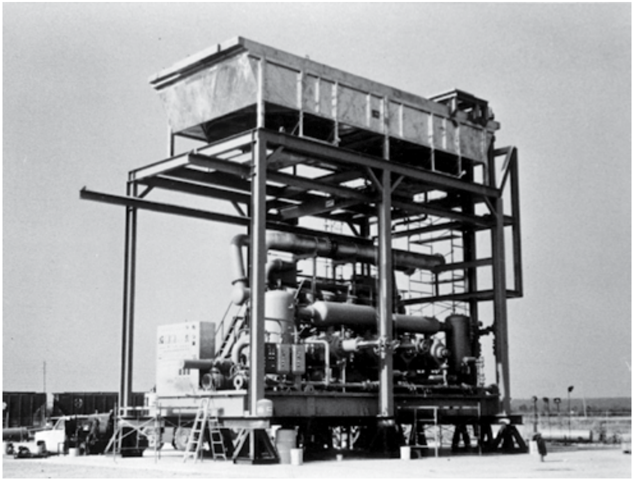

Figure 7.89 Packaged integral gas engine offshore gas lift.

Large horsepower drivers, 300 to 3600 hp, are generally all direct connected and operate in speed ranges from 600 to 1200 rpm. There is a general industry trend to further increase driver speeds, consistent with increased compressor operating speeds. Figure 7.90 shows a typical, packaged, medium-speed, compressor installation.

Figure 7.90 Gas engine-driven two-stage compressor in a mini-gas plant.

The natural gas engine driver is currently the most common driver used in the oil and gas industry Care should be used when selecting a gas engine driver to be certain that the horsepower rating meets the required service. Engine manufacturers can rate their engines for maximum, intermittent, or continuous use, all of which are applicable, depending on the application. Some gas engines are rated on a Diesel Engine Manufacturers Association (DEMA) basis, which is a continuous-use industrial rating.

Recent exhaust emission legislation, both federal and state, makes it necessary to review the emission characteristics of the engine prior to purchase. Low-emission engines or catalytic converters are two potential solutions to exhaust emission requirements.

Electric Motors

Electric motors are the most widely used compressor drivers when considering air compressor and process compressor applications. As previously stated, natural gas engines are the primary oil and gas field drivers; however, electric motors are used more frequently than formerly due to environmental considerations. These motors do not have emission problems.

All major motor manufacturers in the United States build their motors to NEMA (National Electrical Manufacturer Association) standards. There are three basic types of motors available that apply to process and oil field compressors. These are induction motors, synchronous motors, and DC motors. Proper motor selection is a very critical decision and involves several key factors. Each motor selected as a driver should be analyzed from a technical and economic point of view.

Induction and synchronous motors are generally selected according to the chart in Fig. 7.91. This selection chart does not, however, take into account all the economic and technical factors that should be reviewed prior to final selection.

Economic considerations in selecting a motor should include the motor cost, maintenance cost, operating cost, and, most important, an analysis of the accessory equipment required to operate the motor. Some of the key accessories that should he reviewed are the motor starter, motor controls, any transformer requirements, and cost of power lines, if applicable.

The technical considerations include the following items:

- A proper motor-to-compressor speed match
- Torsional analysis, including the flywheel effect (WR^2 of the system)
- Proper motor enclosure for the area classification and weather conditions in which the motor will operate
- Voltage and frequency required
- Speed: torque requirements for starting and operation
- Current restrictions, including KVA inrush required for starting
- Analysis of power-factor corrections in the system
- Altitude of installation
- Ambient temperatures at job site
- Desired motor efficiency
- Number of starts per hour required for normal service
- Service factor of the motor
- Insulation requirements

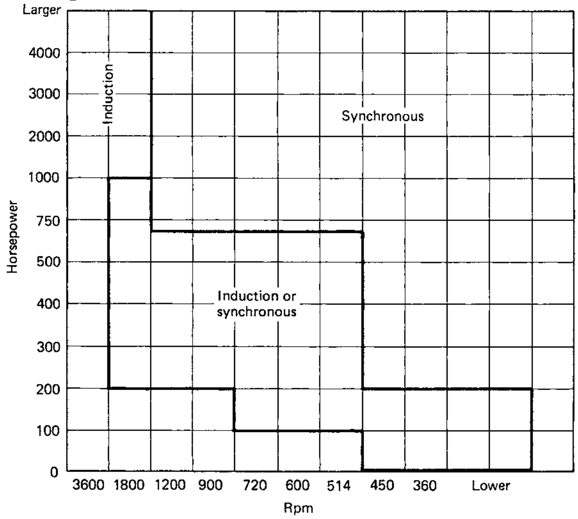

Figure 7.91 Motor selection chart for general areas of application of induction and synchronous motors.

It is very important to apply an electric motor properly. Motors must be designed to deliver a minimum full-load continuous horsepower. Good design practice dictates motor power at least 10% over the worst-case compressor horsepower requirements. This safety factor can be obtained in the service factor of the motor.

Induction motors. The most common compressor motor driver is the induction motor. Induction motors generally have good efficiency and excellent starting torque, but rather high inrush current requirements. Inrush current is the amount of current required to start the motor and driven equipment. Induction motors operate at speeds below synchronous speed by a value known as slip, which varies with the load. Full-load slip varies from 1 percent for very large motors to 5 percent for smaller motors. Induction motor efficiencies are in the high 80 or low 90 percentile, depending on the horsepower available. Smaller-horsepower induction motors are generally less efficient. Figure 7.92 shows an induction-motor-driven oil field compressor package.

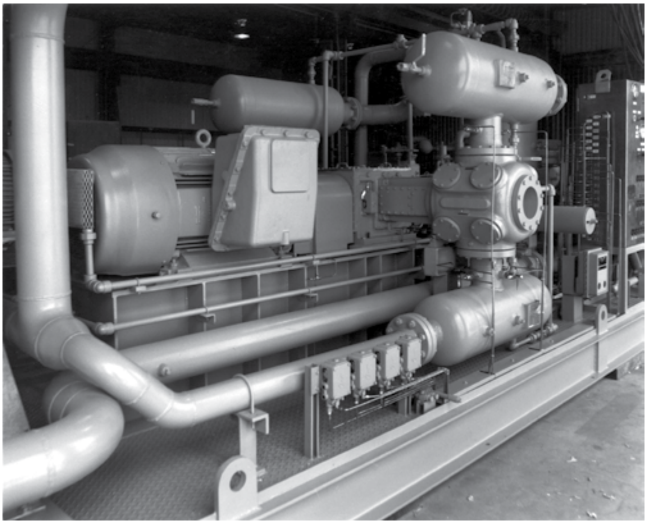

Figure 7.92 Induction Motor-driven Oil Field Compressor Package

Synchronous motors. Synchronous motors are the most common com-pressor drivers used for higher-horsepower applications as may be seen in the chart in Fig. 7.87. These motors are typically more efficient than induction motors, with efficiencies in the range of 93 to 97%.

Synchronous motors must be carefully analyzed because of their lower torque characteristics. The torque requirements must be analyzed along with the starting current inrush requirements to assure that proper starting power will be available. Synchronous motors operate at synchronous speed and do not have the slip characteristics of induction motors. Figure 7.93 shows a typical synchronous-motor-driven process compressor.

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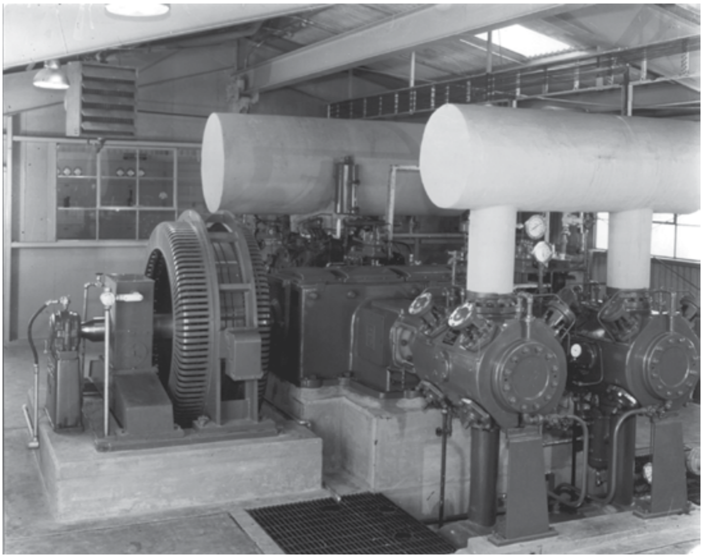

Figure 7.93 Synchronous Motor-driven Process Compressor

DC motors. In the past few years, the use of DC motors as oil field compressor drivers has increased in popularity. The reason for these increases are threefold:

1. Availability of DC traction motors.
2. Variable-speed capabilities of DC motors to control compressor capacity.
3. Economic considerations of motor drive versus engine drive.

Offshore oil field compressors are using more DC motor drivers because of the added speed flexibility, lower initial cost, and projected lower maintenance costs. When utilizing DC motors in a hazardous atmosphere, it is necessary to provide a continuous positive air pressure in the motor enclosure to assure that no gas can get into the motor and be ignited by the motor. Figure 7.94 shows a DC-motor-driven offshore separable package.

There are numerous motor enclosures available to meet various operating requirements. Enclosures can be designed to prevent the entry of water or dirt and permit operation in a hazardous atmosphere.

There are several hazardous-area classification groups. These hazardous groups have been defined by the National Electric Code. Class 1, group D, is the general section that normally applies to compressor motors. This section is broken down into two further divisions:

1. Division 1 locations are those in which hazardous concentrations of flammable gas or vapors exist either continuously or periodically during normal conditions.

2. Division 2 locations are those in which flammable gases are handled, processed, or used. The gases will normally be confined within closed systems from which they can only escape in the event of accidental breakdown or abnormal pressure requirements.

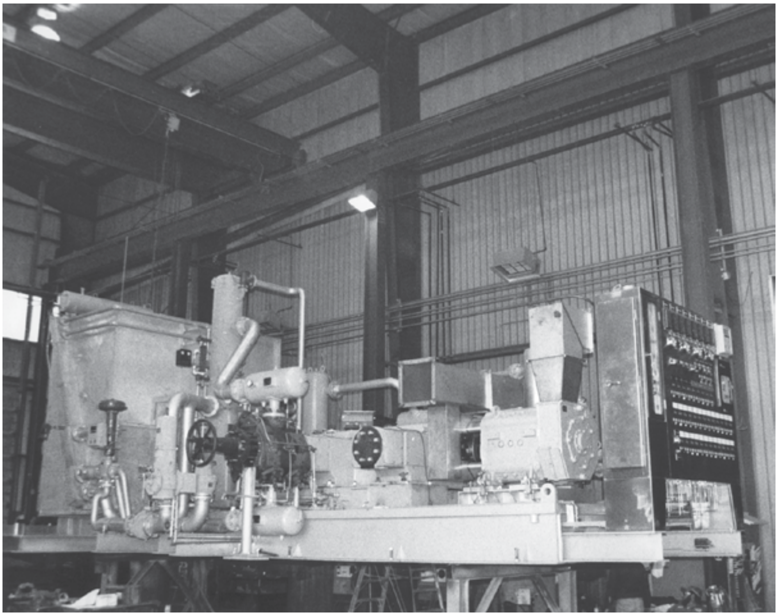

Figure 7.94 Direct Current Motor-driven Separable Gas Compressor

A class 1, group D design is not explosion proof unless specifically certified by Underwriters Laboratories, Inc.

The improvement in the electronics control industry has greatly increased the potential for motors to be utilized as compressor drivers, especially in oil field applications. This has happened because of technological advances in motor controls. It is now economical to buy induction motors or synchronous motors with variable-speed controls to adjust the compressor operating speed. DC motors, having inherent variable-speed capability, already provide the needed variable speed with little further equipment needed. Variable speed to control compressor performance is a very desirable characteristic of a compressor prime mover.

Gasoline Engines

Gasoline engines are seldom used in the process and oil and gas industry because of high fuel cost. They are primarily used as drivers for standby compressors. The operating and application characteristics are similar to those of natural gas or diesel engine drivers.

Diesel Engines

Diesel engines are used rather infrequently as process and oil and gas industry compressor drivers. However, there are some applications, such as air drilling compressors, kick-off compressors (used to start an oil field gas lift), fire floods, or standby compressors, where diesel engines are the most economical drivers.

Diesel engines that can burn either diesel oil or natural gas also have limited applications. The dual-fuel configuration allows the operator to select the most economical fuel. For example, natural gas may be readily available in the summer but not economically available in the cold winter. Thus, the operator can use diesel fuel in the winter and gas in the summer. Some dual-fuel engines can be converted to spark ignition gas engines rather easily. Figure 7.95 shows a diesel-driven air compressor package used for air drilling.

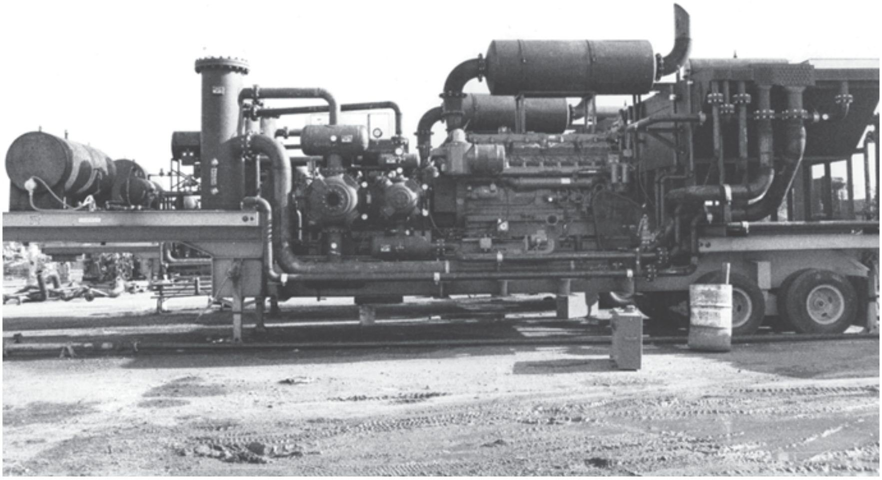

Figure 7.95 Diesel Engine-driven Oil Field Air Compressor

Steam Turbines

Steam-turbine drivers are ordinarily used to drive positive-displacement compressors in process applications where steam is readily available as a power source. Generally, it is not economical to use steam as a driver fuel unless it is available as a result of a process in a refinery or process plant. Figure 7.96 shows a steam-turbine-driven process compressor.

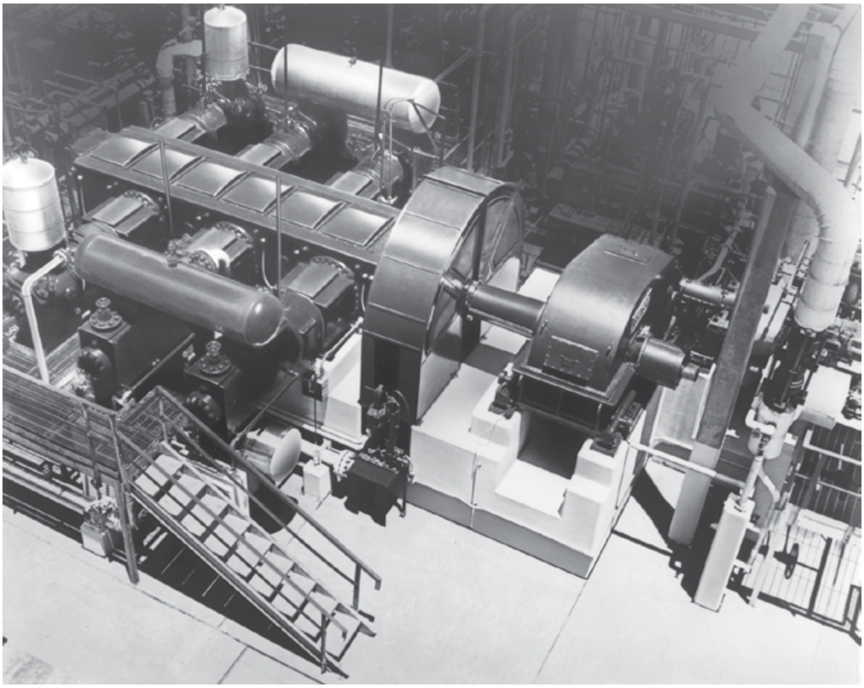

Figure 7.96 Steam turbine-driven process compressor. Note gear box in drive train.

Careful analysis of the mechanical drive train is necessary. A complete torsional analysis must be done by the compressor supplier with particular emphasis on the coupling and quill shaft selection.

Steam turbines for mechanical drive service are available in horsepower ranges from 10 hp up to several thousand horsepower. Speed ranges vary with horsepower size. It is generally possible to find a turbine and compressor speed match so that a single or double reduction gear can be selected for the drive train.

Several types of steam turbines are available. The proper steam turbine should be selected from the various types available to suit the overall economics. Single-stage noncondensing, multi-stage condensing, or multi-stage noncondensing turbines are the types most generally used in process applications.

The steam conditions available will generally determine the type of steam turbine to be used. If the turbine exhaust steam can be utilized in the plant process, a back-pressure turbine should be used. If steam consumption is the prime economic factor, then a condensing turbine should be selected. Many specialty turbines are available, such as mixed-pressure turbines in which the steam not only drives the turbine, but is bled off at various pressures for process use.

A complete steam thermal balance analysis should be done to select the proper design of steam turbine. Steam costs versus alternate fuel costs should be reviewed prior to final selection.

Gas Turbines

Gas turbines used as prime movers for positive-displacement compressors have limited application in the process and oil and gas industry. The gas turbine is relatively new as a driver compared to the gas engine, motor, or steam turbine. However, there are some applications in which gas-turbine-driven reciprocating compressors may have an advantage. One application is offshore compression, where weight is an important consideration. Figure 7.97 shows a gas-turbine-driven, offshore packaged compressor. Another application could be in a refinery or process plant where the gas turbine exhaust heat could be utilized to improve the overall plant thermal efficiency.

Mechanical analysis of the drive train, including a proper torsional analysis and coupling/quill shaft selection, is very important. Other important considerations are the turbine controls and how they interface with the process or compressor control requirements. Proper care must be taken to be sure that the turbine horsepower available exceeds the horsepower required by the driven compressor, and that the torque is adequate throughout the compressor operating range. The loading and unloading steps must also be reviewed to be certain that no operating problems are encountered during normal start-up or shutdown or during emergency shutdown.

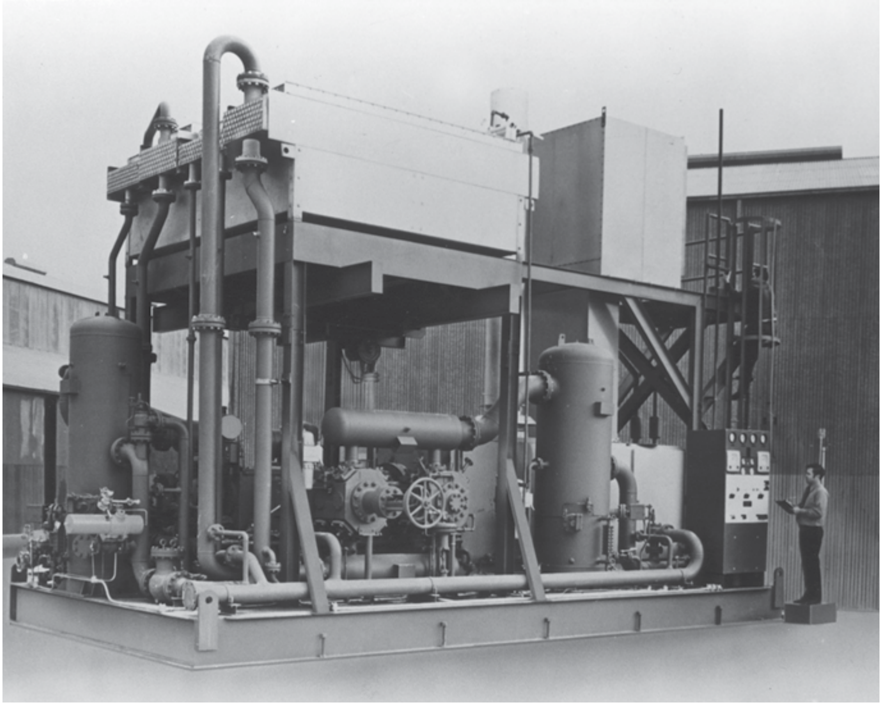

Figure 7.97 Gas-turbine-driven oil field compressor package for offshore application.

Hydraulic Turbines

An hydraulic turbine is basically a centrifugal pump operating in reverse. The application of an hydraulic turbine would be a specialty situation where sufficient high-pressure liquid exists in a refinery or process plant. By decreasing the liquid pressure across the turbine, the pressure of the liquid is reduced to a desirable level and power is recovered. When high-pressure liquid is available, this type of driver offers essentially free energy.

The drive train normally would involve a reduction gear or, in the case of small horsepower, a V-belt drive. A torsional analysis of the drive train is required, along with the proper coupling selection.

Environmental Considerations

The Federal Clean Air Act of 1963 and subsequent amendments provided funds to develop, establish, and maintain air-pollution control programs for state and local agencies. Each state has subsequently taken the basic federal guidelines and developed more in-depth, detailed requirements that suit its local population and environmental considerations. Most states have adopted laws that limit the maximum NOX and CO emissions to 250 tons per year per plant site. As previously mentioned, each state has its own regulations, and each individual application must be reviewed for conformity to current regulations.

In the oil field, two methods have been developed to deal with the emission control requirements for engine exhaust systems. One method is to use catalytic converters, which use proprietary precious metal combinations dispersed over a metal or ceramic subbase. When the engine exhaust gas is passed over the converter, it assists in converting these gases to acceptable alternatives and reduces the NOX in particular.

Some manufacturers use a pelleted catalyst instead of the metal or ceramic subbase. These catalytic converters generally tend to achieve 90% reduction of NOX, CO, and HC, with the most emphasis put on NOX. Several companies manufacture catalytic converters. Some of these companies can furnish specialized catalysts designed specifically to meet specified emission requirements and codes.

The second, and preferred method, is to utilize an engine with a modern-day, clean-burn combustion system that burns the fuel cleanly and emits very low levels of NOX, CO, and HC.

INSTALLATION AND CARE OF STATIONARY RECIPROCATING COMPRESSORS

Many installation requirements of positive-displacement gas compressors are the same as those for other positive-displacement compressors. To avoid repetition, the reader is referred to the section on installation and care of stationary, reciprocating compressors in Chapter 2. Included there are discussions of compressor location, foundations, subsoil characteristics, concrete, grouting, foundation bolts, alignment, leveling, and other topics.

The following further discussions include material that applies specifically to gas compressors, as well as some more general material discussed here for convenience.

Skid-mounted Units

The selection and preparation of a site for the installation of a skid-mounted unit should be made prior to the arrival of the unit.

1. The site should be selected to:
    a. Isolate the package from hazardous areas.
    b. Minimize the piping system.
    c. Allow space for transport truck, service equipment, and personnel.
    d. Keep sufficient distance from areas that are potential problems regarding noise and exhaust emissions.
    e. Provide adequate drainage.
    f. Optimize cooling.
2. The pad area should be solid, reasonably smooth, and level. Uniform contact between the support material and the underside of the skid is the most important consideration to prevent high centering (teetering) and resultant vibration problems. The manufacturer should be contacted for recommendations for the specific location being considered.

Platforms

Offshore platform design constraints must take into consideration the following: water depth, deck height above mean water level, wave conditions (height, speed, direction, and frequency), wind speed and direction, platform load, unbalanced forces, couples and vibrations transmitted by equipment to the platform, natural frequency of the structure, and sea bottom conditions (mud, gravel, rock, etc.). Advice of a consultant versed in platform design is recommended.

Site Considerations

Offshore. The location of a platform must take into consideration the following: water depth, sea bottom conditions, proximity to undersea facilities (wells, pipelines, control modules, etc.), as well as sea and wind conditions. The employment of a consultant familiar with code requirements, platform design, and local conditions is recommended.

On shore. Site considerations include proximity to the facility served (pipeline, refinery, industrial plant, etc.), soil conditions, availability of utilities, elevation, ambient temperature range, neighbors, noise levels, exhaust emissions, and ambient conditions (desert, swamp, woodlands, industrial park, etc.).

Packaged compressor installation guides. Portable packaged reciprocating compressors are highly popular in oil and gas field operations. The following suggestions may be helpful to the engineer planning or supervising the installation of a packaged unit (Figs. 7.98 and 7.99). These installation comments are not intended to replace specific requirements of the package or compressor manufacturer nor to supersede piping codes, safety regulations, and prevailing laws and ordinances that may be applicable to a particular installation.

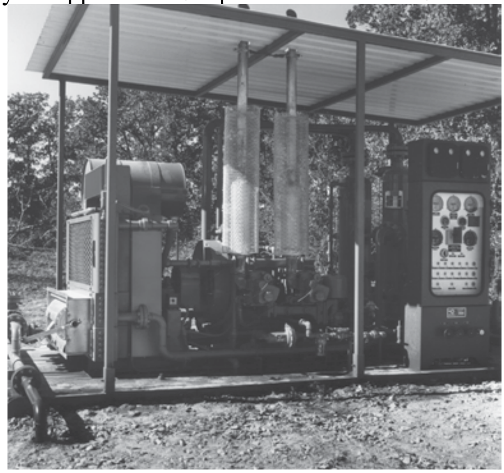

Figure 7.98 Field installation, 32 Bhp, 1400 rpm integral engine-compressor in casinghead gas service.

Figure 7.99 400 Bhp, 1400 rpm, balanced-opposed three-stage compressor being trucked to dock for offshore platform casinghead gas service.

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Selection and site preparation for packaged compressors. Prior to arrival of the compressor at the job site, certain preparations should be made.

Site. The compressor site should be selected in much the same way as for skid-mounted units. However, since the requirements are not completely identical, a separate list is given for convenience. The site should:

1. Isolate the package from hazardous areas, such as storage tanks, open flames, and vents.
2. Minimize length of piping runs to be made.
3. Allow space to maneuver the truck that will transport the unit to the site and set it on location. Other space considerations would be work space for mechanics and access for trucks that will deliver bulk oil.
4. Keep sufficient distance from area homes or facilities to minimize potential noise problems or safety hazards. (*Note:* Special silencers are available for restricted areas.)
5. Provide highest possible elevation to allow adequate drainage in bad weather.
6. Allow for maximum exposure to prevailing winds in warm climates, but shelter the unit in cold areas. In warm climates, face the cooler into the prevailing winds.

Responsibility. The compressor package can be handled in a conventional manner by oil field trucks. In most cases, transportation is the responsibility of the end user.

Arrangements. It is normally possible for the delivery trucker to set packaged compressors without additional equipment if the work area is of sufficient size. When ordering a truck, accurate compressor weight must be given and self-loading and unloading capability specified. If this is not done, additional costs for a crane or other trucks for the setting of the unit could be incurred. *Not all truckers have oil field winches; you must specify.*

Large Unit Requirements. All but the largest packages are shipped as a single haul. Very large units require two trucks and also require field installation and handling of coolers.

Setting. Once on location, the trucker will normally tailboard the unit on his trailer, depending on size and location, and set the unit where indicated.

If a site is dirt, sand, or other material that is not too hard, the trucker can back up on the pad area, lower one end of the compressor skid to the ground, and drag a flat surface on the soil. The unit will be set on this surface. In some cases, the setting and positioning of the unit and the weight of the truck cause ruts or depressions in the pad area. Once set down, the unit may be lifted slightly in order to replace compacted soil to provide an even surface. Again, an even contact with the underside of the skid is all-important. The package manufacturer can provide information on proper handling if the installation requires a crane.

The following points must be checked before the delivery truck leaves the site:

- Is the unit set in the proper direction according to climatic conditions?
- Is the surface underneath the skid smooth and free of voids?
- Has everything been unloaded from the truck? This includes parts, parts books, manuals, mufflers, and so on.
- Has any damage to the unit been noted on the bill of lading and acknowledged with the trucker's signature?

Connection. When planning suction, discharge, and fuel gas lines, routes should be considered that will minimize the obstruction of the work area around the compressor. Buried lines reduce personal injury hazards and allow better access to the unit.

Suction and Discharge Lines. Most companies run welded suction and discharge lines. In any case, line pipe must be suited to the pressure requirements.

Most units are generally furnished without suction and discharge companion flanges, but with studs, nuts, and gaskets. Flange sizes are noted on the specification sheet for each unit. It is important that piping is properly braced to avoid hanging excessive weight from these connections.

Each suction line requires a full opening block valve. It should be installed as close to the inlet flange as possible. The discharge line requires two valves. The first valve downstream from, and as close as possible to, the discharge flange is a full-opening check valve. A full-opening block valve should be installed downstream of the check valve. Positioning in this sequence permits easier removal of the check valve for repair, if required.

General Hints

1. Piping support: Be sure off-skid piping is adequately supported to minimize vibration and fatigue.
2. Spacers: Full-flow paddle spacers are generally installed between cylinder and scrubber flanges to permit installation of orifice plates, if required.
3. Pipe: The use of schedule 80 pipe and forged fittings in all screwed pipe assures that lightweight material will not find its way into hazardous service. This is a reasonably inexpensive safety precaution. The thickness of pipe must be specified by calculation, of course, but must be checked by actual measurement.
4. Glycol units: Glycol units installed directly downstream from the compressor are subject to some oil contamination and a 20°F temperature range above ambient.
5. Meters: Meters located close to compressors are susceptible to vibration or pulsation. Reasonable distances and proper bracing are recommended.

6. Scrubbers: Compressor skid-mounted scrubbers are not adequate for proper cleanup of extremely wet or contaminated field gas. Good separation prior to compression is very desirable and usually results in less expense. On-line time is drastically affected. A clean, dry stream will provide the most efficient operation.
7. Pressure controllers: Surges in wells can cause operation problems with compressors. Suction pressure controllers are available in various degrees of sensitivity. If a pressure controller is needed, it is important to get a good one with more sensitivity than actually needed. Slow response time is often the same as none at all if the compressor overloads and shuts down.
8. Bypass valves: Automatic bypass valves are available to maintain compressor operation when gas flow from the well is restricted below the capacity of the compressor. Without an automatic bypass valve, a sharp pressure decrease would shut the unit down due to low suction pressure. An automatic bypass valve can also be used on the discharge line to switch the unit to bypass operation if discharge pressure becomes too high. If volume changes are abrupt, the automatic bypass valve must be very sensitive. A less sensitive valve can be employed for gradual changes. Again, a better controller can justify the cost difference in increased on-line time.
9. Start-up screens: Start-up screens are sometimes placed at the suction flange of a compressor to catch welding slag, rock, and the like, which might be left in the flange line. Their purpose is to protect the compressor from damage. However, experience has shown that these screens are often forgotten and left in the line. In time, the gas flow can erode the screen, break it up, and send it into the unit. An alternative is to adequately purge all lines prior to final connection to the compressor and omit the screen.
10. Makeup oil supply: Because the unit can run only a very short time without makeup oil to the lubricators, the oil supply tank should be filled prior to startup. It is critical that the oil type and viscosity and other specifications be approved by the manufacturer. Additionally, it is very important to remove all contaminants from the tank, lines, and skid before allowing oil to flow into the compressor. These areas are often purged with air or gas. Some oil should be allowed to drain from the tank through the connection line. This initial oil should be collected in a receptacle and then discarded prior to final connection to the skid.
11. Coolant: If your compressor arrives without coolant, the initial fill should be made with water. This allows any leaks to be repaired without waste of more expensive coolants. As soon as possible after the start-up and the correction of any attending problems, the water should be drained and replaced with an antifreeze solution.

The use of one of the commonly available, premixed coolants available in 55-gal drums is strongly recommended. More specific recommendations can be furnished by the compressor manufacturer. These products are used full strength from the drum. The initial expense is offset by effectiveness of the product. Excess is then stored near the unit and later additions are pure, proper strength solutions, thereby eliminating later addition of "ditch water" or even salt water and other contaminants and minerals. Longer component life and less corrosion are the benefits, resulting in lower costs and more on-line time.

Starting a New Reciprocating Compressor

The following rules should always be observed in starting up a new reciprocating compressor in addition to those specified by the manufacturer in the operator's manual.

Check over the compressor to ensure that all parts are assembled properly and securely tightened. Remove the crankcase covers and clean out the interior thoroughly, being sure that the oil strainer is clean. Check the main and connecting rod bearing cap nuts to see if they are tight and the locking devices are in place. Bar the compressor over at least two complete revolutions to make certain that the moving parts are free from interferences. Watch the crankshaft to see that all the connecting rod bearings run true and free. Remove the lubricating oil filter cover, if an oil filter is provided, and clean out the inside and the filter elements and then reassemble. Fill the crankcase with the proper grade of lubricating oil to the required level. Liberally oil the running gear in the frame. On units equipped with pressure oiling systems for the frame running gear, the oil pump should be primed.

Fill the cylinder lubricator with the proper grade of compressor cylinder oil and open all feeds on the lubricator to permit maximum feed. Disconnect the lubricator oil lines at the check valves on the cylinders and operate the lubricator, filling the oil lines until the oil drips from them at the point where the lines are disconnected. Reconnect these lines and operate the lubricator a minute or so longer to inject oil into the cylinders to ensure lubricating oil will be present immediately upon starting the compressor.

Metallic packing of the segmental type should be installed in the packing gland, as follows: The packing is made up of numerous sections or cups, each cup containing two segmental rings. Each ring is divided into segments held to the rod by a spiral spring around the periphery of the packing rings. The parts of the rings and sections of the packing are numbered and lettered. The rings are installed on the rod by placing the spiral springs around the rod, then the segments under the springs.

Each pair of rings must be inserted in its proper packing section or cup. The lettered side of the rings is centered between any of the spaces in the straight cut rings or in a hole drilled for it in its companion ring; otherwise, rings will not enter their case or section. The segments of each ring are numbered 1-2, 2-3, 3-l, and rings must be assembled so that the numbers mate accordingly (i.e., 1 to 1, 2 to 2, 3 to 3). The rings are lettered A, B, C, D, and so on. The A rings should be placed nearest the cylinder bore, followed in sequence by rings bearing consecutive letters of the alphabet.

Check to ensure that the suction line leading to the cylinders is thoroughly cleaned, free from rust, pipe scale, welding shots, sand, water, or condensate and other foreign material that will damage the compressor cylinder if allowed to enter. Also, see that the suction and discharge lines are unobstructed and that all valves therein are properly set. On multi-stage units, it may be advisable, if the unit has been in storage or subjected to questionable handling during the shipment, to remove the tube bundles from the intercoolers to make certain that the shell and tubes are thoroughly cleaned and free from foreign material. The suction ports of the cylinder should also be thoroughly cleaned, removing any material that may have accumulated therein during the erecting of the unit.

Check the valves in the cylinders to be sure that the valves are placed properly in the cylinder ports, suction, and discharge, and securely tightened in place. Considerable loss of capacity and possible serious damage may result if the valves are placed in the incorrect ports and are not seated and held solidly in the valve ports.

Before starting the compressor, turn on the cooling water to the cylinders and to the coolers used with the compressor. Check to ensure that the cooling system is filled and that flow is indicated at the outlets. Adjustment of the amount of cooling should be made after the compressor has warmed up. In areas of hard water, the lining (or scaling) temperature of the water should be determined and, if feasible, the discharge temperature of the cooling water should be kept below this temperature to prevent scale buildup.

On a motor-driven unit, push the starting button and immediately push the stop button to observe the direction of rotation of the unit, thus making sure that the unit will run in the correct direction of rotation as indicated by the manufacturer.

After the correct rotation is established (motor driven or otherwise), the unit should be started and run for a period of several minutes. As soon as the unit is started, note immediately that the lubricator is feeding properly and that all parts are being lubricated. Also note that the unit operates without any noise or knocks. After this short run, stop the unit and feel all bearings to make sure that there is no tendency of parts to heat too rapidly and that the lubrication to all parts is adequate.

Start the unit again and allow it to run for a longer period. Repeat these run-in periods, increasing their duration until a continuous run-in of at least 2 hours is obtained without any overheating or knocking of the running gear, and it is observed that all parts are operating and wearing in properly. The pressure drop across the filter or strainer, or both, should be monitored, with corrective action being taken when the drop is excessive.

At this point, consideration should again be given to the cleanliness of the suction line. This is an extremely vital consideration, particularly in process compressors where the suction lines are usually long and the gas to be compressed is drawn from a closed system, from gas-generating or process equipment, towers, heat exchangers, formers, and the like. Suction equipment to clean and dry the gas to be compressed should be used before the gas is drawn into the compressor cylinders. Although every precaution has been taken in the thorough cleaning of the suction piping and vessels therein during the installation and connecting up of such equipment, foreign material may have entered these lines during installation. It cannot be stressed too strongly that if such foreign material is drawn into the compressor cylinders, excessive wear and failure of elements of the cylinder such as valves, pistons, and piston rings, and severe scoring of the cylinder bore may result.

Several methods are used in starting up a unit to prevent foreign material from entering the compressor cylinders. One method, although considered not fully effective, is to blow out the suction pipe line and equipment therein. If such a procedure is followed and a considerable amount of foreign material is blown from the line, it may be an indication that all may not be removed by this method. To ensure complete removal and cleanliness, sections of the suction lines and equipment therein should be dismantled and again cleaned thoroughly.

Another method, usually very effective, is to install a reinforced conical fine screen with sufficient flow area in the suction line as close as possible to the compressor cylinder suction flange. The screen must be readily accessible for periodic removal, cleaning, and reinstallation. The use of this screen in the suction line is suggested also even when the blow-back method is used. It is recommended that a pair of pressure gauges or a manometer be connected on either side of the screen for the purpose of checking the pressure drop through the screen. When the drop becomes excessive, the screen should be removed, cleaned, and replaced.

On multi-stage units, particularly where the gas compressed flows between stages through long fabricated lines, coolers, separators, and the like, similar screens should be installed at the suction flange of all stage cylinders. Generally, on a two-stage air compressor, the suggested methods apply to the first-stage suction only.

The valves, which have been left out during the idle run-in period, should next be installed. The unit should then be run with the suction being taken from the atmosphere with a free discharge. It may be necessary on a gas compressor to disconnect the suction and discharge lines temporarily from the system during the run-in period. Such an open suction line should be screened temporarily to prevent foreign material from being drawn in. The compressor should be run in this manner for about 1/2 hour, after which the unit should be brought up to full load by gradually increasing pressure in increments over a period of 4 to 8 hours. It is recommended during the break-in run (except on non-lubricated compressors) that oil be applied to the piston rod to facilitate the running in of the packing. This is especially true when metallic packing is used.

The additional time taken to run-in the unit to ensure that it is operating properly will be compensated for by the increased satisfaction resulting in the subsequent performance of the unit. In the running-in of an engine or turbine-driven compressor, the same procedure as outlined for a motor-driven compressor should

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be followed. However, in this case, the advantage of the variable-speed feature should be utilized by running the unit at the start at slow speeds and gradually building up to full-rated speed as the pressure and load are built up.

Regular Inspection

The manufacturer's manual should be used and followed for maintenance. Regular inspection should be made at definite intervals, at which time any necessary corrections may be made, such as replacement of worn parts or packing, adjustment for wear of working parts, and cleaning of valves and crankcases. Particular attention on air compressors should be paid to the suction filter to make certain that it is clean and unclogged at all times. The frequency of regular inspections will depend on the conditions prevailing at the installation. It is recommended that the unit be checked very frequently during the first few weeks of operation, and extension of the periods between inspection be dependent on the experiences observed during the earlier periods of operation.

The presence of any deposits on the valves indicates either that the intake is dirty or that too much oil or unsuitable oil is being used or that there are leaking cylinder valves or valve gaskets. All ports and passages should be examined and any obstructions, such as carbon and sticky oil, removed.

Oil used in the crankcase of a reciprocating compressor must be changed at intervals. Here, again, the frequency of changes will depend on the actual conditions prevailing, but the oil should be changed at least once a year. Before refilling, the crankcase should be thoroughly cleaned and care taken not to leave shreds of lint that would obstruct oil passages. It is inadvisable, in any case, to use waste for wiping out the crankcase, and gasoline or other flammable liquids should never be used for washing it.

Water jackets should be inspected and washed out as frequently as water conditions may require. On multi-stage units, particular attention should be given to the intercooler water surfaces, which must be kept as clean as possible in order to keep the compressor operating at peak economy. It must be remembered that inefficient intercooling will result in an increase in the horsepower required to drive the com-pressor and, consequently, an uneconomical unit.

It is recommended that the inlet and discharge line from each cylinder be equipped with a thermometer. The normal operating temperature should be observed during the early operating period of the unit. Any increase in the normal discharge temperature from a cylinder can indicate the presence of worn valves, incorrect speed, defective capacity controls, inadequate cooling-water quantity, excessive cooling-water temperature, excessive discharge pressure, inadequate cylinder lubrication, worn piston rings, or scored cylinders.

It is recommended that an operator's log be used to record the operation of the compressor. The log should show the operating conditions such as temperature, interstage pressures, and so on, at regular intervals and should include all normal maintenance performed on the machine, such as oil changes and oil added. The log should also contain a report of any unusual conditions or events such as power or water failure. A good oil analysis program is helpful in indentifying problem areas. Oils with viscosity and flash point on the high side are preferable for compressor cylinders working under high temperature conditions (i.e., single-stage machines with compression ratio of 7 or higher) and for cylinders handling refinery gas (i.e., methane and the like).

The preceding general comments will be applicable to most air compressor installations. For installations that involve gases, very high ambient temperatures, or high cooling-water temperatures or for those units operating at high discharge pressures and temperatures, it is advisable to consult the compressor manufacturer for recommended oil specifications.

Rate of Oil Feed for Compressor Cylinders

The amount of oil required will vary somewhat with the type of machine, the local conditions of operation, and the compressor service as well as the gas handled. In general, the best way to determine the maximum amount of lubrication to be fed is to remove valves from the cylinders periodically and examine the bores to determine the amount of oil present. A film of oil should be felt upon all parts, but there should be no excess oil present. If the elements feel dry, the feed should be increased. If oil lies in the bore with excessive quantities in the discharge ports, the feed should be reduced. This periodic examination should determine the final amount to be fed. Although it is difficult to predict the exact amount of lubrication necessary for all conditions, it has been observed that the rates will fall generally within the limits shown in Tables 2.8 and 2.9. The feeds given, the total to the cylinder bore and packing, are based on empirical formulas and may be varied to suit the particular conditions of service of the compressor cylinders and the gas compressed. The figures given are the suggested feeds when clean and dry conditions prevail in the compressor cylinder. Wet and dirty conditions of the air or gas compressed may require increased feeds as conditions may indicate.

In starting a new compressor, the oil to the various cylinders should be fed in liberal quantities, approximately twice the amount given in the tables. After a glazed surface is formed on the cylinder bore, the amount can be gradually cut down to that required, as determined by periodic examination.

Oil fed to the piston-rod metallic packing will also depend on the condition of the air or gas compressed and may vary from service to service. Under normal, clean, and dry conditions, two to three drops of oil per feed per minute should be satisfactory. Periodic examination of the piston rod to ensure that an oil film is present will indicate whether the oil feed is correct.

Frame and Bearing Lubrication

Where the same oil is used to lubricate both the compressor cylinder and the running gear, it must be selected primarily to suit the requirements for the compressor cylinder, except that consideration should be given to the ambient temperature in which the compressor is required to operate. If freezing ambient temperatures are experienced, a pour test of the oil is of great importance.

Selection of oil for the running gear depends on:

1. Size and speed of compressor.
2. Type of lubrication system used:
   a. Splash system, where some parts of running gear dip into the oil.
   b. Flood type, where an oil pump circulates the lubricating medium, flooding the bearings with oil.
   c. Pressure type, where an oil pump forces the lubricating medium under pressure into the bearings.

Oil used for frame and bearings only should be selected in accordance with specifications in Fig. 7.73. Because of wide variations in construction and in lubrication systems used with compressors, the manufacturer's recommendations are of great importance and should be closely followed.

Synthetic Lubricants

Several synthetic or fire-resistant lubricants are available. These lubricants are primarily intended for cylinder lubrication and are not normally needed for the crank-case because of the relatively slight hazard in the crankcase. Before these lubricants are used, it is recommended that their use be discussed with both the manufacturer of the lubricant and the manufacturer of the compressor.

The manufacturer of the lubricant should be familiar with the conditions of service, including such items as the gas being handled, the ratio of compression, moisture content of the gas handled, and the operating temperatures, as well as the intended use of the compressed air or gas. The compressor manufacturer should be consulted to make certain that the materials of construction as well as the lubricating system are suitable for using synthetic fluids. Because of the high specific gravity, the usual sight-glass fluids (diluted glycerine, salt solutions, etc.) cannot be used. The lubricants will sometimes soften and lift many common paints; therefore, any painting on surfaces that may come into contact with these lubricants should be done with a special paint having adequate resistance to the synthetic fluid. In general, it is recommended that no internal painting be done.

Synthetic lubricants tend to swell many types of rubber, including neoprenes and buna N rubber. It is necessary, therefore, to make certain that packings and gaskets are made of a material not affected by synthetic fluids.

The rate of feed of synthetic lubricant must be watched, just as with petroleum lubricants, to make certain that the cylinders and valves are properly lubricated. Experience has shown that the feed rate for synthetic lubricants should be increased to at least 1.5 times the petroleum lube rate.

Synthetic lubricants are not as compatible with water as are compounded petroleum oils and, therefore, efficient separation after any cooler must be exercised. In addition, cylinder jacket temperatures should be maintained relatively high at all times.

The synthetic fluids are classified as essentially nontoxic. It is recommended, however, that repeated or prolonged contact with the skin be avoided. Similarly, precautions must be taken to avoid oral ingestion of the fluids or accidental contact with the eyes.

Preventive Maintenance

Routine preventive maintenance is essential to the safe and efficient operation of the unit. The purpose of this guide is to provide suggestions concerning the most basic facets of a routine maintenance program and, consequently, to aid you in obtaining maximum on-line time. The most important point in any maintenance program is that it be carried out regularly and completely.

Accurate records of operating conditions are an important part of a good preventive-maintenance program. All readings should be recorded daily and compared frequently. This enables the operator to note changes that might indicate internal problems and can also be useful in developing a maintenance schedule.

Experience indicates that good operating techniques and an efficient inspection schedule will result in longer life of the engine or compressor, or both, and fewer periods of shutdown for repairs. The inspection and operating suggestions in this section have been found to be applicable at practically every installation. Careful observation of the unit during its initial period of operation will probably indicate other inspections to be made periodically. It is always recommended that one refer to the manufacturer's manual for detailed instructions.

Operating Routine

The manufacturer's manual should be followed at all times. The inspection and routine maintenance schedule listed next gives the minimum recommended time intervals between inspections. Many of the time intervals can be extended beyond those listed for a particular unit, depending on operating conditions and other related factors.

EVERY DAY

1. Inspect the engine's fuel gas scrubber to see that it is functioning properly.

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2. Check the crankcase oil level; it should be at the full mark on the oil level gage. Add oil if necessary.
3. Fill each lubricator compartment with the proper grade of lubricating oil. This should be done each shift or every 8 hours.
4. If appropriate, examine the fuel-injection valves. If solid-type push rods are used, check the tappet clearances periodically. Check the clearance against those recorded after the cylinders were balanced.
5. Check both engine and turbine speed periodically.
6. Check lube oil pressure periodically. See manufacturer's operation data for correct value.
7. Turn the handle on the oil filter two or three complete revolutions every 8 hours if a scraper type oil filter is supplied.
8. Compressor suction and discharge pressures and temperatures should be checked periodically. The operator should occasionally measure the compressor cylinder valve cover temperature; an increase in this temperature usually indicates a leaky valve.
9. If compressor is multi-stage, check the suction and discharge pressures and temperatures of all stages as well as all the compressor valve cover temperatures occasionally. An increase in these temperatures usually indicates a valve leak.
10. Check the temperature of the lubricating oil entering the cooler. See manufacturer's operation data for correct value.
11. Check exhaust temperatures periodically. Compare them with temperatures that were previously recorded. Higher exhaust temperature in a cylinder usually indicates carbon deposits in the cylinder parts.
12. Check cooling-water flows and temperatures. See manufacturer's operation data for correct value.
13. Check traps or separators, or both, to be sure that they are functioning.

EVERY MONTH

1. Remove and clean spark plugs if necessary. Check the gap clearance and reset as required.
2. Check out safety shutdown switches and controls to be sure they are functioning correctly.
3. Check compression pressures. Low compression can be caused by sticking piston rings.
4. Inspect and service your particular ignition device as described in the manufacturer's instructions.
5. Check lubricator reduction gears; oil if necessary.
6. On motor-driven units, inspect the brush holders. They should move freely; clean if necessary.
7. Change oil and filter in small engines (see manufacturer's recommendations).

EVERY THREE MONTHS

1. On large engines, examine air starting valves and air check valves for wear. Check to see that they are operating correctly. In low altitudes where humidity is high and the starting air is likely to be moist, more frequent checking is advisable.
2. Inspect the fuel-injection valves and seats. If the valves are leaking due to worn valves or seats, it will be necessary to grind the valves and replace the seats. Correct seating of the valves and properly adjusted tappet clearances are necessary to keep the power cylinders balanced.
3. Clean the turbocharger blower wheel on engines so equipped. The time between cleanings will be different for each installation since the amount of dirt, soot, and oil vapors entering the engine air filter will vary.

EVERY SIX MONTHS

1. On large engines, examine the internal drive chains for correct tension. If there is excessive slack in the chains due to wear, adjust the tension as required.
2. Check the ignition device for signs of wear.
3. On large motor-driven compressor units, remove the oil filter cover or housing and clean the filter elements and case.

EVERY YEAR

1. Inspect heat exchangers for fouling or leakage, or both.
2. Examine the compressor cylinders as follows:
   a. Check piston rings and cylinder bore for excessive wear.
   b. Inspect the pressure and oil wiper piston rod packings; clean or replace as required.
   c. Check piston rings for wear and damage.
   d. Examine the valve seats, discs, springs, and stop plates; replace worn or damaged parts.
   e. Examine water jackets for scale and corrosion; clean if necessary.
   f. Replace any damaged gaskets.
3. Check unloaders and unloader controls, if used, for correct operation.
4. Engine and compressor analysis and maintenance equipment is available that can assist the operator to determine what part of the unitneeds maintenance, minimizing costly down time.

EVERY THREE YEARS

1. Check all the main bearings and the crankshaft for correct alignment. If bearing clearances are excessive, replace the bearing liners.

OTHER PERIODIC CHECK-UPS

1. Examine the compressor valves. The frequency of inspection depends on the type of compressor service. Keeping the valve seats in good condition is important. Reface the seats when necessary, making sure that the proper lift is maintained. Replace cracked, broken, or warped discs. Check the valve springs against a new one; they must have the same tension.
2. Check the foundation bolts for tightness.
3. Check the gas regulators in the engine's fuel gas line to see that they are maintaining correct gas pressure. The required gas pressure will vary with the Btu value of the gas.
4. Clean the lubricator reservoir with a suitable solvent as required.
5. Check the accuracy of the compressor gages. If service is severe, a check should be made each week.
6. Examine the air intake filter for excessive dirt and other foreign materials; clean when necessary Excessive pressure drop across the filter is evidence that cleaning is necessary.
7. Drain any condensation from the starting air system. The frequencyof draining will depend on the location of the installation.
8. Check the cooling water if it has been treated for use in a closed cooling system. Balance the concentration of treating chemicals as required.

OPERATING HINTS

1. Stop oil leaks as quickly as possible. The leaking oil may run down the foundation, seep under the grout, and destroy the grouting job.
2. Do not put a heavy load on a cold engine. Allow it to warm up first.
3. Never admit cool water to a hot engine. If you have forgotten to turn on the cooling water when starting up, shut down the engine and let it cool off.
4. Never attempt to start the engine without first unloading the compressor cylinders.
5. Check temperatures and pressures after the engine has been started. Do not permit an engine to continue to run if no reading is shown on the oil-pressure gage or if the pointer on the gage surges back and forth. Shut the engine down and investigate the problem.
6. Be sure the load is balanced on all the cylinders at all times.
7. Should the lubricator reservoir be allowed to run dry, it will probably be necessary to prime each pump again.

8. Break in new packing gradually, especially high-pressure packing.
9. Keep the scavenging air filter clean at all times. Low scavenging air pressure and low turbocharger speed indicate a dirty air filter or a dirty blower wheel. High scavenging air pressure indicates defective impeller blades or retarded spark, overspeeding the turbine.
10. When operating at maximum load, set the fuel supply pressure at a reading as low as possible.
11. Never stop an engine without unloading it and giving it a chance to cool down for at least 5 minutes, except in case of an emergency.

To close the section on reciprocating compressors, although it may be used for other types as well, the following compressor inquiry form, Fig. 7.100, is included. It may be used as a guide to the information that should be submitted with an inquiry about a new compressor.

( ) Any Customer Written Specifications Yes________No________(If ''yes,'' attach).
( ) Description of Application:

( ) Duty Cycle________hrs/day. Average continuous operating time________minutes
( ) Gas Handled____________Clean____________Contaminated with____________
Dry______ Wet______
( ) Gas Analysis:—(If available give ''N'' value, mol. wt., compressibility factor

( ) Any comments on previous experience or preference as to materials in piston rings, piston rod, cylinder liners, type stuffing box, type packing, valve material?

( ) Barometer________psia or Altitude________ft. above sea level
( ) Intake pressure________psig Intake Temp________°F Rel. Humidity________%
Possible variation of intake pressure from________psig to________psig
( ) Disch. Press.________psig. Possible variation from________psig to________psig
( ) Capacity required____________cfm, cfh, cfd measured at________psig at________°F dry or not.
Acceptable variation of capacity from____________cfm to____________cfm
( ) Regulation required to control from________intake pressure________discharge pressure
( ) Automatic start and stop Cut-in________psig Cut-out________psig
( ) Constant speed control Cut-in________psig Cut-out________psig
( ) Type of Drive____________________________
( ) Electrical Conditions____________Steam Conditions____________
( ) Type of Mounting____________________________
( ) Location of Unit: outdoors or indoors; hot or cold ambient; ventilated or nonventilated space.
( ) Cooling Water: Temp________°F. Clean________Dirty________Salt Water________
Fresh Water____________Corrosive?____________
( ) Special accessories required:
Filter______Aftercooler______Receiver______Type Starter________Belt Guards______
( ) Other pertinent information not covered above:

( ) Number of units required____________Shipment needed____________
( ) Only estimating information needed by____________________
( ) Firm quotation needed by____________________

Figure 7.100 Compressor Inquiry Sheet

Dynamic Process Compressors

Dynamic compressors are machines in which air or gas is compressed by the mechanical action of rotating vanes or impellers imparting velocity and pressure to the air or gas. In an axial compressor, as the name implies, flow is in an axial direction. In a centrifugal compressor, flow is in a radial direction.

DEFINITIONS

The following definitions will be helpful in understanding the construction and application of dynamic-type compressors:

Base plate is a metal structure on which the compressor is mounted and often the driver as well.

Blower is a term applied in the past to a compressor in a specific low-pressure application, for example, cupola blower, blast furnace blower.

Capacity is the rated maximum flow through a compressor at its rated inlet and outlet temperature, pressure and humidity. Capacity is often taken to mean the volume flow at standard inlet temperature, pressure and humidity.

Casing is the pressure containing stationary element that encloses the rotor and associated internal components, and it includes integral inlet and discharge connections (nozzles).

Diaphragm is a stationary element between the stages of a multi-stage centrifugal compressor. It may include guide vanes for directing the air or gas to the impeller of the succeeding stage. In conjunction with an adjacent diaphragm, it forms the diffuser surrounding the impeller.

Diaphragm cooling is a method of removing heat from the air or gas by circulation of a coolant in passages built into the diaphragm.

Diffuser is a stationary passage surrounding an impeller in which velocity pressure imparted to the flowing medium by the impeller is converted into static pressure.

Efficiency: Any reference to the efficiency of a dynamic-type compressor must be accompanied by a qualifying statement that identifies the efficiency under consideration. (See Chapter 8 for definitions of adiabatic, polytropic, etc.)

Exhauster is a term sometimes applied to a compressor in which the inlet pressure is less than atmospheric pressure.

Flange connection (inlet or discharge) is a means of connecting the casing to the inlet or discharge piping by means of bolted rims (flanges).

Guide vane is a stationary element that may be adjustable and that directs the flowing medium to the inlet of an impeller.

Impeller is the part of the rotating element that imparts energy to the flowing medium by means of centrifugal force. It consists of a number of blades mounted so as to rotate with the shaft. Impellers may be classified as follows:

a. *Open face* (Fig. 7.101a), without enclosing cover, may be cast in one piece, milled from a solid forging, or built up from castings, forgings, or plates.
b. *Closed type* (Fig. 7.101b), with enclosing cover and hub disk, which may be cast in one piece or built up from castings, forgings, or plates. Blades may be attached to the enclosing cover and hub disk with separate rivets, with rivets machined integral with the blades or by welding.

Impellers are further classified with respect to blade form, as follows:

a. *Radial bladed*, having straight blades extending radially.
b. *Backward bladed*, having straight or curved blades installed at an angle to the radius and away from the direction of rotation.

Inducer is a curved inlet section on an impeller (Fig. 7.101A).

Multi-casing compressor. When two or more compressors are driven by a single motor or turbine, the combined unit is called a multi-casing compressoror compressor train.

Multi-stage axial compressor is a machine having two or more rows of rotating vanes operating in series on a single rotor in a single casing. The casing includes the stationary vanes and the stators for directing the air or gas to each succeeding row of rotating vanes. These stationary vanes or stators can be fixed or variable angle, or a combination of both. This type of machine usually has two bearings, with the driver coupled to the compressor shaft (Fig. 7.102).

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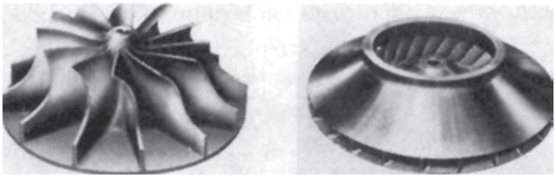

A B

Figure 7.101 (A) An open, radial impeller. (B) A closed backward blade impeller.

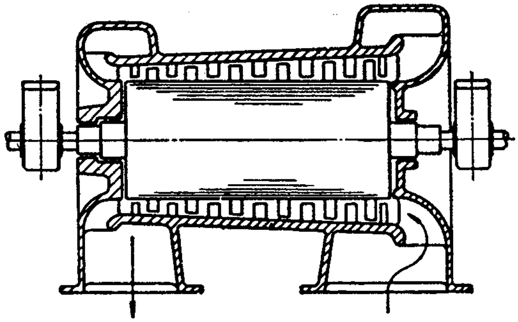

Figure 7.102 Multi-stage Single-flow Axial Compressor

Multi-stage centrifugal compressor is a machine having two or more stages. Such compressors may be described as:

a. *In-line:* All impellers are on a single shaft and in a single casing. A single-flow compressor of this type is seen in Fig. 7.103. A double-flow unit is seen in Fig. 7.104. Units may be further identified as internally or externally cooled, the latter shown in Figure 7.105.
b. *Integrally geared:* These units have bull gear drive with one or more pinions. Impellers are mounted singly at one or both ends of each pinion, and each impeller has its own separate casing. Normally used only on air and nitrogen service, these machines usually have provision for external cooling between stages. On small plant-air units, coolers and controls are often packaged on the same base as the compressor. These are described in Chapter 2.

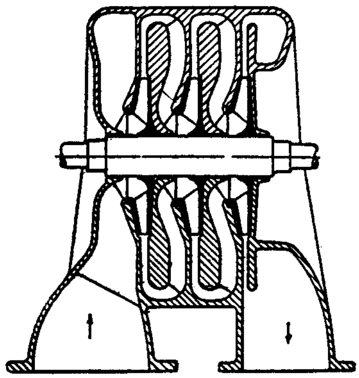

Figure 7.103 Multi-stage Single-flow Centrifugal Compressor

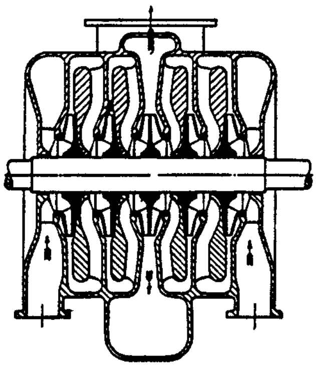

Figure 7.104 Multi-stage Double-flow Centrifugal Compressor

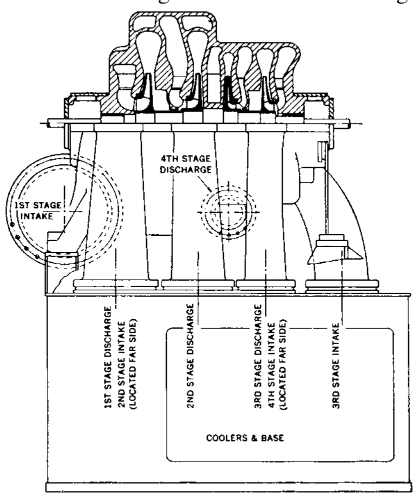

Figure 7.105 Multi-stage Single-flow Externally Cooled Centrifugal Compressor

Overhung-type centrifugal compressor denotes a single-inlet compressor with the impeller mounted on an extended shaft of the driver (i.e., one in which the compressor has no shaft of its own); for example, an extended pinion gear shaft.

Pedestal-type centrifugal compressor denotes a single-inlet compressor with the impeller mounted on a shaft supported by two bearings in a pedestal, with the driver coupled to the compressor shaft.

Performance curve is a plot of expected operating characteristics (e.g., discharge pressure versus inlet volume flow, or shaft horsepower versus inlet volume flow; see Fig. 7.106).

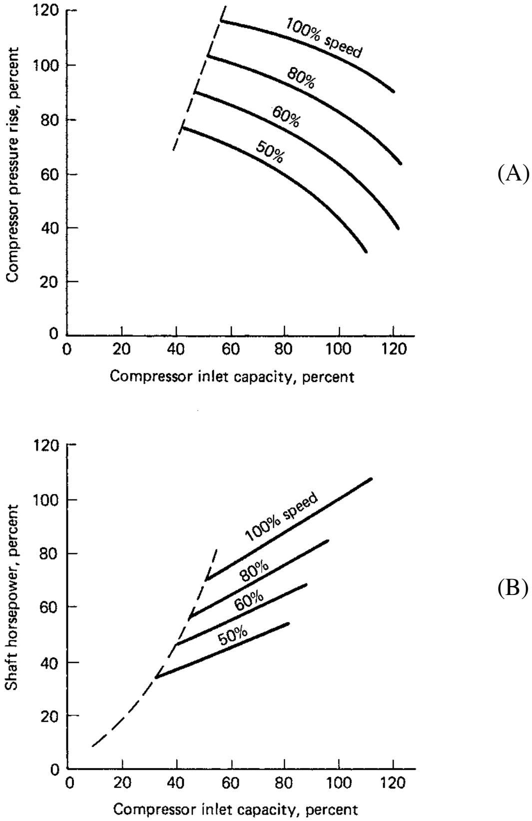

Figure 7.106 (A) Typical performance curves for a centrifugal compressor, either single-stage or multi-stage; (B) comparable performance curves for an axial compressor.

Rotor is the rotating element and is composed of the impeller or impellers and shaft and may include shaft sleeves, thrust bearing collar; and a thrust balancing device.

Seals are devices used between rotating and stationary parts to separate and minimize leakage between areas of unequal pressures. Basic types include clearance-type metallic labyrinths: single and multiple, injection-type labyrinths, eductor-type labyrinths, or a combination. Dry contact seals include carbon ring and synthetic materials such as Teflon. Liquid-injection types use water or oil seals.

Shaft is that part of the rotating element on which the rotating parts are mounted and by means of which energy is transmitted from the prime mover.

Shaft sleeves are devices that may be used to position the impeller or to protect the shaft.

Single-stage centrifugal compressors are machines having only one impeller. They may be classified as follows:

a. Single-flow (Fig. 7.107)
b. Double-flow (Fig. 7.108)

Sole plate is a metal pad, usually embedded in concrete, on which the compressor feet are mounted.

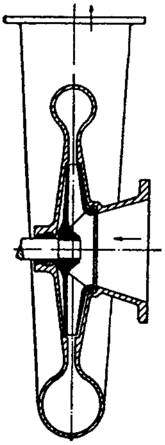

Figure 7.107 Single-stage, Single-flow Centrifugal Compressor

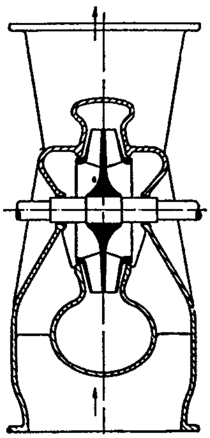

Figure 7.108 Single-stage, Double-flow Centrifugal Compressor

Stability or percentage stability is 100 minus the surge limit at rated discharge pressure, where the surge limit is expressed in percentage of rated capacity.

Surge limit (pulsation point) is the volume flow below which partial or complete cyclic flow reversal occurs, resulting in unstable aerodynamic operation.

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Thrust balancing device (balance piston or drum) is the part of the rotating element that serves to counteract any inherent axial thrust developed by the impeller.

Volute is a stationary, spirally shaped discharge passage that converts velocity head to pressure.

CENTRIFUGAL COMPRESSOR CHARACTERISTICS

Compression of gas by means of a reciprocating compressor is easily pictured and is generally well understood by engineers and operators. The capacity of such a unit at constant speed is essentially constant, and the discharge pressure is that required to meet the load conditions.

A centrifugal compressor, on the other hand, develops its pressure within itself, independent of the load, but the load determines the flow to be handled. This is a generalized statement, of course, limited by the physical size of the unit and the size of the driver.

Both of the preceding statements are made on the assumption of constant speed and no controlling devices. On both centrifugal and reciprocating compressors, it is possible either to vary the speed or to provide integral regulatory means so that any desired pressure or flow requirement may be met, providing it is within the limits of the compressor and its driver (Fig. 7.108).

In its simplest form, a centrifugal compressor is a single-stage, single-flow unit with the impeller overhung on a motor. Such a unit is shown in Fig. 7.109 with a section cut away so that the flow of gas through the unit may be traced. This single-flow unit consists of the inlet nozzle, the impeller, the diffuser, the volute, and the driver. The passage of gas through the unit follows the order above. The gas enters the unit through the inlet nozzle, which is so proportioned that it permits the gas to enter the impeller with a minimum of shock or turbulence. The impeller receives the gas from the inlet nozzle and dynamically compresses it. The impeller also sets the gas in motion and gives it a velocity somewhat less than the tip speed of the impeller.

The diffuser surrounds the impeller and serves to gradually reduce the velocity of the gas leaving the impeller and to convert the velocity energy to a higher pressure level. A volute casing surrounds the diffuser and serves to collect the gas to further reduce the velocity of the gas and to recover additional velocity energy.

The maximum discharge pressure that may be obtained from a single-stage unit is limited by the stresses permissible in the impeller. Where the requirement for pressure exceeds that obtainable from a single-stage compressor, it is possible to build a centrifugal compressor with two or more impellers. This requires a return passage to take the gas leaving each diffuser and deliver it to the inlet of each succeeding stage.

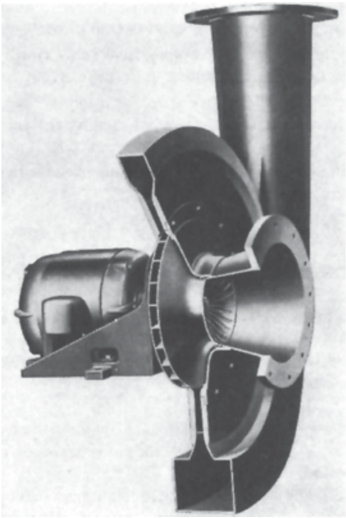

Figure 7.109 Cutaway view of a single-stage, single inlet centrifugal compressor with closed-type impeller.

A typical multi-stage centrifugal compressor is shown in Fig. 7.110. It can be seen that the compression is accomplished by three impellers that act in series and are mounted on the same shaft. Flow of gas between stages is guided by the interstage diaphragms from the discharge of one impeller into the inlet of the next impeller. Sealing between stages is done by the labyrinth rings, which impose restriction on the flow between impellers at the shaft, at the impeller eye, and at the balancing drum.

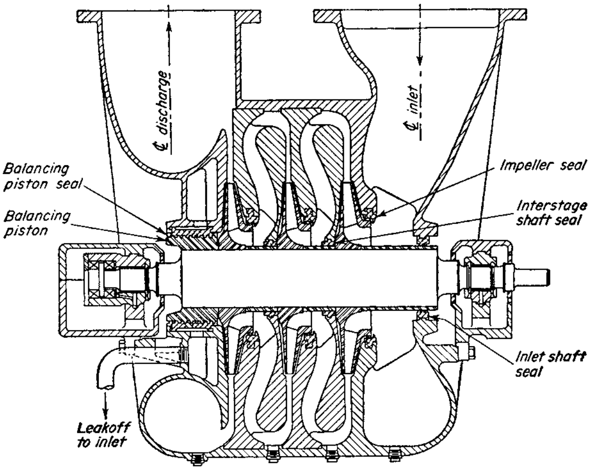

Figure 7.110 Vertical section drawing showing typical multi-stage centrifugal compressor.

Operating Characteristics

A typical set of performance curves for a centrifugal compressor is shown in Fig. 7.111A. Corresponding curves for an axial compressor are shown in Fig. 7.111B.

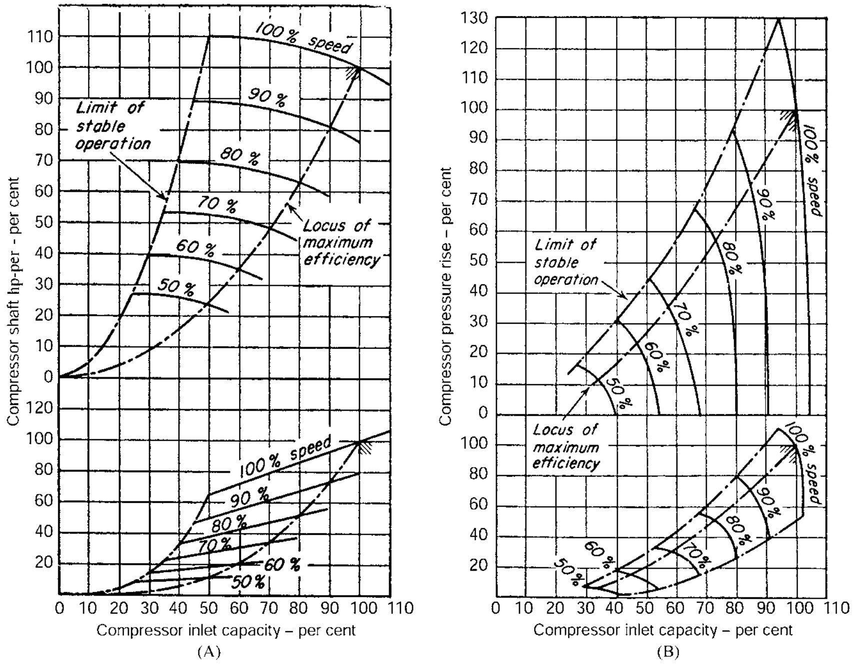

Figure 7.111 (A) Typical performance curves for a centrifugal compressor, either single-stage or multi-stage; (B) comparable performance curves for an axial compressor.

Demand Load

Regardless of the actual service to which a centrifugal unit may be applied, the general nature of the demand load may be divided into three classifications as follows:

1. Frictional resistance.
2. Constant head or pressure.
3. A combination of constant head or pressure plus frictional resistance.

The frictional resistance load is that which would be typically encountered in natural gas transmission. It is the pressure necessary to overcome the frictional resistance of flow through piping or associated equipment.

The fixed-head or pressure load is that which is required to overcome a liquid head or a controlled back pressure. In yeast or sewage agitation, for instance, air is blown into the bottom of a vat or tank, and the level of the liquid is held at a given value. This liquid head presents a fixed pressure regardless of the flow. Also, in certain processes, it is desirable to operate under a fixed pressure that is maintained by some external pressure controller at the exhaust from the process.

A third type of load, which is a combination of the above, is by far the most common. Virtually all loads are, to a certain degree, a combination. An example of this type of load is that of a blast furnace for which the majority of the pressure requirement is to overcome frictional resistance, but, in addition, some is required to maintain a pressure inside the furnace in conjunction with controlling the exhaust from the top of the furnace. Likewise, the above-mentioned yeast or sewage agitation generates to a degree a combination load, since there is some frictional resistance in the pipe between the compressor and the vats, as well as some pressure loss through the nozzles where the air enters the vats.

In Fig. 7.112, the three types of loads are shown graphically. Curve *CD* represents the purely frictional load, curve *AB* represents the fixed head, and curve *AE* represents the combination.

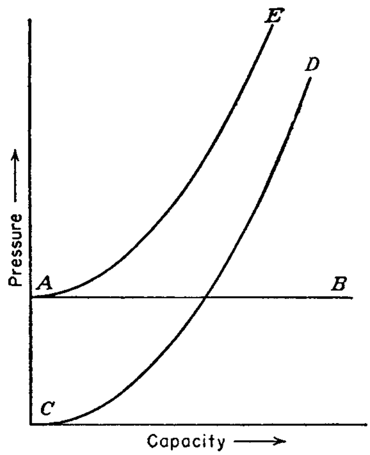

Figure 7.112 Typical curves illustrating three types of compressor loading.

Application to Load

As previously mentioned, a reciprocating compressor is essentially a constant-flow, variable-pressure unit. This is shown in Fig. 7.113 as line *JK*. Actually, because of the decrease in volumetric efficiency at increasing pressures, the reciprocating compressor will have a sloping characteristic, shown by line *JL*. A centrifugal compressor is essentially a variable-flow, constant-pressure unit as indicated by the line *FM*. Because of internal losses, the compressor characteristic is not a straight line but is similar to line *FG*.

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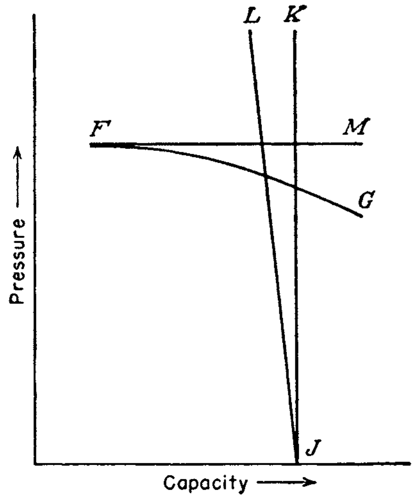

Figure 7.113 Performance characteristics of centrifugal versus reciprocating compressors.

It would be possible to select a centrifugal or a reciprocating compressor for the same flow and pressure as indicated by point *H*, and the characteristic curves are shown in Fig. 7.114 as *FG* for the centrifugal and as *LJ* for the reciprocating units.

To explain the application of a centrifugal unit to any given service, the performance curve, shown by line *FG*, is superimposed on a demand or load curve, *AE*. With this combination of compressor and load, the capacity handled will fall at the intersection of the two curves at point *H*. This is the only point at which the compressor will operate at that given demand requirement.

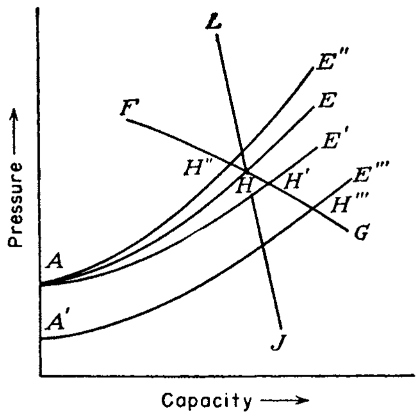

Figure 7.114 Characteristic curves of a centrifugal compressor and a reciprocating compressor superimposed upon demand-load curves.

For many applications, the demand curve will change, and this may result from varying the frictional resistance. This is shown by lines *AE′* and *AE″*, in which case the flow handled by the compressor is at the intersection of these two curves, or point *H″*.

In many processes, it is desirable to maintain a constant flow through the unit or to maintain a constant pressure delivered to the process. This must be done in spite of the fact that the actual demand requirements may vary from *AE* to *AE′*, and it would be desirable to maintain the constant flow corresponding to *H*. This could be done by means of partially closing a valve at the intake or discharge of the compressor so as to give a new demand curve passing through point *H*.

Controlling Pressure or Capacity

The use of a valve as a means of controlling pressure or flow at any given value is the simplest form of control. It is not efficient, however, since this artificially created resistance represents an irrecoverable loss of power. A more efficient way to control the unit for any given pressure or flow is to vary the speed. This creates a family of curves like those shown in Figure 7.115 by curves *FG, F′G′, F″G″*, and so on. By varying the speed, it is possible to set the intersection of the compressor characteristic and the demand curve to any given required pressure or flow within the operating limits of the compressor and driver.

The method of controlling the compressor characteristics, therefore, depends on the type of driver. For steam-turbine drive, the normal method of control would be to vary the speed, permitting efficient operation and a wide range of control. In the case of motor drive, the picture becomes more involved, since the most commonly used motors are essentially constant-speed drivers. For motor drive, the following control possibilities may be considered:

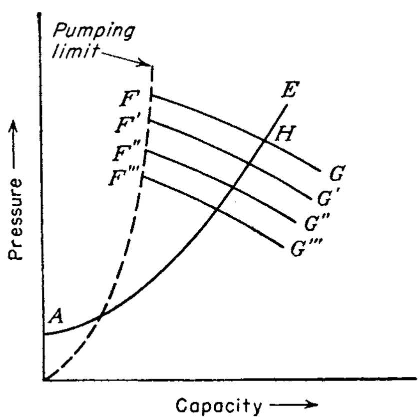

Figure 7.115 Characteristic curves of a centrifugal compressor at variable speed superimposed upon a demand-load curve.

1. Synchronous motor or squirrel-cage induction motor:
   a. Speed variation may be obtained by using hydraulic coupling between the motor and the compressor.
   b. Speed variation may be obtained by using an electric coupling between the motor and the compressor.
   c. Pressure or flow variation may be obtained by means of a butterfly valve or equivalent installed near the compressor inlet or near the compressor discharge, preferably the former.
   d. Pressure or flow variation may be obtained by adjusting the characteristic curve of a centrifugal compressor by use of adjustable inlet guide vanes or adjustable diffuser vanes.
   e. Static-type, variable-frequency control for variable-speed start for limited kVA inrush requirements with synchronous motor drive two-pole motors.
2. Wound-rotor induction motor: Speed variation may be obtained by varying the resistance in the rotor or secondary circuit using either a liquid rheostat or a step resistance (rarely used).

Briefly, a centrifugal compressor can be operated to meet a given pressure and flow requirement by varying either the demand curve or the compressor characteristic curve so that the intersection of these curves will be at the required point.

Effect of Varying Inlet Conditions

The pressure delivered by any given centrifugal unit depends on the density of the gas being compressed. This is demonstrated by Fig. 7.116 in which centrifugal characteristic curves for constant speed have been drawn for varying inlet conditions. Curve *FG* represents the characteristic for a centrifugal compressor designed to handle air at an inlet pressure of 14.4 psia, inlet temperature of 60°F, molecular weight of 28.95 (dry air), and k value of 1.398. This unit develops 15 psig at an inlet capacity of 20,000 cfm.

If the inlet temperature increases to 100°F, all other conditions remaining the same, the discharge pressure developed at 20,000 cfm is 13.7 psig. Likewise, if the inlet pressure drops to 12.4 psia, but the inlet temperature and other conditions remain as first specified, the discharge pressure at 20,000 cfm is 12.9 psig. Furthermore, if the molecular weight is the only variable, then with a 17.35 molecular weight gas and 20,000 cfm, the discharge pressure developed will be 8.2 psig. A decrease in the k value to 1.15, all other conditions unchanged, with result in a discharge pressure of 16.4 psig.

To illustrate this more precisely, Table 7.2 gives three sets of inlet conditions and the resulting discharge pressures. In each case given in Table 7.2, the compressor would develop substantially the same head, which means that the number of stages would be the same. Because of differences in pressure ratios and gas characteristics, the units for the preceding three conditions would not necessarily be physical duplicates.

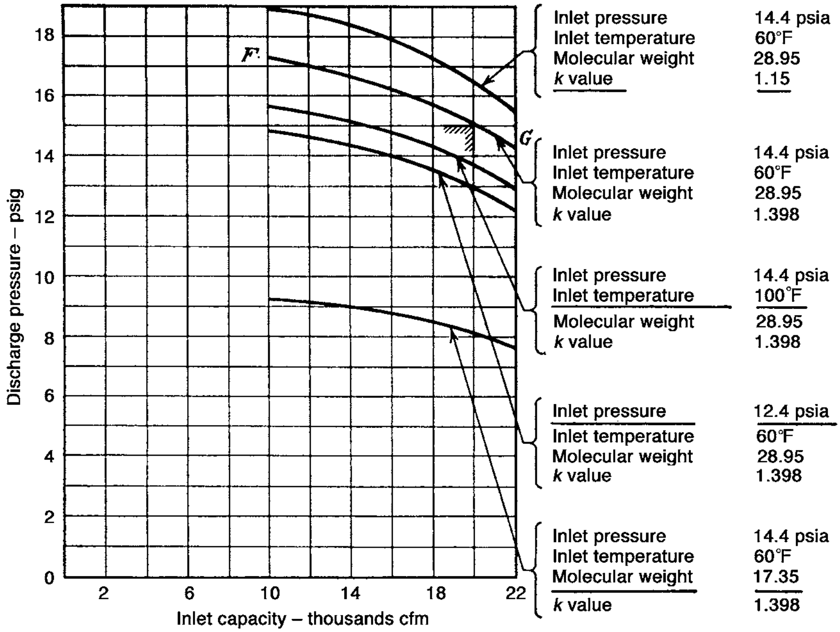

Figure 7.116 Characteristic curves for a given centrifugal compressor operating at constant speed under varying inlet conditions.

Table 7.2*

Barometer, psia	14.4	14.4	14.4
Inlet pressure, psig	- 0	- 0.65	+2.0
Inlet temperature, °F	60	110	60
Molecular weight	28.95	10.1	63.0
k value	1.398	1.36	1.11
Inlet capacity, cfm	20,000	20,000	20,000
Adiabatic head, ft-lb per lb	22,000	22,000	22,000
Discharge pressure, psig	15	3.22	64.8

* For metric equivalents, refer to Chapter 8.

Selection of Unit

The preceding study of the effect of inlet conditions on the characteristic of a compressor emphasizes two points in connection with its proper application. They are (1) the importance of investigating the operating conditions to be sure that the compressor is large enough to meet the job requirements, and (2) the necessity of a means of controlling the centrifugal compressor when variations in the operating conditions are such that a pressure would be developed in excess of the actual requirements.

1. Minimum inlet pressure
2. Maximum inlet temperature
3. Maximum molecular weight
4. Maximum *k* value
5. Maximum discharge pressure
6. Maximum inlet capacity
7. Maximum moisture content
8. Supercompressibility factor at inlet and discharge
9. Gas characteristics (e.g., analysis, corrosiveness, dirt content)

This information is necessary to ensure obtaining a centrifugal unit that will be physically capable of meeting the requirements under the most adverse operating conditions. Also, a knowledge of the range of these conditions is needed to ensure that a driver of sufficient size is furnished. Attention is called to Fig. 7.117, which shows a typical inquiry form.

Whereas a centrifugal compressor delivers practically constant discharge pressure over a considerable range of inlet flows, an axial compressor is characterized by substantially constant inlet flow over a considerable range of discharge pressure. This can be expressed in terms of stability; thus, a centrifugal compressor has a considerably greater range of stability than an axial compressor. This characteristic can be viewed from a different standpoint, however, because it also means that the flow of a centrifugal compressor must be greatly reduced to obtain any increase in pressure ratio. An axial compressor, on the other hand, can develop a very substantial increase in pressure ratio with a reduction of only 5 percent in flow rate.

In selecting a machine for a given application, the preceding characteristics must be carefully considered. The practical lower limit of capacity for which either compressor may be designed is determined by the specific requirements. In connection with low-capacity units, consideration must also be given to an economic comparison with reciprocating and rotary compressors capable of doing the same job, including cost and installation. In some instances, it may be economical to use an axial or centrifugal compressor for a portion of the required pressure range, followed by a reciprocating or rotary compressor for the remainder of the pressure range, followed by a reciprocating or rotary compressor for the remainder of the pressure range.

Compressor Data

	Normal	Maximum	Minimum
1. Inlet flow*			
2. Mass flow			
3. Molecular weight			
4. Average compressibility (Z_{ave})			
5. Average K (C_p / C_v)			
6. Relative humidity %			
7. Barometer			
8. Inlet temperature			
9. Inlet pressure			
10. Discharge temperature			
11. Discharge pressure			
12. Speed required			
13. Power required (all losses included)			
14. Side load: Flow			
Pressure			
Temperature			
15. Indicate guarantee point			
16. Delivered flow			
17. Cooling water temperature			
18. % Turn down			
19. % Rise to surge			
20. Control method			
21. Inlet throttle device			
22. Type of intercoolers and aftercooler			

Gas analysis % volume

Constituent	Guar.	Alt.
Total	Total	100.0

*If inlet flow is listed in SCFM (standard conditions) indicate conditions under which flow was measured i.e. Temperature ________ ; Pressure ________ ; ________ % R.H.

Turbine Data

	Power	Speed	Inlet pressure (Steam conditions)	Inlet temperature (Steam conditions)	Discharge pressure (Steam conditions)
Rated					
Actual (Guar.)					
Part load					
Part load					

Maximum induct / ext. flow ________ Induct / ext. pressure ________

Electric Motor Data

Site data:

Area: ☐ Class ______ Group ______ Div ______

☐ Non-hazardous

Unusual conditions: ☐ Dust

☐ Fumes

Altitude ______ Ambient Temperature ________

Enclosure type:

☐ TEFC ☐ TEWAC

☐ Weather protected ☐ TEIEF (using _____ gas)

☐ Force ventilated

☐ Open-drip proof

☐ Open

☐ Exp. proof; Class______ Group ______

Motor type:

☐ Squirrel Cage

☐ Synchronous (☐ Brushless ☐ Slip ring)

☐ Wound rotor

☐ Direct current

☐ ________

Basic Data:

________ Volts ________ Phase ________ Hertz

Nameplate power ________ Service factor ________

Synchronous rpm ________

Insulation class ________ Type ________

Temperature rise ________ °C above ________ °C by _____

Starting:

☐ Full voltage ☐ Reduced voltage ☐ Loaded ☐ Unloaded ☐ Wye-Delta ☐ Voltage dip _____%

Figure 7.117 Centrifugal compressor inquiry form.

Since the size of the required compressor and the horsepower of the driver are direct functions of inlet capacity, extreme care must be exercised in establishing actual inlet capacity. Standard capacity (scfm) is frequently used in specifications and, since there are several standards in common use, it is necessary to establish the particular standard involved. Air has been frequently specified at 14.7 psia, 60°F and dry. For all applications, the standard capacity must be corrected to the actual inlet pressure, inlet temperature, and moisture content to arrive at actual inlet capacity. Although examples are based upon the above common basis of measurement, CAGI now has adopted the ISO standard of 14.5 psia, 68°F and dry.

Approximate Selection Limitations

Although the following rule-of-thumb approach will vary among compressor manufacturers, a brief survey of the following points will serve to guide the selection of the type of compressor and, in some cases, will eliminate centrifugal units from consideration where conditions make it inherently not suited.

There is no definite minimum inlet capacity. However, single-stage and multistage compressors have been built for inlet capacities as low as 250 cfm.

Numerous commercial centrifugal compressors are in service at discharge pressures up to 2500 psig. For other applications, centrifugal compressors have been built for discharge pressures up to about 10,000 psig.

There are frequent applications involving an inlet capacity within the practical limits of a centrifugal compressor, but because of the high pressure ratio, the inlet capacity to the last stage may be so low as to preclude the use of a centrifugal compressor.

Because of thermal stresses and problems of alignment, a limit of 450°F is normally set as the maximum discharge temperature at design pressure and capacity for centrifugal compressors of standard design using standard materials. Where discharge temperatures must exceed this value, special designs and materials are used. Centrifugal compressors have been built for discharge temperatures up to about 1000°F. In cases where the pressure ratio would result in a discharge temperature in excess of 450°F and where this temperature is neither required nor desired, several alternatives may be used:

1. Multisection compressors with external coolers between sections of the same compressor.
2. Multicasing compressors with external coolers between compressors. If necessary; each compressor in the multicasing arrangement may be a multisection compressor.
3. Internal cooling of the, compressor as with injection of a coolant such as water or a condensed vapor into the gas stream in the diffuser sections of the compressor, or indirectly as with coolers mounted within the compressor casing or external to the compressor.

STAGE THEORY

This discussion will be concerned largely with the conventional compressor stage, that is, one with a radial inlet, closed impeller running at 800 to 900 fps tip speed, feeding a vaneless diffuser. However, sufficient attention will be given to such variations as inducer impellers and vaned diffusers that a general understanding of most combinations of commonly used components should result.

Discussion will center on the impeller and diffuser (Fig. 7.118), because these are the two key elements that determine the characteristic shape of the performance curve. Poorly designed inlets, collectors, or return channels can naturally affect performance, but their influence on characteristic shape is usually small and will henceforth be ignored.

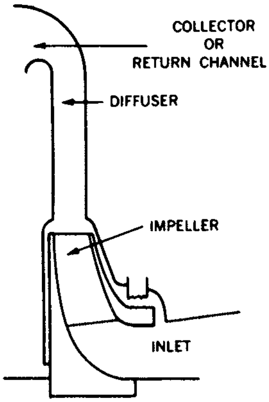

Figure 7.118 Impeller-diffuser arrangement in a centrifugal compressor.

This discussion is directly applicable to a single-stage machine and to each stage of a multi-stage machine, and the approach taken is largely qualitative.

Elements of the Characteristic Shape

Any discussion of characteristic shape must be divided into three parts. We have a basic slope of head versus flow, upon which a choke or stonewall effect must be superimposed in the overload region and a minimum flow or surge point in the underload region. The resulting overall characteristic will then be the basic slope, as altered and limited by choke at high flow, and as limited by surge at low flow in Fig. 7.119.

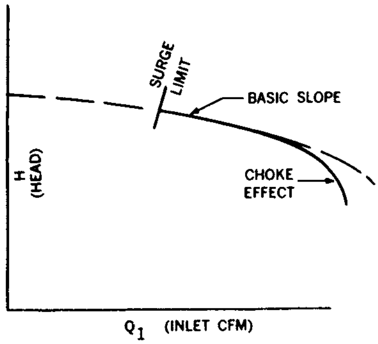

Figure 7.119 Surge and choke (or "stonewall") limitations on compressor flow.

Basic Slope

To understand basic slope, it is necessary to look at what is occurring at the impeller tip in terms of velocity vectors. In Fig. 7.120, V_{rel} represents the gas velocity relative to the blade. U_2 represents the absolute tip speed of the blade. The resultant of these two vectors is represented by V, which is the actual absolute velocity of the gas. (By vector addition, $U_2 + \rightarrow V_{\text{rel}} = V$.) It can be seen that the length of the vectors and the magnitude of the exit angle are determined by the amount of backward lean in the blade, by the tip speed of the blade, and by gas velocity relative to the blade, which is in turn dictated by tip volume flow rate for a given impeller.

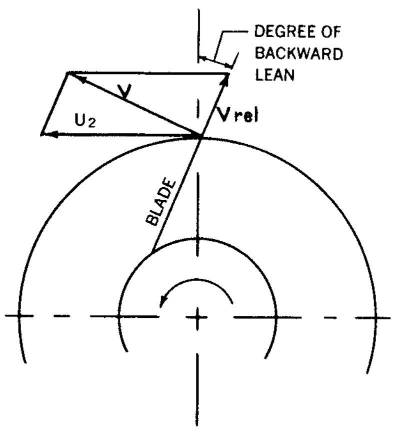

Figure 7.120 Velocity diagrams for gas and blade-tip velocities.

Having the magnitude and direction of the absolute velocity V, this vector may now be broken into its radial and tangential components, V_r and V_t, as in Fig. 7.121. The vector V_t is reduced somewhat by slip factor in a real impeller, an effect that can be ignored in a qualitative discussion such as this. The head output is proportional to the product of U_2 and V_t. For a given rpm, U_2 is constant; therefore, head is proportional to V.

The first question is what happens to the magnitude of the tangential component V_t as we vary the amount of flow passing through the impeller at constant rpm? As the flow is decreased, V_{rel} decreases. As V_{rel} decreases, angle decreases markedly. This makes V_t increase, which increases head output. This head increase with decreasing flow is the basic slope of the stage characteristic.

Blade Angle Is a Compromise

How does the degree of backward lean affect the steepness of the basic slope? One may picture a radial blade (zero backward lean). V_{rel} is now the same as V_r in

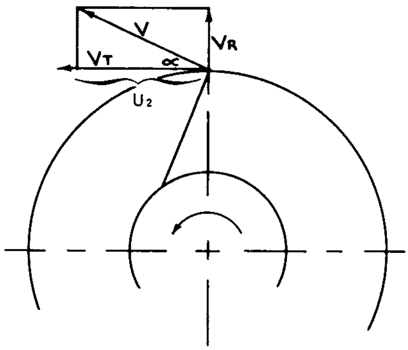

Figure 7.121 Blade velocity diagrams when flow is decreased.

Fig. 7.121, and V_t is now equal to U_2. As the flow is reduced in this impeller, V_r and α decrease as before. However, V_t remains constant. Therefore, head output remains theoretically constant regardless of flow. In a real impeller, of course, the head is reduced on increasing flow by a decrease in efficiency attributable to higher frictional losses. The resulting basic slope normally shows 2 to 3 percent head rise when going from design flow to minimum flow.

Now, for the opposite extreme, an impeller having a very high degree of backward lean (say 45 degrees off radial at the tip), it is seen that a change in flow, and therefore a change in the V_{rel} vector length, will cause very large changes in V_t, and therefore in head. Thus, such an impeller will typically produce a head rise of 20% or more when moving from design flow to minimum flow.

It is evident from the foregoing that the effect of backward lean on head output is minimized at low flow, and a high backward lean impeller will produce almost as much head at minimum flow as a low backward lean impeller running at the same tip speed. As one moves out toward design flow, however, the head difference becomes quite dramatic, as seen in Fig. 7.122. The normal industry standard for conventional closed impellers is represented by the middle line, which is 25 to 35 degrees of backward lean. This configuration is really a compromise between the high head obtainable at design flow with low backward lean blades and the steep basic slope obtainable with high backward lean blades.

One further point should be made concerning basic slope before closing the subject. In the foregoing discussion, the term flow was used without elaboration, the implication being that impeller-tip volume rate is dictated by inlet volume rate regardless of rotative speed and type of gas. This, of course, is not quite true, since gases, unlike liquids, are compressible. It is well known that a heavy gas will be compressed to a greater extent in a given stage than a light gas (i.e., the heavy gas has a higher volume ratio).

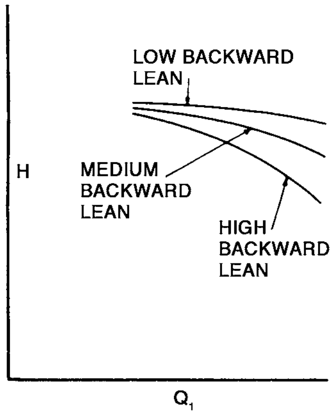

Figure 7.122 Effect of blade angle, or "lean," upon the head-flow characteristic curve.

Therefore, for a given inlet volume flow rate entering a given impeller at a given speed, the magnitude of V_{rel} is less for a heavy gas than for a light gas. If the impeller has backward lean, the magnitude of V_t will be greater for the heavy gas. Since head output is proportional to V_t, a given impeller running at a given speed will produce more head when compressing a heavy gas than when compressing a like volume of a light gas (both volumes expressed in inlet cfm). What is more, the magnitude of the difference increases as inlet flow increases, so the basic slope of a given backward lean impeller is actually less steep for a heavy gas than for a light gas. The higher the backward lean, the more pronounced this effect is (Fig. 7.123).

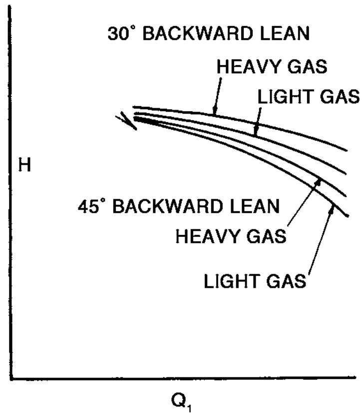

Figure 7.123 Effect of gas density upon blades of 30 degree and 45 degree blade tip angle.

Fan-law Effect

The effect of volume ratio upon what is known as fan-law is worthy of mention. Fan-law states that the cfm potential of a stage is proportional to rotative speed and that the head produced is proportional to speed squared. Reexamination of Figs. 7.120 and 7.121 will demonstrate the logic of this law.

If V_{rel} were truly proportional to inlet cfm, and both inlet cfm and speed were increased by 10%, then the head output would be 21% greater because the tip-vector geometry would maintain exact similarity. Because higher head produces higher volume ratio in a given gas, however, V_{rel} does not increase quite in proportion to speed and inlet cfm. By reasoning similar to that used in discussing heavy gas versus light gas, the head output of a backward leaning stage handling 10% more inlet cfm at 10% higher speed will increase somewhat more than 21%. By similar reasoning, if we reduce speed and inlet flow from 100 to 90%, the head produced will be slightly less than the 81% predicted by fan-law.

Fan-law received the designation of law front the fact that a fan is a low head compressor normally handling air, a light gas. Since volume ratio effects are extremely small when imparting a small head to a light gas, excellent accuracy can be obtained by the dimensional approach upon which fan-law is based. As a general rule, the higher the head, the heavier the gas, and the greater the backward lean, the poorer the accuracy will be. As a practical matter, speed changes up to 30 or 40% can be handled with sufficient accuracy for most purposes when the unit is a typical single-stage air compressor. A little more discretion must be used on multi-stage compressors handling heavy gases, however, because fan-law deviation can become quite significant for speed changes as small as 10%.

Choke Effect

The basic slope of the head flow curve has been discussed at some length, but the choke or stonewall effect that occurs at flows higher than design flow and which must be superimposed upon the basic slope (Fig. 7.119) has not yet been discussed.

Just as basic slope is controlled by impeller-tip vector geometry; the stonewall effect is normally controlled by impeller-inlet vector geometry; In Fig. 7.124, vector U_1 may be drawn to represent the tangential velocity of the leading edge of the blade similar to U_2 at the tip. Vector V may also be drawn representing absolute velocity of the inlet gas, which, having made a 90 degree turn, is now moving essentially radially (hence, the term *radial inlet*). By vector analysis, V_{rel}, which is gas velocity relative to the blade, has the magnitude and direction shown, where $U_1 + \rightarrow V_{\text{rel}} = V$. At design flow, the direction of V_{rel} essentially lines up with the blade angle as shown.

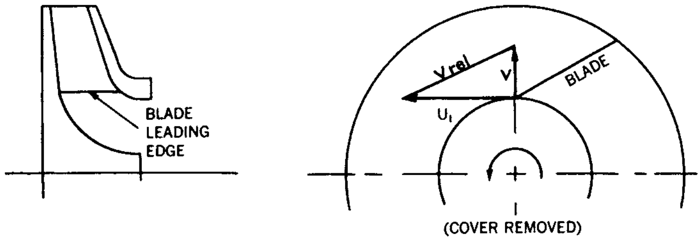

Figure 7.124 Impeller inlet geometry and velocity diagram.

Mach Number Considerations

The magnitude of V_{rel} compared to the speed of sound at the inlet pressure and temperature is called the relative inlet Mach number. It is the magnitude of this ratio that indicates stonewall effect in a conventional stage. While true stonewall effect should theoretically not be reached until the relative inlet Mach number is unity, it is conventional practice to limit the Mach number to 0.85 or 0.90 at design flow.

It is evident from Fig. 7.124 that, for a given rpm, the magnitude of $V\sim$ will diminish with decreasing flow, since V is proportional to flow. If V_{rel} decreases, then relative inlet Mach number decreases, so the stonewall effect is normally not a factor at flows below design flow It is also evident that at low flows the direction of V_{rel} is such that the gas impinges on the leading side of the blade, resulting in positive incidence, a factor that is not very detrimental to performance until very high values of positive incidence are reached.

Let us now increase flow beyond the design point. As V increases, so also does V_{rel} and relative inlet Mach number. In addition, V_{rel} now impinges on the trailing side of the blade, a condition known as negative incidence. It has been observed that high degrees of negative incidence tend to contribute to the stonewall problem as Mach number 1.00 is approached, presumably because of boundary layer separation and reduction of effective flow area in the blade pack.

Significance of Gas Weight

Since values of U_1 are typically in the 500 fps range and values of V in the 250 fps range, it is obvious that, since the speed of sound for air at 80°F is 1140 fps, lighter gases suffer no true impeller stonewall problems as described, even at high overloads. Some head loss below the basic slope will be observed, however even in the lightest gases, due in part to increased frictional losses throughout the entire stage and in part to the extreme negative incidence at high overloads.

The lightest common gas handled by conventional centrifugal compressors for which stonewall effect can be a definite factor is propylene with a sonic speed of 740 fps at - 40°. In order of increasing severity are propane at 718 fps at -40°F, butane at 630 fps at - 20°F, chlorine, and the various Freons. The traditional method of handling such gases is to use an impeller of larger than normal flow area to reduce V, and run it at lower than normal rpm to reduce U_1, thus keeping the value of V_{rel}, abnormally low. This procedure requires the use of more than the usual number of stages for a given head requirement and sometimes even requires the use of an abnormally large frame for the flow handled.

Inducer Impeller Increases Head Output

Much development work has been done in recent years toward the goal of running impellers at normal speeds on heavy gases in order to reduce hardware costs to those incurred in the compression of light gases. One approach has been to use inducer impellers (Fig. 7.125). The blades on this impeller extend down around the hub radius so that the gas first encounters the blade pack while flowing axially. Figure 7.125 shows the vector analysis at the inducer outer radius. Assuming that the inducer radius is the same as the leading edge radius of the conventional radial inlet impeller, the vector geometries of the two are identical.

The advantage of the inducer lies in the fact that, as we move radially inward along the blade leading edge, the value of U_1 and therefore of V_{rel} and Mach number, decreases. As we move along the leading edge of the conventional impeller, the vector geometry remains essentially constant. It can be seen, therefore, that while maximum Mach number for the two styles is the same, the average Mach number for the inducer is less for a given flow and speed. The inducer impeller can therefore be run somewhat faster, resulting in greater head output. The big disadvantage of the closed inducer impeller lies in the difficulty of fabrication. It is obviously more difficult to weld the longer and more curved blade path of an inducer impeller than that of a conventional impeller. Other disadvantages are the greater weight and the greater axial space required by the inducer impeller over that of the conventional impeller.

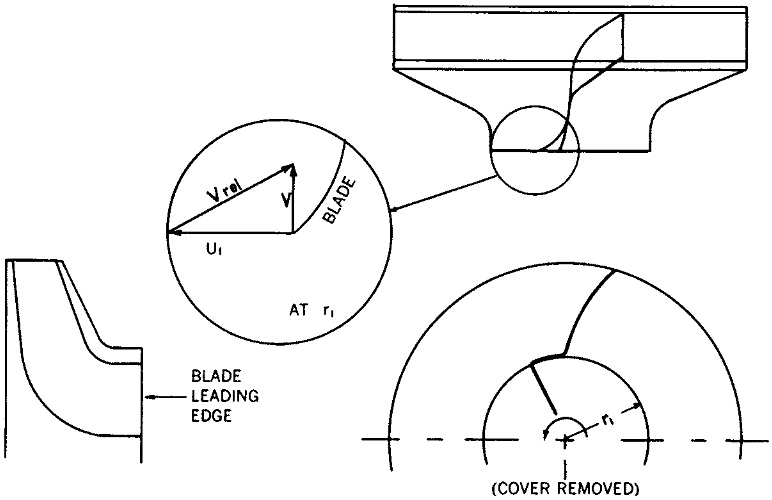

Figure 7.125 Velocity diagram for gas and blade at the outer radius of the inducer.

Another method of obtaining increased head output for a given Mach number is to reduce the backward lean. This expedient, however, has some disadvantages, not the least of which is the flatter characteristic curve that results.

Surge

Having discussed basic slope and choke, we are left with one major task, a discussion of minimum flow or surge. Surge flow has been defined by some as the flow at which the head-flow curve is perfectly flat and below which head actually decreases. This definition has a certain appeal, because straddling a surge flow, so defined, are many pairs of flow values producing identical heads, leading one to conjecture that the flow value is actually jumping back and forth between some such pair. However, since numerous centrifugal stages have been observed to run smoothly at flows below such a rate and others to surge at flows above such a rate, this definition must be considered imperfect at best. Unlike choke flow, which hurts nothing but aerodynamic performance, surge can be quite damaging physically to a compressor and should be avoided. The higher the pressure level involved, the more important this statement becomes.

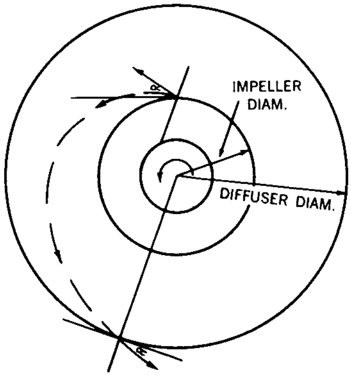

Figure 7.126 Velocity vectors at blade tips in a conventional stage.

To understand what causes surge in a conventional stage, one must refer back to the tip-vector geometry of Fig. 7.121. As flow is reduced while speed is held constant, the magnitude of V_r decreases in proportion and that of V_r remains constant for radial blades or increases for backward lean blades. As flow decreases, therefore, the value of the flow angle decreases. In the normal parallel-wall vaneless diffuser, this angle remains almost constant throughout the diffuser, so the path taken by a particle of gas is a log spiral (Fig. 7.126). The reason that angle remains constant in a parallel-wall diffuser is that both V_r and V_t vary inversely with radius V_r because radial flow area is proportional to radius and V_t, because of the law of conservation of momentum.

It is evident from Fig. 7.126 that the smaller the angle α, the longer the flow path of a given gas particle between the impeller tip and the diffuser outer diameter. When angle α becomes small enough and the diffuser flow path long enough, the flow momentum at the walls is dissipated by friction to the point at which pressure gained by diffusion causes a reversal of flow, and surge results. The angle α at which this occurs in a vaneless diffuser has been found to be quite predictable for various diffuser impeller diameter ratios. The flow and angle α at which surge occurs can be lowered somewhat by reducing diffuser diameter, but at the cost of some velocity pressure recovery.

Vaned Diffusers

Before we discuss the foregoing in more detail, let us briefly discuss vaned diffusion, a device sometimes used in high-performance air machines. Figure 7.127 shows the configuration of diffuser vanes. The vanes force the air outward in a shorter path than unguided air would take, but not so short a path as to cause too rapid deceleration with consequent stream separation and inefficiency.

The leading edge of the diffuser vane is set for shockless entry of the air at approximately design flow. It is evident that at flows lower than design the air impinges on the diffuser vanes with positive incidence. Conversely, at flows higher than design, negative incidence prevails. In a typical high-speed, high-performance air stage, positive incidence at the leading edge of the diffuser vane triggers surge on decreasing flow.

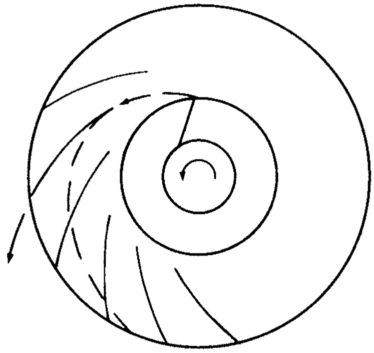

Figure 7.127 Arrangement of diffuser vanes in the centrifugal compressor casing.

On increasing flow, negative incidence at the inducer vanes can cause choking before impeller inlet stonewall is reached. In spite of this disadvantage, vaned diffusion is sometimes used for air because stage efficiency is improved 2 to 3 percent. The short flow-range problem can be alleviated by making the diffuser vanes adjustable.

Vaneless Diffusers

Having complete discussion of vaned diffusion, let us return to the more common vaneless diffuser. We have seen that when V_r and α become too small there will be surge. What can be done if the parameters are such that there is a low value for α at design flow? One may artificially increase V_r and α by pulling the diffuser walls together until it reaches the proper value at design flow. This brings to light an important distinction: head output, as discussed earlier, is controlled by vector geometry in the impeller tip largely irrespective of what happens in the diffuser. Surge point is controlled by vector geometry in the diffuser, largely irrespective of what occurred in the impeller. In the common case where impeller tip width and diffuser width are the same (Fig. 7.118), the two sets of vector geometry are the same, ignoring impeller blade solidity. If such a stage has poor stability, it is frequently possible to lower the surge point by narrowing the diffuser without markedly changing the basic slope of choke flow. This procedure can be carried only so far, however, because extreme positive incidence at the impeller inlet will eventually trigger surge, regardless of diffuser geometry

Just as we did when discussing choke flow, let us look at the effect of heavy gas compression on surge point. Since a heavy gas is compressed more at a given speed than is a light gas, it is evident that the critical value for a will be reached on decreasing flow at a higher inlet flow of heavy gas than that of a light gas. A given stage, therefore, has a higher surge flow or a lower stable range when compressing heavy gas than when compressing light gas at the same speed (Figs. 7.128 and 7.129).

By similar reasoning, a given stage compressing a given gas at varying speed will surge at somewhat different inlet flows than those predicted by fan-law When speed is 10% above design speed, for instance, surge flow will be more than 10% higher than surge flow at design speed. When speed is 90% of design, the stage will surge at less than 90% of design speed surge flow.

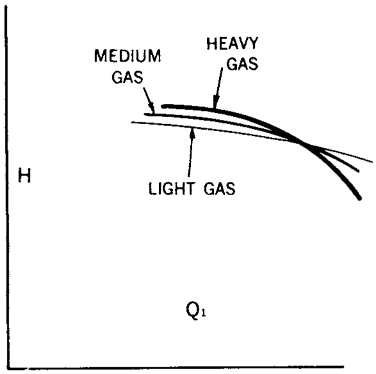

Figure 7.128 Head-flow characteristic, showing effect of gas density.

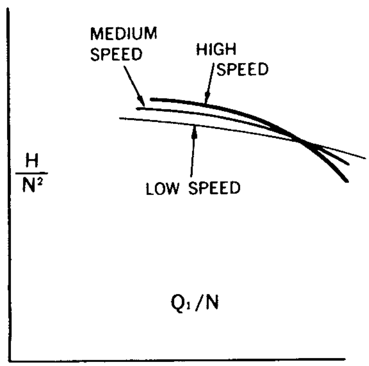

Figure 7.129 Effect of speed upon the operating characteristic of the centrifugal compressor.

GUARANTEES

Because of varying conditions of installation and operation, performance guarantees are subject to a tolerance. The limits of this tolerance are normally stated by the manufacturer. Certain chemical and petroleum standards require the capacity and head to be guaranteed with no negative tolerance.

In the case of variable-speed compressors, the specified capacity and pressure can be obtained by adjustment of speed that is not guaranteed, and the horsepower will be guaranteed within the specified tolerance.

In the case of constant-speed compressors, the specified capacity; head, and horsepower are subject to the specified tolerance, but the tolerances should be non-cumulative; that is, for a given head and capacity, the Bhp per 100 cfm shall be within the specified tolerance.

Because of difficulty in accurately predicting the characteristic curve of an axial or centrifugal compressor, only one capacity and one discharge pressure rating together with corresponding power input are normally guaranteed. The shape of the characteristic curve is seldom guaranteed.

SPECIFIC SPEED

The fundamental geometry of a centrifugal or axial compressor stage is dictated by the three variables: flow, head, and rotative speed. Specific speed is a useful tool for quickly evaluating the type of stage that will be required to relate these three variables properly for a given application. Mathematically, it is expressed by the following equation:

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

where:

N_s = specific speed
N = design rotative speed, rpm
Q = design gas flow, icfm
H = design isentropic head per stage, ft-lb/lb

It is not necessary to grasp the physical significance of the definition of specific speed to use it intelligently. It should be considered to be a type characteristic of the impeller that specifies its general proportions and characteristics, rather than as an rpm for special conditions (Fig. 7.130).

In general, the radial impeller will be used for specific speeds between 400 and 950. For specific speeds of 800 to 1400, the mixed-flow impeller is used, and axial compressors operate in the 1300 and higher ranges.

Specific speed as previously defined, and as it has been traditionally used in American industry, is not truly dimensionless. A simple conversion into 51 units will, therefore, not yield the same parameters as those stated here. Until this matter has been restudied, it is suggested that the compressor variables involved be converted from SI to English Engineering units and use made of Fig. 7.131 and the parameter ranges given previously. Obviously, these ranges are overlapping, since specific speed is only a general guide to design. Its usefulness lies in providing an estimated evaluation of compatibility of compressor operating requirements and driver speed. The specific speed for a given set of conditions may be readily obtained from Fig. 7.131.

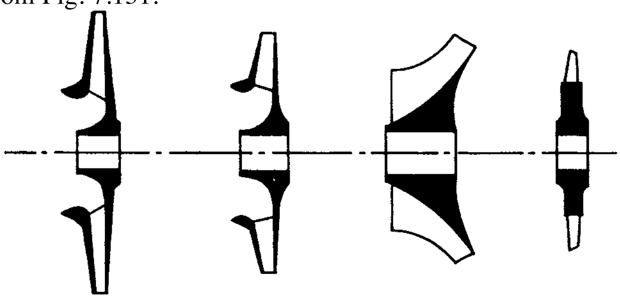

Figure 7.130 Typical impeller proportions for various specific speeds.

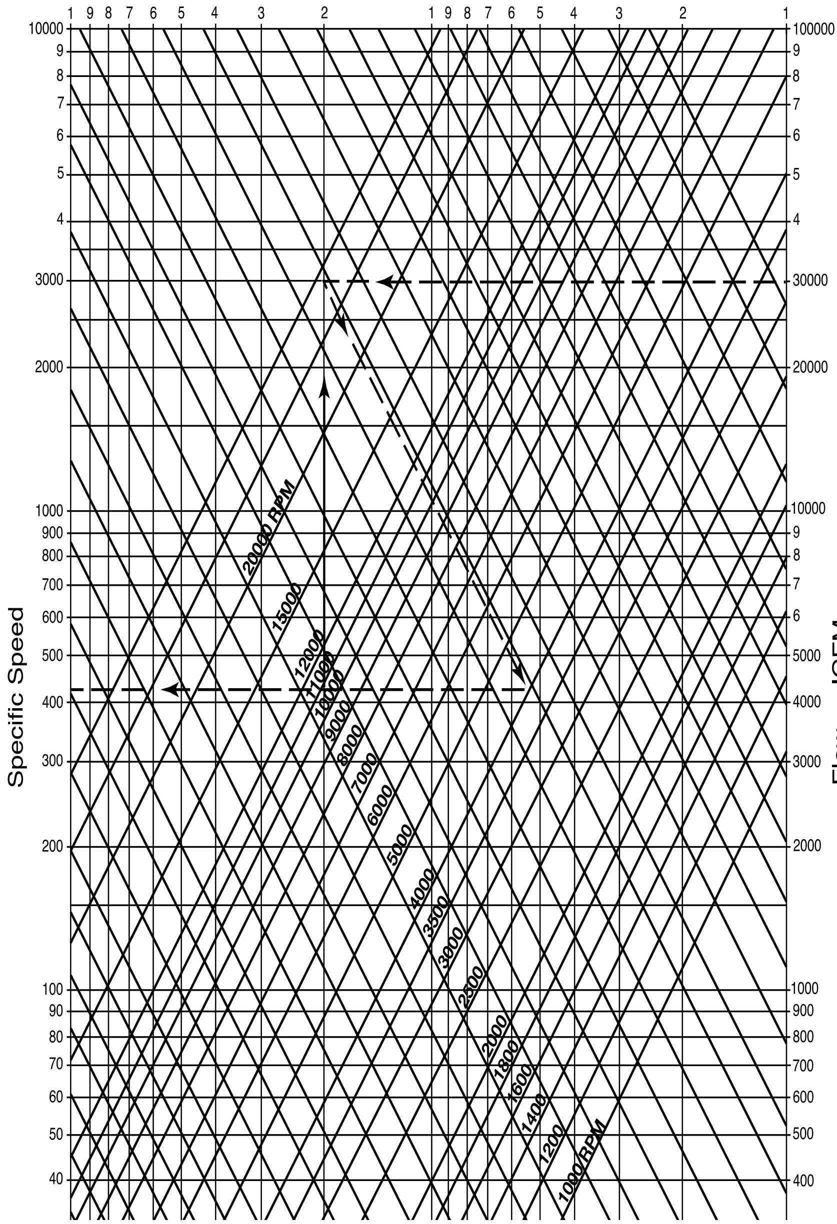

Figure 7.131 Specific speed versus flow, speed and head.

DRIVERS

The compressor driver, whether turbine, motor, or engine, is an important part of any compressor installation. It requires proper selection and matching to the compressor to ensure a satisfactory installation. It is preferable to have the compressor manufacturer select the driver and furnish a complete unit, including compressor, driver, lubrication system, seal system, coupling, and control, so that all elements may be properly coordinated. Thus one manufacturer assumes the responsibility for the complete package.

The selection of the type of driver is determined by the service requirement of the compressor, the conditions at the proposed site, and the availability and economy of electricity and various types of fuels or waste process gases.

The drivers commonly used with centrifugal compressors include:

1. Steam turbines
2. Electric motors
3. Gas expansion turbines
4. Combustion gas turbines
5. Internal combustion engines

Steam Turbines

Since centrifugal compressors are inherently high speed units with rpm sometimes exceeding 15,000 and with horsepower up to 100,000 hp, they are well suited for use with direct-connected steam turbines. Steam turbine drivers may be designed for practically any set of steam conditions and are, therefore, readily adaptable, within limits, to whatever steam may be available. Where low-pressure exhaust steam from the turbine can be utilized in the plant system, a short, sometimes single-stage, turbine may be used. Where steam consumption of the turbine is of prime importance, a multi-stage turbine may be selected. When low-pressure steam in small quantities is available continuously or intermittently, a mixed-pressure turbine may be selected. An automatic-extraction turbine can be designed to supply low-pressure steam when such steam is required.

For ratings up to 4000 or 5000 hp, turbines are generally equipped with a single, governor-controlled steam inlet valve supplemented by part load hand valves. For larger ratings and higher steam flows, 100,000 lb/hour or higher, turbines equipped with multiple, governor-controlled steam inlet valves result in better speed control and improved economy with varying loads and are generally preferred to the single-governor valve design.

Construction and materials of the turbine should be suitable for the maximum steam conditions, horsepower, speed, and other requirements as specified, in accordance with the latest edition of NEMA standards for mechanical drive applications. The turbine should be rated so that it will carry the maximum load requirement of the compressor under the most adverse operating conditions. When specifying the turbine, the following minimum information should be given:

Rated Conditions*

1. Horsepower.
2. Speed, rpm.
3. Initial steam pressure at throttle, psig.
4. Initial steam temperature at throttle, °F.
5. Steam pressure at exhaust, psig or in. Hg absolute.
6. Adjustable speed range, percent.
7. Speed variation, percent.
8. Service, continuous or intermittent.
9. Type of automatic speed control.
10. Altitude at place of installation.
11. Construction, indoors or weather-protected.

Electric Motor Drivers

Standard electric motors are limited to a maximum speed of 3600 rpm and, therefore, usually require a speed-increasing gear between the motor and the compressor. A number of centrifugal air compressors are manufactured with an integral or built-in speed-increasing gear, which allows direct connection of the compressor to a standard-speed motor. For a constant-speed compressor, an alternating-current, squirrel-cage induction or synchronous motor is normally used. For a variable-speed compressor, a wound-rotor induction or direct-current motor may be used. Where variable speed is required but where conditions indicate a constant-speed driver to be most suitable, a variable-speed transmission such as an electric or hydraulic coupling may be used to vary the compressor speed.

The selection of the type of motor and form of enclosure to ensure successful operation and reliable service should be made only after giving careful consideration to the following conditions:

1. Voltage, number of phases and frequency.
2. Speed-torque curves for both starting and operating conditions.
3. Flywheel effect, WK^2, of compressor, gear, and coupling, all referred to the motor shaft speed.
4. kVA inrush limitation of power source.
5. Need for system power factor correction.
6. Permissible motor overload (service factor).
7. Altitude at place of installation.
8. Ambient temperature.

*Conditions other than rated should be clearly specified.

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9. Required speed range.
10. Continuous or intermittent service.
11. Desired motor efficiency.
12. Frequency of starting in normal service.
13. Presence of gritty or conducting dust, lint, oil vapor, salt air, corrosive fumes, flammable or explosive gases, or existence of any other ambient condition that may affect the successful operation or life of the motor.
14. Any other conditions that may affect the performance or life of the motor.

When other than a direct-connected motor is used, the choice of the speed for the motor is determined by selecting the most economical combination of motor and gear.

For geared rotary or dynamic compressors, the equivalent flywheel effect referred to the motor shaft is given by

$$WK_{e}^{2q} \quad WK_{cr}^{2} \quad = \frac{(\text{compressor speed in rpm})^2}{(\text{driver speed in rpm})^2} + WK_{8}^{2}$$

where subscripts *cr* stands for compressor rotor and *g* for gear.

Where a synchronous motor must be used and uninterrupted service is required, automatic resynchronization is normally furnished. This is accomplished by specifying 100% pull-in torque. Direct-current motors are sometimes used for compressor drives, but the speeds usually available make geared units necessary. Variable speed, controlled either manually or automatically, is obtained by a suitable field rheostat.

Motor Control

Motor control may be purchased separately if desired. The control must be properly coordinated if the installation is to work satisfactorily. There are frequently some restrictions in connection with the power supply system that will determine the type of motor control required. Both the motor manufacturer and the control manufacturer should be completely informed of what is contemplated.

The types of control available and most frequently used for the different types of motors are:

Squirrel-cage Induction Motors

1. Manual, full voltage.
2. Manual, reduced voltage.
3. Magnetic, full voltage.
4. Magnetic, reduced voltage.

Wound-rotor Induction Motors

1. Manual, secondary resistance.
2. Motor driven, secondary resistance.
3. Magnetic, secondary and primary control.

Synchronous Motors

1. Magnetic, full voltage.
2. Magnetic, reduced voltage.
3. Semi-magnetic, reduced voltage.

Gears and Variable-speed Couplings

Gears and variable-speed couplings, when used with centrifugal compressors, should be selected so that they are of ample continuous rating so as to meet the maximum power requirements of the compressor. When excess power is specified in the driver, such as a 15% service factor stipulated for the motor, they should have a continuous horsepower rating equal to the maximum continuous horsepower rating of the driver, if this is greater than the maximum specified power requirement of the compressor.

Gas Expansion Turbines

Gas expansion turbines, sometimes called expanders, are selected according to the same general specifications as outlined for steam turbines. Gas expansion turbines are adaptable only where an adequate supply of gas at elevated temperatures, up to 1400°F, is available and when this gas is at such a pressure that the necessary power can be developed by its expansion. An electric motor or steam turbine is often used with an expansion turbine to provide starting and supplementary power.

Combustion Gas Turbines

A combustion gas turbine includes an expansion gas turbine as well as a fuel combustion chamber. Due to the inherently high speed characteristic of gas turbines, they are well suited for driving dynamic compressors. Waste-heat recovery systems can be used with a combustion gas turbine to increase the operating efficiency of the gas turbine.

Gas turbines are generally of two basic designs; general industrial and jet-engine derivative. The latter utilizes a jet engine as a gasifier in conjunction with a separate power turbine that drives the compressor.

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Internal Combustion Engines

Reciprocating engine drivers, because of their relatively low rotative speeds, require speed-increasing gears for centrifugal compressor drives. Such engines use natural gas from the pipeline as fuel. Torsional vibration studies, normally made on engine applications, usually indicate the need for a damping provision built into the gear or coupling. This is not a very common method of driving centrifugal compressors.

CONTROL OF DYNAMIC COMPRESSORS

Control systems for dynamic compressors, axial and centrifugal, may become quite complex; however, they have fundamentally only two functions to accomplish:

1. Antisurge control to prevent the compressor from operating below its stable operating range and thereby exposing either itself or the process equipment to damage.
2. Performance control to adjust the output of the compressor to the demands of the user's process.

All dynamic compressors have what is commonly called a surge limit or minimum flow point below which the performance of the compressor is unstable. This instability manifests itself in pulsations in pressure and flow, which may become severe enough to cause damage.

The surge limit and type of antisurge control are affected by:

1. Type of compressor
2. Design pressure ratio
3. Characteristics of the gas handled
    a. Inlet temperature
    b. Gas constant R
    c. *k* value
4. Speed

Generally, the gas handled is of constant composition, and the constants *R* and *k* can be neglected in the design of the anti-surge equipment. Where variation in suction temperature may be wide, temperature must be compensated for.

To keep a compressor from surging, all that is required is to maintain a flow greater than the safe minimum. If the consumer's requirements are not greater than the minimum stable flow, the difference must be either blown off or recycled to the suction of the compressor.

The control engineer's responsibility, then, is to match the compressor surge line with a control system characteristic so that compressor surge is never reached. To accomplish this, the engineer must know the surge characteristics of the compressor being designed.

Figure 7.132 shows the typical pressure-volume characteristics of an axial compressor, and Fig. 7.133 shows those of a centrifugal compressor. From these curves, it can be seen that an axial compressor at a given speed approximates a constant volume variable pressure characteristic. Thus, the axial compressor anti-surge protection is most frequently oriented to pressure and that of the centrifugal unit to flow.

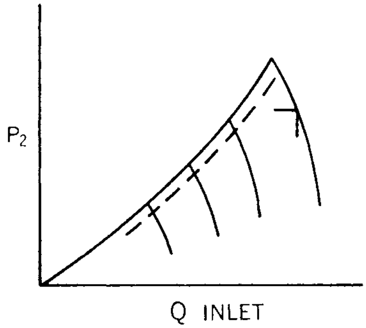

Figure 7.132 Typical performance characteristic curve for an axial-flow pressor with variable speed.

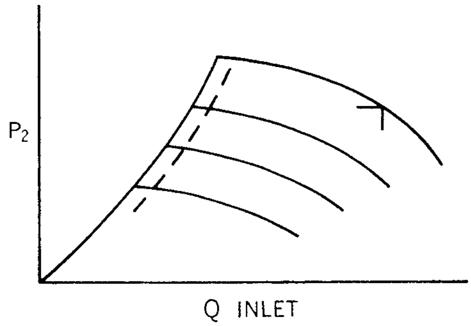

Figure 7.133 Typical performance characteristic curve for centrifugal compressor with variable speed.

Assume that the compressor is operating at the rated point at full speed (Fig. 7.134), and the process demand decreases with a pressure-oriented anti-surge control system. The discharge pressure will increase. The pressure transmitter, *FX*, will monitor the pressure, sending a signal to *SC*, which will open the blow-off valve, *BOV*, when the pressure reaches the set point or dashed line. If the compressor were driven by a variable-speed drive, a speed transmitter, *SX*, would supply a signal to *SC*, which would modify the set point. The dashed line is the locus of the speed-modified set point. In the same manner, a temperature transmitter, *TX*, could be used to modify the set point for temperature variations. If the performance control is by suction throttling and not variable speed, a pressure transmitter at the inlet will supply a signal to the *SC* to modify the set point. In this case, the *SC* is actually a pressure ratio controller.

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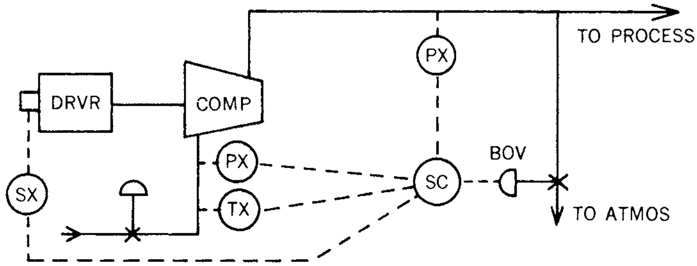

Figure 7.134 A typical pressure-oriented anti-surge control system.

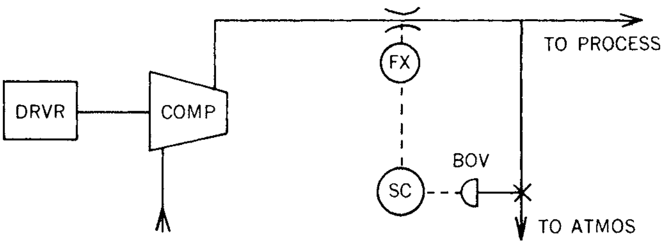

Figure 7.135 A typical flow-oriented anti-surge control system.

Assume again that the compressor is operating at the rated point at full speed (Fig. 7.135), and the process demand decreases. The flow will decrease. The flow transmitter, *FX*, will monitor the flow sending a signal to *SC*, which will open the blow-off valve, *BOV*, when the flow reaches the minimum value or set point. With flow monitoring at the discharge of the compressor, no modification of the set point is required with speed, since the flow element ΔP remains essentially constant for normal speed variations in a machine taking in atmospheric air.

This insensitivity of the set point to speed variation is illustrated as follows:

The ΔP across a flow element is proportional to ρV^2, where

ρ = density
V = velocity through the flow element

At the discharge, ρ is proportional to the pressure, which is proportional to the square of the speed, neglecting the minor effect of temperature. Therefore,

$$\rho d \sim N^2$$

The actual suction volume at which the compressor surges varies approximately directly with the speed. If the outlet condition to the compressor is essentially constant, the weight flow at which surge occurs is proportional to the volume and, therefore, varies directly with the speed. This would not be true if suction throttling performance control were used; however, suction throttling and variable-speed control are not normally both employed in the same control system. Therefore,

$$W \sim N$$

and

$$Qd \sim \frac{W}{\rho d}$$

But the velocity through the flow element is directly proportional to Qp. Therefore,

$$V \sim \frac{W}{\rho d}$$

Substituting these proportionalities into V^2 in terms of N,

$$\Delta P \sim \rho d$$

$$\Delta P \sim N^2 \left(\frac{N}{N^2} \right)^2$$

$$\Delta P \sim \frac{N^2 N^2}{N^4} = 1$$

or, rather, the ΔP is independent of the speed.

ANTI-SURGE CONTROL WITH SUCTION THROTTLING PERFORMANCE CONTROL, CONSTANT SPEED

Figure 7.136 shows the typical performance characteristics of a compressor controlled by suction throttling of the inlet at constant speed. The upper curve, 1, represents the performance with no suction throttling. Curves 2 and 3 are the performance at increasing degrees of suction throttling. The actual inlet volume to the compressor at surge is the same at points *a, b,* and *c*, and so is the pressure ratio. Flow, Q, as plotted in Fig. 7.136, therefore is the volume measured before the suction throttle valve.

There are three methods of protecting the compressor from surging when this mode of performance control is used.

1. Figure 7.137A illustrates a pressure ratio control. When pressure ratio control is used the ratio of the discharge pressure to the inlet pressure is calculated by the controller and maintained at an arbitrary value somewhat lower than the surge pressure ratio by opening the blow-off valve. The disadvantage of this system is that, if the compressor char-

acteristic is flat, much of the range must be sacrificed (Fig. 7.137B). The shaded area indicates the range that is not available. However, it is a very easy and inexpensive system to install.

2. A second method, which increases the operating range of the compressor, is illustrated in Fig. 7.138A. This system maintains a minimum ΔP as the set point. The operating range loss is illustrated in Fig. 7.138B. Again the shaded area indicates the range that is not available through this mode of control.

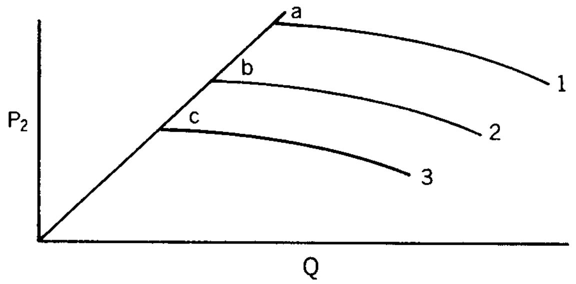

Figure 7.136 Performance characteristics of a compressor with inlet throttling control.

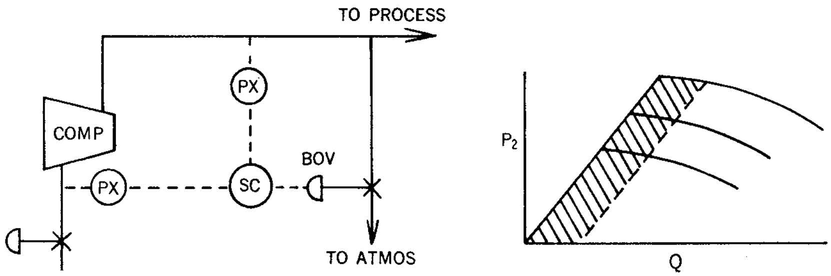

Figure 7.137 (A) Diagram of pressure-ratio control, and (B) the resulting performance characteristic curve.

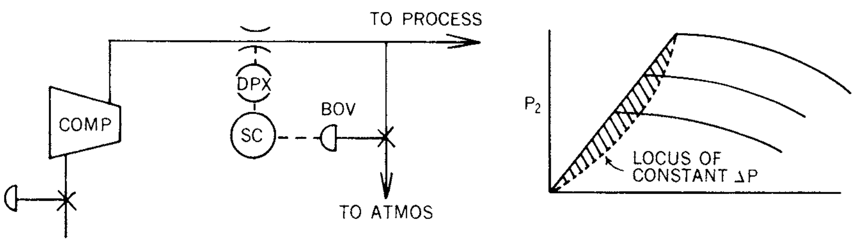

Figure 7.138 (A) and (B) Alternate method to increase the operating range of the compressor.

3. The system illustrated in Fig. 7.139A and B will match the suction throttle surge line exactly and provides the maximum possible operating range. In this mode of control, a minimum $\Delta P/P_2$ is maintained. Since a constant value of $\Delta P/P_2$ can be established that exactly matches the surge line, the maximum range of operation is available. In actual practice, a value of $\Delta P/P_2$ slightly to the right of the surge line would be used.

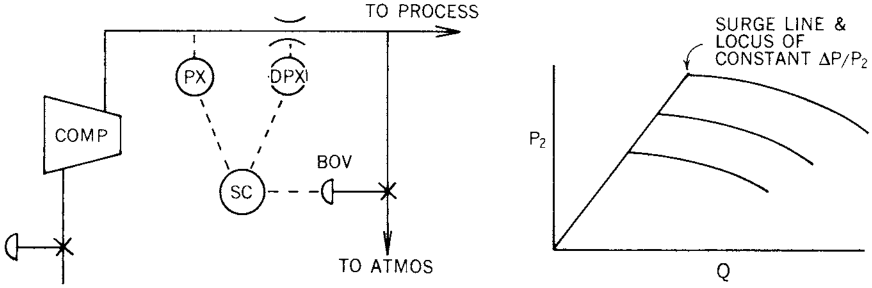

Figure 7.139 (A) and (B) Alternate control system to provide maximum operating range.

ANTI-SURGE CONTROL WITH ADJUSTABLE INLET GUIDE VANES

Compressors equipped with adjustable inlet guide vanes present an entirely different anti-surge control problem. The surge line of the compressor is a function of both the flow and the guide vane setting. Several schemes are used, and their complexity is dependent on how closely the surge line is to be matched. The simplest system is shown in Fig. 7.140A. The pressure drop, ΔP, across the flow element is continuously monitored. If the consumer's requirements decrease to a point where the flow element ΔP would be less than the set point, the surge controller opens the *BOV* to maintain the set point. Figure 7.140B shows the compressor characteristic and the control characteristic. Note, however, the extensive range that is lost with this mode of anti-surge control.

A wider useful range of the compressor characteristic can be retained with the anti-surge control system seen in Fig. 7.141. Both the flow element ΔP and the discharge pressure P_2 are monitored, and a minimum ratio of $\Delta P/P_2$ is maintained by the *BOV* if the consumer requirements are not sufficient. As can be seen, this mode increases the usable range.

The maximum usable range is obtained with the antisurge control system seen in Fig. 7.142. The differential signal from the flow element is modified by a signal that is a function of the inlet guide vane position. By proper shaping of a cam in the guide vane position transmitter *SX*, the set point can be modified to exactly parallel the compressor surge line and the loss in usable range held to a minimum.

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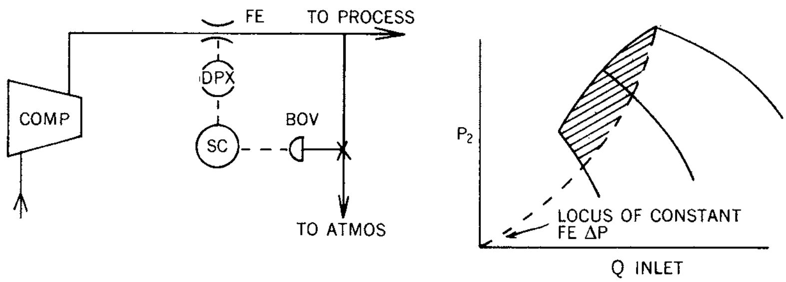

Figure 7.140 (A) and (B) Simplest anti-surge control with adjustable inlet guide vanes.

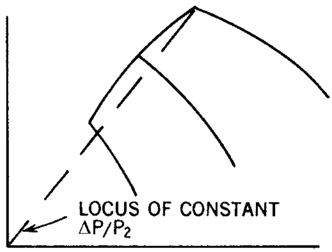

Figure 7.141 Alternate system of adjustable inlet guide vane control.

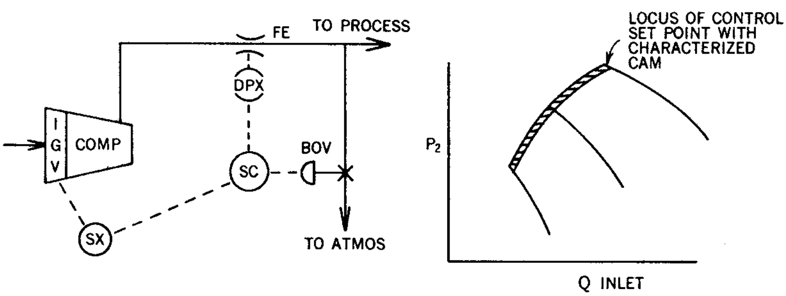

Figure 7.142 (A) and (B) System which results in maximum usable range.

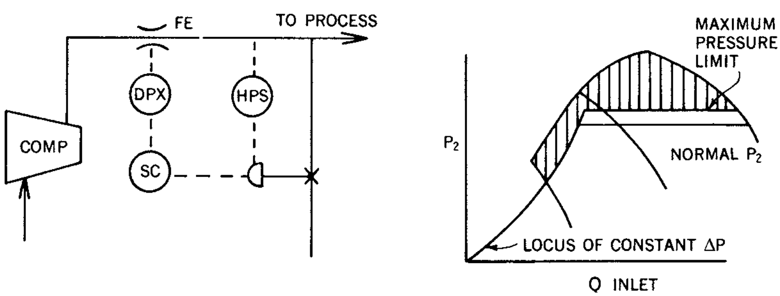

Figure 7.143 (A) and (B) Second alternate system of adjustable inlet guide vane control utilizing a pressure switch.

If the performance control is one of reasonably constant discharge pressure, a modification of the system (Fig. 7.140) is usually sufficient, is the simplest, and gives adequate range. It is frequently used in plant air anti-surge control systems.

In the mode in Fig. 7.143, the minimum set point ΔP is set lower, thus increasing the usable range. However, by so doing, the vane positions near full open are not protected from surge. By the addition of a pressure switch, this area can be protected. The pressure switch, set 5 to 10 psig higher than control pressure, directly opens the *BOV* on an increase in pressure above set point.

PERFORMANCE CONTROL

In the previous paragraphs, the basic methods of anti-surge control have been discussed. Anti-surge control is generally a passive control until the pre-established conditions have been reached, at which time it then controls the compressor to protect the system from surging. In addition to this protective control, further controls are necessary to adapt the compressor performance to the varying load requirements of the process it supplies.

A compressor control system can be designed to maintain a desired pressure to a process or a desired flow to a process. It cannot be designed to maintain both.

PRESSURE CONTROL

Figure 7.144 shows a typical system requiring constant pressure control. The process shown here might be a petrochemical process where the pressure P_2 at the process must be maintained at a fixed value, regardless of the flow through the process. It could also represent a plant air system where the plant air pressure must be maintained at 100 psig regardless of usage.

Constant pressure control can be accomplished by:

Variable speed
Adjustable inlet guide vanes or adjustable diffuser vanes
Intake throttling
Discharge throttling
Blow-off (recycle)

The methods are listed in decreasing order of efficiency.

In Fig. 7.144, the pressure is monitored, and a signal from the pressure transmitter (*PX*) is sent to a pressure controller (*PC*), which adjusts one of the above devices to maintain a constant pressure.

Each of the five methods of pressure control will be discussed. In the discussion, a centrifugal compressor will be assumed; however, the principles are the same for an axial compressor.

Variable-speed Constant Pressure Control

Figure 7.145 shows a typical set of variable-speed compressor characteristic curves. Each curve shows the pressure at which the compressor supplies a certain volume rate of flow, Q. If the compressor outlet is choked below a certain rate at any given speed, a pulsating flow, called compressor surge, will occur, as shown by the limiting curve.

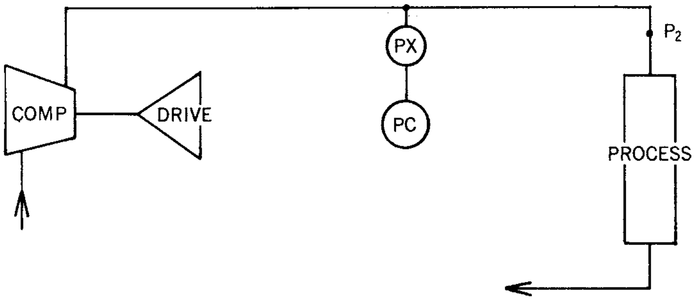

Figure 7.144 A typical system requiring constant pressure control.

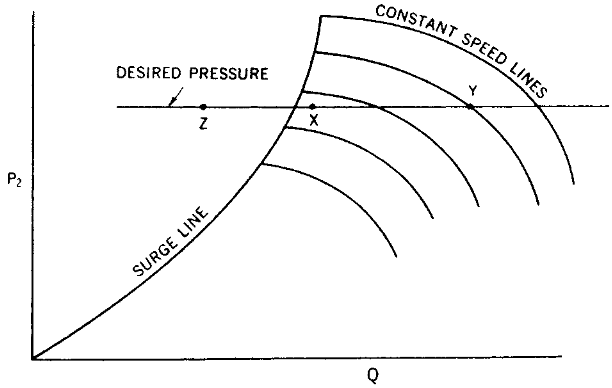

Figure 7.145 Characteristic curves for variable speed, constant pressure control.

We may assume that the compressor is operating at point Q_Y and the process requires a higher flow. The pressure immediately tends to fall as the operating point moves out to the right and downward along the characteristic curve for the given speed. However, the control system, sensing this drop in pressure, will increase the speed of the driver to return the pressure to the desired value. Conversely, had the flow decreased, the pressure would have tended to increase and the control system would decrease the speed at the driver until the desired pressure was reached. The flow could be reduced by speed reduction until point X was reached. The compressor must operate at this reduced speed in order to maintain the desired pressure. The anti-surge controls discussed earlier will prevent the operating point moving to left of a point on each speed curve similar to point *X*. If the process required a flow rate of only Q_z the volume $Q_x - Q_z$ would have to be blown off or recycled. This is accomplished by transferring from variable speed to blow-off control, which is the only suitable control when the process requires flows below the stable operating range. (For a more complete discussion see the paragraph on Blow-off, Constant Pressure Control).

ADJUSTABLE INLET GUIDE VANE CONSTANT PRESSURE CONTROL

Figure 7.146 shows a typical set of adjustable inlet guide vane characteristic curves. Let us assume that the compressor is operating at flow rate Q_Y and the process requires a higher flow. As the pressure tends to fall, the control system immediately opens the guide vanes to return the pressure to the set point. Conversely, the control system will close the vanes as the flow requirements decrease until point *X* is reached, at which time the anti-surge control comes into play. Further reduction in process requirements, as with variable-speed control, can only be accommodated by blow-off.

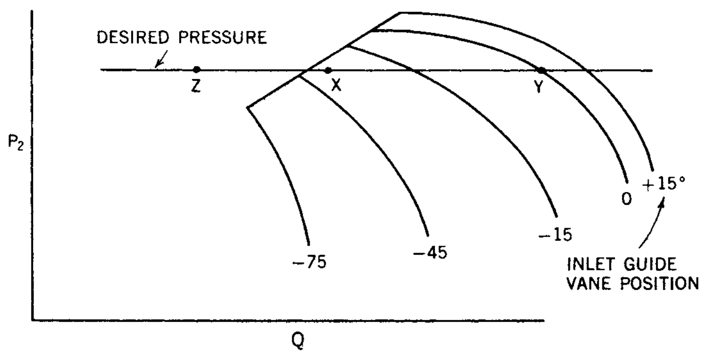

Figure 7.146 Characteristic curve typical of adjustable inlet guide vanes.

SUCTION THROTTLING, CONSTANT PRESSURE CONTROL

The two previous methods of control utilized the families of characteristic curves available with variable-speed or inlet guide vanes for constant pressure control. Intake, or suction, throttling control is usually used where the compressor is not equipped with inlet guide vanes and is driven by a constant-speed drive. It therefore has a single pressure-volume characteristic curve. However, variable performance can be achieved through suction throttling through valve TV-1.

Assume that the compressor is operating at point W on its unthrottled characteristic curve and there is a reduction in process requirements. If unthrottled, the pressure would increase to Y^1. However, by throttling across TV-1, the inlet pressure can be reduced and, although the compressor is operating at the pressure ratio and volume of Y^1 the discharge pressure and volume flow to the process will be equivalent to point Y. An example will help to clarify this: Assume TV-1 is open:

$$Q_w = 100\% \qquad \frac{P^3}{P_{1w}} = 2.0$$

$$Q_{y1} = 80\% \qquad \frac{P^3}{P_1} Y^1 = 2.10$$

Inlet pressure P_1 14.7 psia = P_2 (no throttling)
Desired P_3 = 29.4 psia

At $(P_3/P_1)y^1$, the pressure ratio is 2.10. To maintain the discharge pressure of 29.4, the inlet pressure P_2 must be reduced to 14.0 (29.4/2.1). The volume to the compressor at y1 is 80%, but the equivalent volume at 1 is less than the ratio of 80% x (14.0/14.7) or 76.3%. This is the actual volume of P_1 conditions delivered to the process.

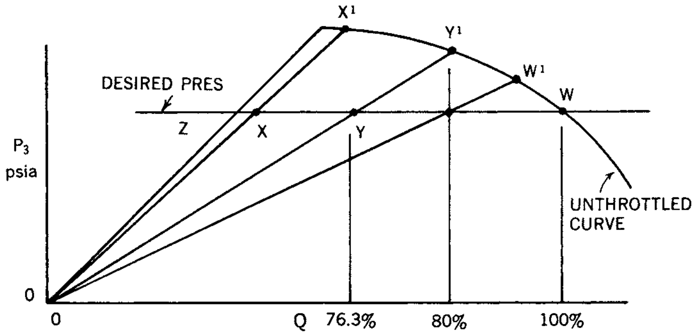

Figure 7.147 Typical throttled inlet characteristic curves.

The preceding calculation can be done by a graphical solution, if the pressure-volume characteristic is plotted with ordinates and abscissa oriented from zero, as seen in Fig. 7.147. Suction throttling control can reduce the flow to point X. Any further reduction in flow again requires blow-off.

DISCHARGE THROTTLING, CONSTANT PRESSURE CONTROL

As with intake throttling, there is only a single pressure-volume characteristic curve with discharge throttling. Pressure control is maintained simply by throttling actual compressor discharge pressure to the value desired. Discharge throttling requires more power than intake throttling for the same flow, as illustrated by the following example.

Referring to Fig. 7.147, we assume that the process requires 80% flow with discharge throttling, TV-1 would be in the discharge line, the compressor must operate at Y^1, and the gas must be throttled to the desired pressure. With suction throttling, the compressor operates at W^1 at a lower pressure ratio. Although the actual inlet volume to the compressor is higher with suction throttling, the weight flow to the process is the same; and since the pressure ratio is lower, the horsepower required is lower. For this reason, discharge throttling control is seldom used. A diagram of the controls involved in inlet throttling is shown in Fig. 7.148.

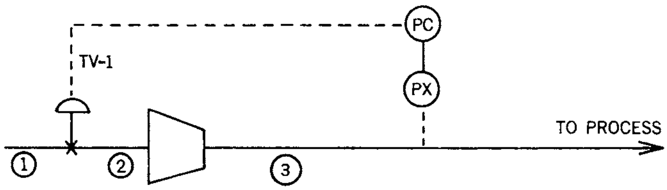

Figure 7.148 Schematic diagram of the controls involved in inlet throttling.

BLOW-OFF (RECYCLE), CONSTANT PRESSURE CONTROL

This is the least efficient method of control and is used only in conjunction with the more efficient control methods to extend their control range. If only blow-off control were used (Fig. 7.147), the compressor would always operate at point W regardless of the process requirements. The difference in flow between the process requirements and Q_w would have to be blown off and all the work expended on this extra flow wasted. For flows less than the surge limit, no other recourse is available, and blow-off must be used. The anti-surge control system utilizes blow-off control, as described earlier.

FLOW CONTROL

Figure 7.149 shows a typical system requiring constant volume control. The process shown might be a blast furnace where the flow delivered to the furnace must be held constant regardless of the varying resistance of the furnace and its charge.

As with pressure control, volume control can be accomplished by:

Variable speed
Adjustable inlet guide vanes (IGV) or adjustable diffuser units
Suction throttling (STV)
Discharge throttling (DTV)
Blow-off

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In Fig. 7.149, the flow is monitored by the flow element *FE*, and a signal proportional to the flow is sent to the flow controller to adjust the necessary device to utilize one of the control methods above. Not all the equipment shown in Fig. 7.139B would be required for any given single method of control.

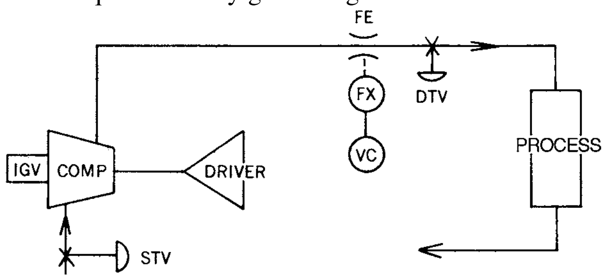

Figure 7.149 Typical system requiring constant volume control.

VARIABLE-SPEED, CONSTANT-FLOW CONTROL

Figure 7.150 shows a typical variable-speed compressor characteristic with constant-flow requirements superimposed on it. We will assume that the compressor is operating at point *Y* and the process resistance increases. The compressor will immediately tend to decrease in flow as the operating point thus moves up to the left along its characteristic curve. However, the control system sensing the decrease in flow will increase the speed until the desired flow at the higher resistance is again maintained at point Y^1. Conversely, if the resistance decreases and the flow increases, the speed will be reduced. Any desired flow may be chosen and controlled within the shaded area. If the compressor has flow-oriented, anti-surge control, the flow element and flow transmitter utilized for process volume control are the same as those used in the anti-surge system; and once the anti-surge system comes into play, flow control of the process is lost. If flow control were required in the area to the left of the surge line, separate flow elements and transmitters would be required, one serving the process control and the other the anti-surge system.

ADJUSTABLE INLET GUIDE VANE, CONSTANT-FLOW CONTROL

Figure 7.151 shows a typical adjustable inlet guide vane characteristic with constant volume requirements superimposed on it. We will assume that the compressor is operating at *Y* and that the process resistance decreases. The flow will begin to increase as the compressor operates lower on its characteristic curve. However, the control system will immediately sense the increased flow and close the *IGV* until the desired flow is reestablished at the lower pressure, point Y^1. The reverse occurs with an increase in resistance. The desired volume set point may be chosen anywhere to the right and below the surge line. However, the usable portion of this area is determined by the type of anti-surge control utilized. The reader is referred to the discussion of anti-surge control a few paragraphs earlier.

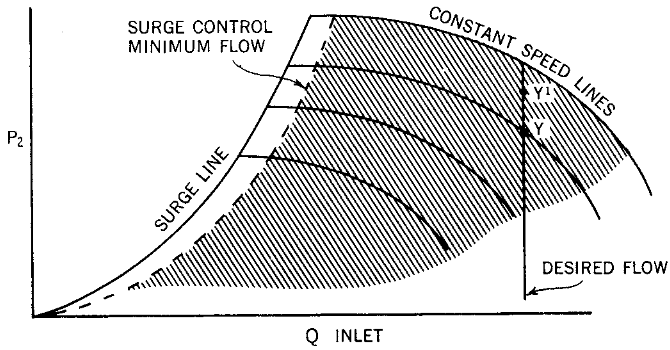

Figure 7.150 Variable speed characteristic curve with superimposed constant flow requirement.

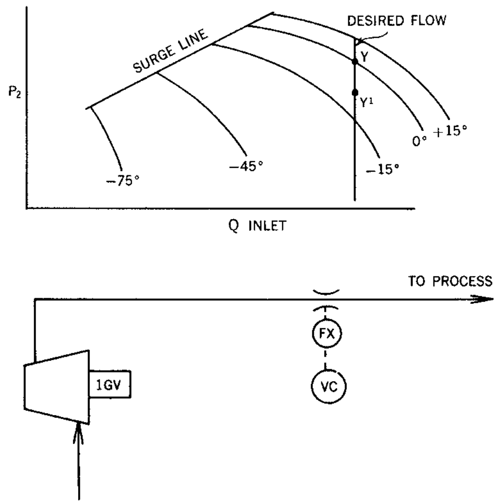

Figure 7.151 (A) and (B) Adjustable inlet guide vanes with constant volume requirements superimposed.

SUCTION THROTTLING, CONSTANT-FLOW CONTROL

Figure 7.152 shows a typical suction-throttling characteristic with constant-flow process requirements. Suction-throttling for flow control operates in exactly the same manner as it does for pressure control except that the throttle valve is operated in response to a change in flow, rather than a change in pressure. We may assume that the compressor is operating at point *W* on its throttled characteristic curve and that there is a reduction in process resistance. If unthrottled, the flow would increase toward Y^1 until the process resistance were again matched. However, the control system, sensing the increased flow, partially closes the suction throttle valve to compensate for the reduced process resistance. The control system then throttles the flow until it is equal to the desired flow, *Y*, as measured at (1) the actual inlet flow to the compressor at the reduced pressure and (2) is equal to the flow at Y^1.

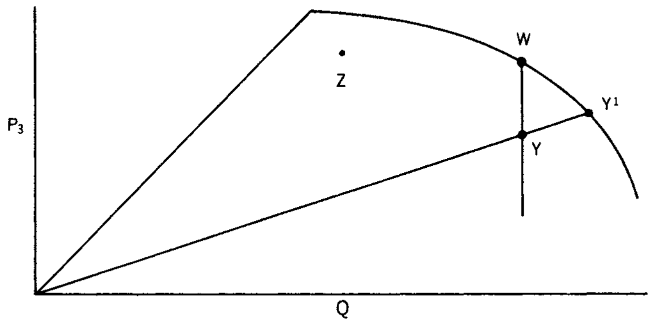

Figure 7.152 Suction throttling characteristic curve with constant-flow process requirement superimposed.

Discharge Throttling, Constant-flow Control

Constant-flow control can also be accomplished with discharge throttling; however, as with discharge throttling and constant-pressure control, it is less efficient and requires more power for the same flow than suction throttling. We will assume that the compressor (Fig. 7.152) is operating at *W* and that a reduction in process resistance occurs. If unthrottled, the flow will increase toward Y^1 until the process resistance is again matched. However, the control system, sensing the increased flow, will partially cover the discharge throttle valve, forcing the compressor operating point back up along the characteristic curve to *W* and to the original flow. With discharge throttling of the compressor for a given flow, the compressor will operate at a maximum power level regardless of the process resistance. Suction-throttling control allows the power level to reduce as the process resistance decreases. Discharge throttling, therefore, is seldom used as a method of flow control.

Blow-off, Constant-flow Control

As with constant-pressure control, blow-off control is used only to extend the operating range and as antisurge protection for more efficient control methods. As an example of the inefficiency of this method, let us assume that the process requires operation at point *Z*. The compressor will operate at point *W* on its characteristic curve, and the flow $Q_w - Q_z$ will be blown off, all the work done upon this extra flow being wasted.

CENTRIFUGAL COMPRESSOR SHAFT SEAL

Centrifugal compressors are inherently high speed machines. They compress gases as mild as air and those as toxic, flammable, and corrosive as hydrogen sulfide. They operate at all pressure levels from high vacuum to over 10,000 psig. As one might expect, the compressor shaft seal can take many forms, from a mere restriction to minimize gas losses to a complex buffered design that must be 100% leak-tight to minimize the possibility of fire or of harm to personnel. There can be five basic types of shaft seals applied in centrifugal compressors:

1. Labyrinth seal
2. Restrictive-ring seal
3. Mechanical (contact) seal
4. Liquid-film seal
5. Pumping liquid-film seal

These seals may be used independently or in combination with one another to satisfy various requirements.

Labyrinth-type Shaft Seal

The labyrinth-type seal, when applied in its simplest form, is not a true seal. It is merely a device to limit the loss of a gas without contact between the shaft and the compressor casing. However, the labyrinth can be sectionalized to provide one or more annuli, which are buffered or educted, or both, to eliminate the process gas loss or to channel its flow in a controlled manner.

The labyrinth consists of a series of straight or staggered, close-clearance, short-length restrictions that take on the appearance of knife edges. These thin restrictions can be machined from either aluminum, bronze, filled TFE, or other suitable material and mounted in the compressor end housing as shown in Fig. 7.153. The labyrinths may also be made from an 18-8 stainless-steel strip, 0.10 to 0.015 in. thick, formed into the cross-section of a J and caulked into the shaft with a soft, stainless-type wire (Fig. 7.154).

Labyrinths can be applied at any rotating speed because the physical contact area is nil, and a slight rub during acceleration or deceleration should not precipitate a serious failure. The knife edges merely wipe or bend. However, when the labyrinth is buffered or educted, proper care should be given to sizing the annuli and the associated system to ensure the effectiveness of the desired gas seal. In the case of a machine touch point seal, a babbitt or other similar mating material surface may be provided.

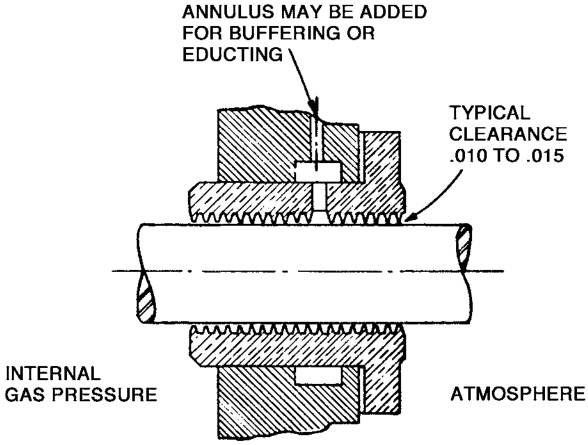

Figure 7.153 Labyrinth type shaft seal.

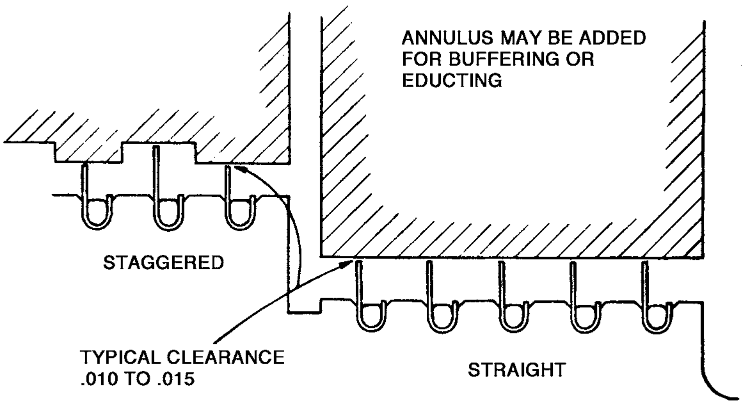

Figure 7.154 Labyrinth type shaft seal.

Restrictive-ring Shaft Seal

The restrictive-ring seal, like the labyrinth seal, is not a true seal when applied in its simplest form. It is a device to limit the loss of gas to a greater extent than the labyrinth without prolonged contact between rings and shaft. The rings may be arranged with one or more annuli for buffering or educting, or both (Fig. 7.155). The ring-type seal is considerably more effective per unit of length than the labyrinth seal. The clearances can be made smaller because the rings are usually free to float with the shaft to avoid binding or severe rubbing.

The rings are usually made of carbon, using either one piece or segmented construction. A garter-type spring is used with the segmented construction to hold the segments together and impart an axial force to the carbon through an angled joint.

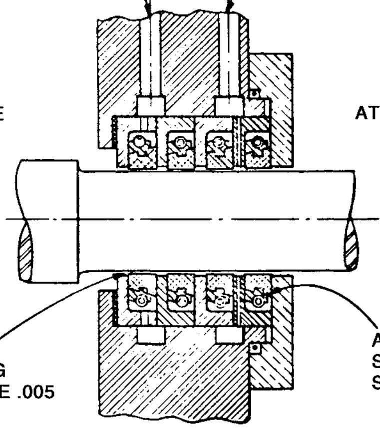

Figure 7.155 Restrictive ring type shaft seal.

The ring is thus held tight against the housing to maintain position and to avoid side leakage. When the rings are buffered or educted, the same precaution must be taken in sizing the annuli and the associated system as with the labyrinth-type seals.

Mechanical (Contact) Shaft Seal

The mechanical seal is the next step toward positive sealing in that it limits the loss of gas to almost a negligible amount because there is an extremely small clearance between the carbon and the mating shoulder (Fig. 7.156). The life of the seal is not very predictable when run dry. Therefore, most mechanical seals are buffered with a lubricating oil, which is supplied in sufficient quantity and at high enough differential pressure above the gas to ensure positive sealing and lubrication of the rubbing faces. The oil-buffered mechanical seal entails a second restrictive seal that limits the liquid sealant flow to the atmosphere to that amount required for heat carry-off. This clean-side leakage is returned to the reservoir.

There is a small amount of inner sealant leakage toward the gas. This leakage is gathered in a trap and is manually or automatically drained. This sealant is either discarded or reclaimed. The possibility of imperfect contact always exists with a mechanical seal. However, most designs inherently provide a positive seal at shutdown without a buffering fluid as long as the carbon face and mating shoulder are in intimate contact, a feature that makes such seals very attractive in refrigeration applications.

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Efforts have been made to minimize the rubbing speed between the carbon face and the mating shoulder. One method that reduces the rubbing speed employs a carbon ring that rubs both the rotating seat affixed to the shaft and a stationary spring-loaded seat, like that in Fig. 7.157. The carbon ring rotates at a speed less than the shaft rotational speed.

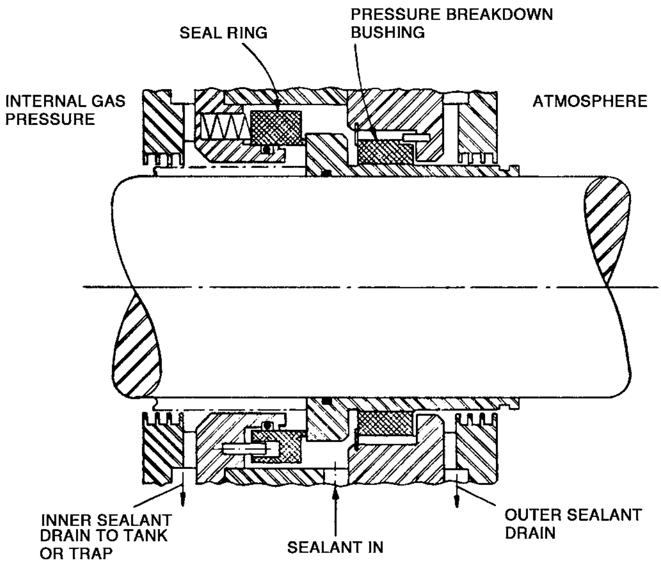

Figure 7.156 Mechanical contact type shaft seal.

Liquid-film Shaft Seal

A liquid-film seal employs double floating bushings or rings that have small clearances with respect to the shaft. They are buffered in between with a sealant, usually oil, to create a positive flow of sealant toward the gas within the compressor and toward the atmosphere. This type of seal is seen in Fig. 7.158. This seal is one of the simplest to manufacture and operate, if the inner sealant leakage is reusable, because the clearances can be made comparable to those in journal bearings. However, if the inner sealant leakage must be discarded or put through a clean-up process, the running clearances have to be made very small, and such small clearances can lead to mechanical problems.

Pumping Liquid-film Shaft Seal

The pumping liquid-film seal is essentially a buffered liquid-film seal with large clearances, backed up with a viscous-type pump positioned adjacent to the gas side of the inner bushing. The pump creates a back pressure that impedes the sealant flow toward the gas side of the seal during operation (Fig. 7.159).

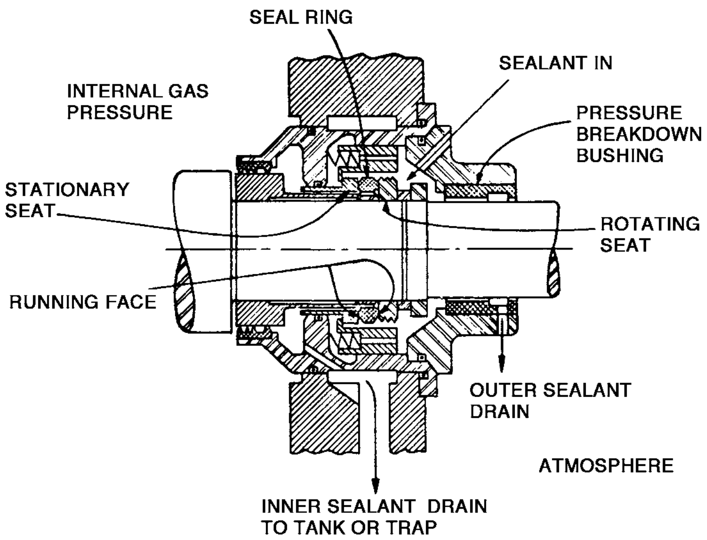

Figure 7.157 Mechanical contact type shaft seal.

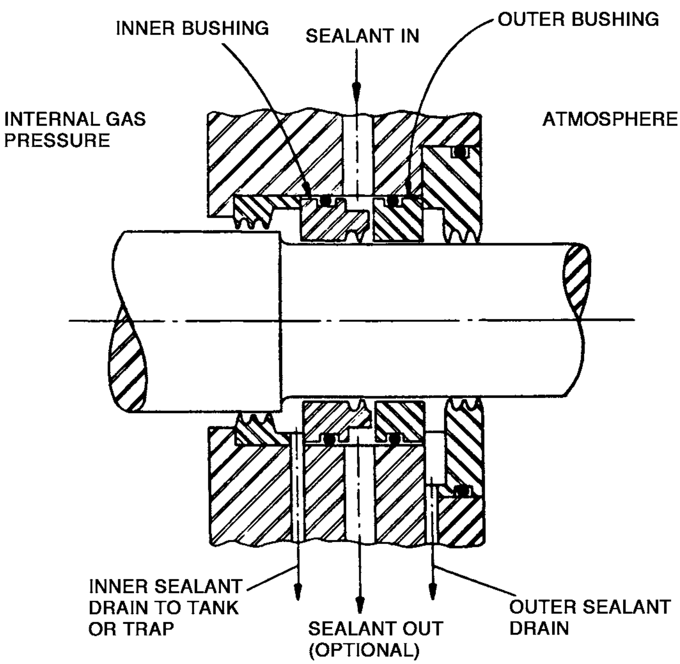

Figure 7.158 Liquid film type shaft seal.

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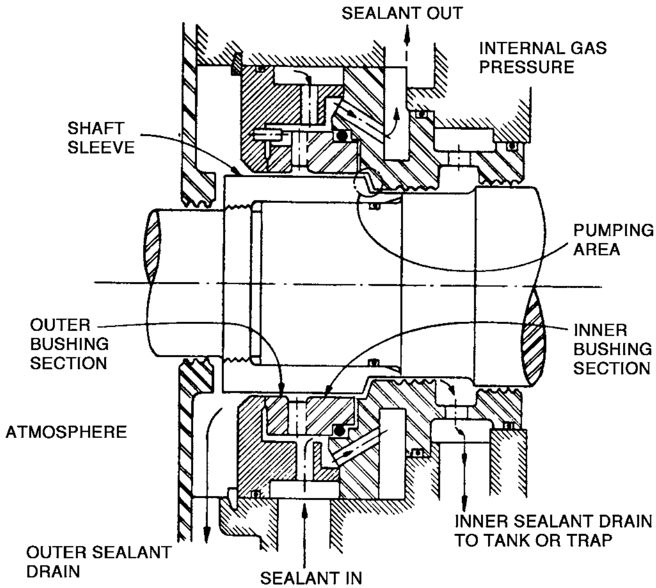

Figure 7.159 Pumping liquid film type shaft seal.

This back pressure is a function of speed. It counterbalances the oil-to-gas pressure differential during operation in the normal speed range of the compressor. It performs like a mechanical seal with respect to the amount of sealant reaching the gas side of the seal. This can be as little as only a few gallons a day. At standstill and at low-speed operation, it performs like the high clearance seal described previously. The running clearances within this seal between the bushing and the shaft are comparable to those in journal bearings, all others being several times this magnitude. The inherent advantage of this type of seal is its ability to operate with large clearances and yet experience low sealant leakage at operating speeds.

Selection of Compressor Shaft Seal and Associated Systems

Selection of the compressor shaft seal and the associated system depends primarily on gas composition, gas pressure levels, process requirements, and the need for reliability. However, where buffer gas or liquid-type seals are involved, the seals can perform no better than the associated sealant system. A seal selection should not be made until all aspects of the process and compressor are considered.

Among the process requirements and limitations to be considered are the following: The gas to be compressed, pressures at inlet and outlet, and whether a buffer gas or liquid sealant will be used. Disposal of the leakage gas must also be considered. Compressor details and operating cycles also influence seal selection or design.

Cost of the gas being handled is a factor, as well as its temperature, flash point, toxicity, and chemical activity or inertness. Pressures during operation and off design, during recycle, and at stagnation or holding are important, the last especially so in refrigeration compressors. Pressures during process upsets and safety relief valve settings may dominate the design considerations. The operating cycle, whether continuous or intermittent, and the plant turnaround cycle will also influence the design.

If there is to be a buffer gas, it must be compatible with the process gas, and its composition, whether it is inert or sweet, or sweet but flammable, must also be taken into account. Availability, cost, pressure level, and reliability must also be considered. In the case of a liquid sealant or buffer, the same factors enter the picture, and one may also need to look into the cost of recovering leakage to the gas side of the seal.

The leakage gas may be allowed to escape to the atmosphere, or it may be transferred to another part of the process. Its pressure, tolerance for air and moisture, and any educator and operating medium requirements must be taken into account.

In the compressor, the shaft diameter and peripheral speed must be known, plus the axial length of the seal, how isolated it will be from the compressor bearing, and how it will be influenced by critical speeds, either during normal running or while accelerating through the critical speeds.

In the case of buffered seals, it is imperative to have a quality sealant supply system; that is, the sealant system must contain auxiliaries, spares, or means for isolating and bypassing components so that they may be maintained or replaced without upsetting the pressure level or flow of the sealant. Power supplies must be from separate sources or must be different: steam, AC power, DC power, or gas. System monitoring has to be employed to warn of impending failure, actuate auxiliaries, and trip the entire unit to avoid extensive damage. Many applications require a large rundown tank referenced to the process gas pressure level, which provides the sealant for normal compressor operation for a specified time after a total sealant pump failure. After the unit is tripped, the run-down tank also provides sealant during the coastdown of the compressor, isolation of the compressor, venting of the compressor, and during any purge of the compressor in the case of a flammable or toxic gas.

PERFORMANCE CALCULATIONS

Within the ranges of pressure and temperature usually encountered, air follows the perfect gas laws. Some other diatomic gases also follow perfect gas laws, but hydrocarbon and other gases and many mixed gases deviate to a considerable extent. Since many applications involve performance calculations for air only, this section has been divided into parts, as follows:

1. Information required for compressor calculations, compression formulas, and step-by-step explanation of performance calculations.
2. Performance calculations for air with corrections for humidity.
3. Performance calculations for gases, including determination, from the gas analysis or applicable gas constants.

Introduction to Curves and Performance Calculations

Table 7.3 contains formulas for calculating centrifugal compressor head, discharge temperature, and horsepower. Derivations are based on commonly accepted thermodynamic relations for gases. Isothermal compression calculations are generally used only when extensive cooling is accomplished during the compression cycle. This cooling can take the form of shell and tube intercooling between stages of compression or some form of liquid injection. Expected overall isothermal efficiency must be determined by the designer of the compressor. Discharge temperatures depend on the type of cooling and location of coolers and cannot be determined directly from isothermal efficiency. Hence, isothermal compression calculations are of little use to the estimator.

Either adiabatic or polytropic relations can be used as the basis for comparing centrifugal compressor performance. In recent years, however, polytropic relations have generally replaced adiabatic relations for this purpose. When the performance of a given compressor or stage is known, application of this performance information to gases with different specific heat ratios is more straightforward with polytropic relations. Calculations and examples used in this chapter, therefore, will be based on polytropic compression.

Explanation of Centrifugal Compression Calculations

A. Information Required for Compressor Calculations

1. Physical properties of gas being compressed
   Molecular weight, MW
   Adiabatic exponent, k
   Supercompressibility factor, Z
   Relative humidity, RH
   (These can be determined from complete gas analysis)
2. Inlet conditions at compressor flange
   Capacity, cfm, lb/minute
   mm scfd (at defined conditions) Inlet temperature, °F
   Inlet pressure, p, (psia)
3. Discharge pressure at
   compressor flange, p_2 (psia)
4. Water temperature, if intercooling is used, t_w, °F
5. Process temperature limitations, if any

	Adiabatic	Polytropic	Isothermal
Compression process	$Pv^k = C$	$Pv^n = C$	$Pv = C$
Determination of exponent	$k = \frac{c_p}{c_v}$	$\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\eta_p}$	
Theoretical discharge temperature, °F abs	$T_2 = T_1 r^{(k-1)/k}$	$T_2 = T_1 r^{(n-1)/n}$	$T_2 = T_1$
Discharge temperature, °F abs	$T_2 = T_1 + \frac{T_1[r^{(k-1)/k} - 1]}{\eta_{ad}}$	$T_2 = T_1 r^{(n-1)/n}$	$T_2 = T_1$
Head (H) (ft-lb/lb)	$H_{ad} = Z_{av} R T_1 \frac{r^{(k-1)/k} - 1}{(k-1)/k}$	$H_p = Z_{av} R T_1 \frac{r^{(n-1)/n} - 1}{(n-1)/n}$	$H_i = Z_{av} R T_1 \ln_e r$
Gas horsepower, ghp (using capacity)	$\text{ghp} = \frac{Q_1 P_1 \frac{Z_1 + Z_2}{2Z_1}[r^{(k-1)/k} - 1]}{229\eta_{ad}\left(\frac{k-1}{k}\right)}$	$\text{ghp} = \frac{Q_1 P_1 \frac{Z_1 + Z_2}{2Z_1}[r^{(n-1)/n} - 1]}{229\eta_p\left(\frac{n-1}{n}\right)}$	$\text{ghp} = \frac{Q_1 P_1 \frac{Z_1 + Z_2}{2Z_1} \ln_e r}{229\eta_i}$
Gas horsepower, ghp (using weight)	$\text{ghp} - \frac{WH_{ad}}{33{,}000\ \eta_{ad}}$	$\text{ghp} - \frac{WH_p}{33{,}000\ \eta_p}$	$\text{ghp} = \frac{WH_1}{33{,}000\ \eta_i}$
Brake horsepower, Bhp	Bhp = ghp + mech. losses	Bhp = ghp + mech. losses	Bhp = ghp + mech. losses

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B. Preliminary Calculations

1. From inlet capacity in cfm, the approximate polytropic efficiency may be determined from Fig. 7.160. The actual value will vary due to speed, specific wheel design, compression ratio, and other factors. The compressor manufacturer must be contacted when accurate data is desired.
2. Polytropic ratio $(n - 1)/n$ may be found from the ratio of specific heats (k) and the polytropic efficiency (η_p) using Fig. 7.161.

$$\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\eta_p}$$

3. Compression ratio, r, may be found from inlet and discharge pressures.

$$r = \frac{p_2}{p_1}$$

C. Discharge Temperature

$$T_2 = T_1 \times r^{\frac{n-1}{n}}$$

The discharge temperature must be considered in the selection of materials and design of a compressor.

D. Polytropic Head

1. If supercompressibility factor is involved, discharge supercompressibility Z_2 may be found from the gas analysis. The discharge temperature and pressure may then be calculated.

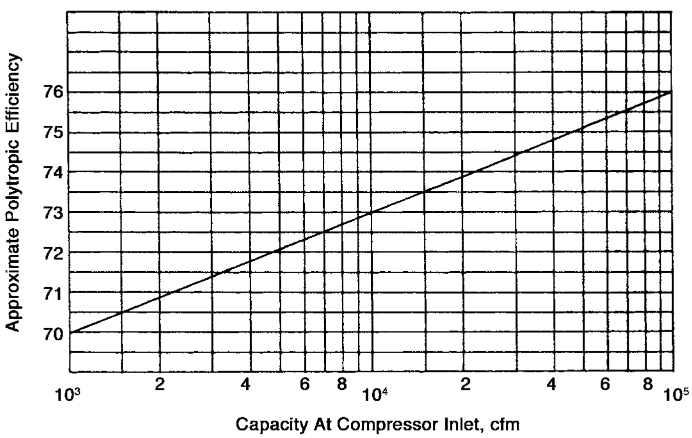

Figure 7.160 Approximate polytropic efficiency versus compressor inlet capacity.

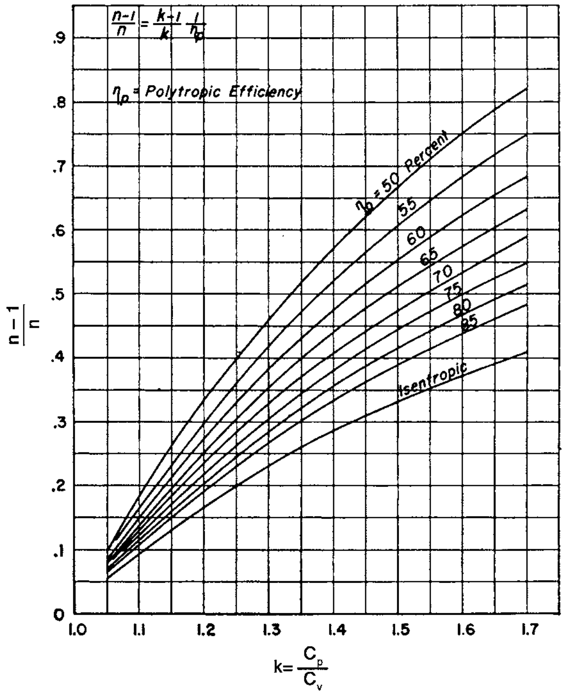

Figure 7.161 Polytropic ratio $\frac{n-1}{n}$ versus adiabatic exponent k.

2. Polytropic head:

$$H_p = \frac{Z_1 + Z_2}{2} RT_1 \frac{r^{(n-1)/n} - 1}{(n-1)/n}$$

The value of $r^{(n-1)/n}$ may be obtained from Fig. 7.162. Polytropic head is an indication of the number of impellers required for the conditions specified.

E. Gas Horsepower

1. From weight flow,

$$\text{ghp} = \frac{W \times H_p}{33{,}000 \times \eta_p}$$

2. Or, alternatively, from inlet capacity and inlet pressure,

$$\text{ghp} = \frac{Q_1 \times P_1 \times \dfrac{Z_1 + Z_2}{2Z_1} \times \dfrac{r^{(n-1)/n} - 1}{(n-1)/n}}{229 \times \eta_p}$$

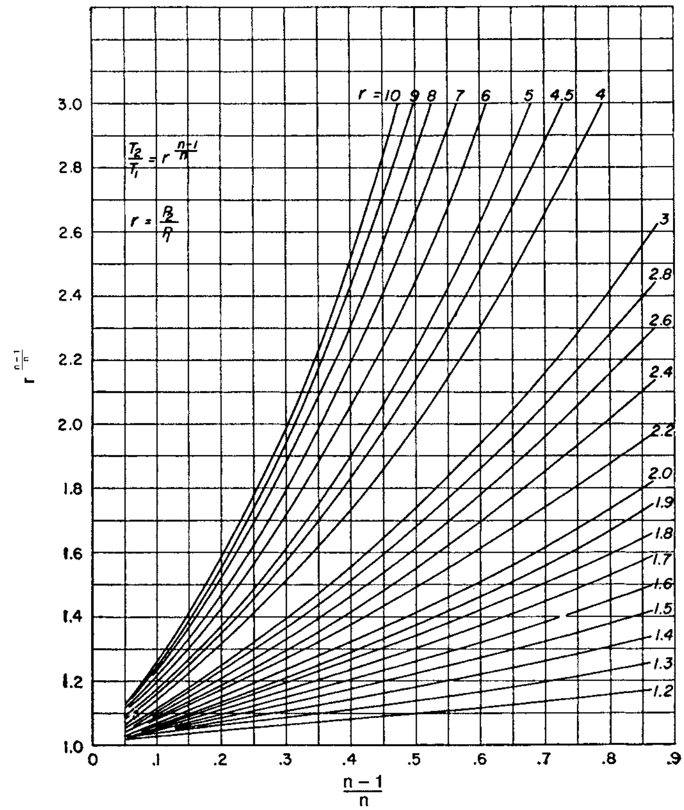

Figure 7.162 Polytropic temperature ratio versus ratio $\frac{n-1}{n}$.

F. Brake Horsepower

The gas horsepower (ghp) calculated is not the true input horsepower to the compressor. Mechanical and hydraulic losses to be considered are:

a. Bearing losses.
b. Seal losses.
c. Other losses that can be ignored in most estimating work include radiation losses, labyrinth-seal losses, and recirculation due to balancing devices.

In estimating bearing and seal losses, figures of 30 and 20 horsepower may be used, respectively. These figures are quite approximate since a given-size bearing will have variable losses depending on bearing load, rotative speed, oil temperature, and so on.

Performance Calculations for Air

1. Example Using Dry Air

A. Conditions

Capacity, 75,000 inlet cfm
Gas, dry air
k, 1.395
Molecular weight, 28.95
Inlet pressure (and barometer), 14.7 psia
Inlet temperature, 100 °F
Discharge pressure, 30 psig

B. Preliminary Calculations

1. Discharge pressure

$$p_2 = 30 + 14.7 = 44.7 \text{ psia}$$

2. Estimated polytropic efficiency (Fig. 7.160):

$$\eta p = 75.6\%$$

3. Polytropic ratio:

$$\frac{n-1}{n} = \frac{k-1}{k} \times \frac{1}{\eta_p} = \frac{0.395}{1.395} \times \frac{1}{0.756} = 0.378$$

4. $$r = \frac{p_2}{p_1} = \frac{44.7}{14.7} = 3.04$$

C. Discharge Temperature

$$T_2 = T_1 \left[r^{(n-1)/n} \right] = 560\left(3.40^{0.378}\right)$$
$$= 560 \text{ x } 1.522 = 853 \qquad (853 - 460 = 393°\text{F})$$

D. Polytropic Head

$$H_p = RT_1 \frac{r^{(n-1)/n} - 1}{(n-1)/n} = \frac{1544}{28.95} \times 560 \frac{\left(3.04^{0.378} - 1\right)}{0.378}$$
$$= 53.3 \text{ x } 560 \text{ x } \frac{0.522}{0.378}$$
$$= 41{,}200 \text{ ft-lb/lb}$$

Compressor will normally have three to five stages.

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E. Horsepower

1. $$\text{ghp} = \frac{Q_1 \times p_1 \times \dfrac{r^{(n-1)/n} - 1}{(n-1)/n}}{229 \times 0.756}$$

$$= \frac{75{,}000 \times 14.7 \times (0.522 / 0.378)}{229 \times 0.756}$$

$$= 8800 \text{ hp}$$

2. Bhp = ghp + mechanical losses (bearings)
   = 8800 + 30 = 8830hp

Humidity Corrections

Air or gas can hold varying amounts of water vapor, the amount depending on temperature and pressure conditions and the degree of saturation. All compressor calculations must take into account the presence of water vapor and include the resulting capacity corrections. In addition, centrifugal compressor calculations must also take into consideration any changes in the density and k value, both of which are affected by the water vapor.

A. To correct for water vapor, it is necessary to determine its partial pressure.
   1. If the mixture is saturated with water vapor, the dry bulb temperature of the mixture determines the vapor pressure. The vapor pressure is then the saturation pressure at mixture temperature. Dry bulb temperature, wet bulb temperature, and mixture temperature are then identical.
   2. If the mixture is not saturated, (i.e., the vapor is superheated, the dew point temperature or equivalent information must be known, as it is the temperature at which the mixture will be saturated with water vapor if cooled at constant total pressure.
   3. The amount of vapor in unsaturated mixtures may also be expressed in terms of relative humidity. Relative humidity is the ratio of the actual water vapor pressure to the partial pressure of the water vapor if the air were saturated at the dry bulb temperature of the mixture.

B. Use is made of the partial pressures in order to correct weights and capacities for water vapor present in the mixture.
   1. Whenever the weight of one constituent of a mixture of two gases is known, the weight of the other can be determined from the following form of the general gas equation.

$$W_2 = W_1 \times \frac{MW_2}{MW_1} \times \frac{p_2}{p_1}$$

Total pressure of mixture: $p = p_1 + p_2$.

For mixtures of air and water vapor, the preceding formula becomes

$$W_v = 0.622 \times W_a \times \frac{p_v}{p_a}$$

Total pressure = p_a, + p_v, where subscript a represents dry air and subscript v represents water vapor.

2. The volume of the mixture calculated by using the weight and partial pressure of any one of its constituents.

$$\frac{W_a \times R_a \times T_1}{p_a \times 144} = \frac{W_a \times 1544 \times T_a}{p_a \times 144 \times MW_a}$$

C. 1. The k value and molecular weight can he calculated from the mol percentage of the individual constituents making up the mixture (see Chapter 8).

2. Specific humidity (SH = lb of water vapor/lb dry air) is useful in determining k and MW for air-water vapor mixtures from Fig. 7.163.

$$\text{SH} = \frac{W_v}{W_a} = 0.622 \frac{p_v}{p_a}$$

Figure 7.164, which gives specific humidity of air-vapor mixtures at saturation, is also useful in calculating drop out or condensation, when mixtures are cooled between two or more stages of compression.

Example of Compressor Calculation Involving Humid Air

In many cases involving air, inlet capacity is referred to standard conditions, dry, which is another way of indicating the weight of dry air to be compressed. The actual capacity handled by the compressor must be corrected for temperature and pressure differing from the standard conditions as well as the relative humidity. Changes in the previous example illustrate this.

A. Conditions

Inlet capacity, dry air at 60 °F, 14.7 psia, 75,000 cfm
Inlet temperature, 100 °F
Inlet pressure (and barometer), 14.7 psia
Relative humidity, 60%
Discharge pressure, 30 psig

B. Preliminary calculations

1. Weight flow dry air $W = \dfrac{QP_1 \times 144}{RT}$

$$= \frac{75{,}000 \times 14.7 \times 144}{53.3 \times 520}$$

$$= 5730 \text{ lb/minute}$$

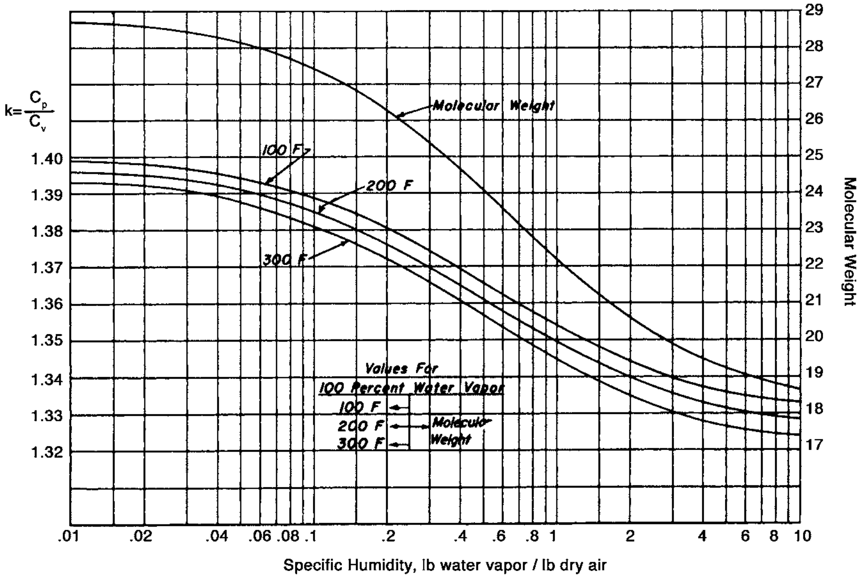

Figure 7.163 Molecular weight and adiabatic exponent for air vapor mixtures.

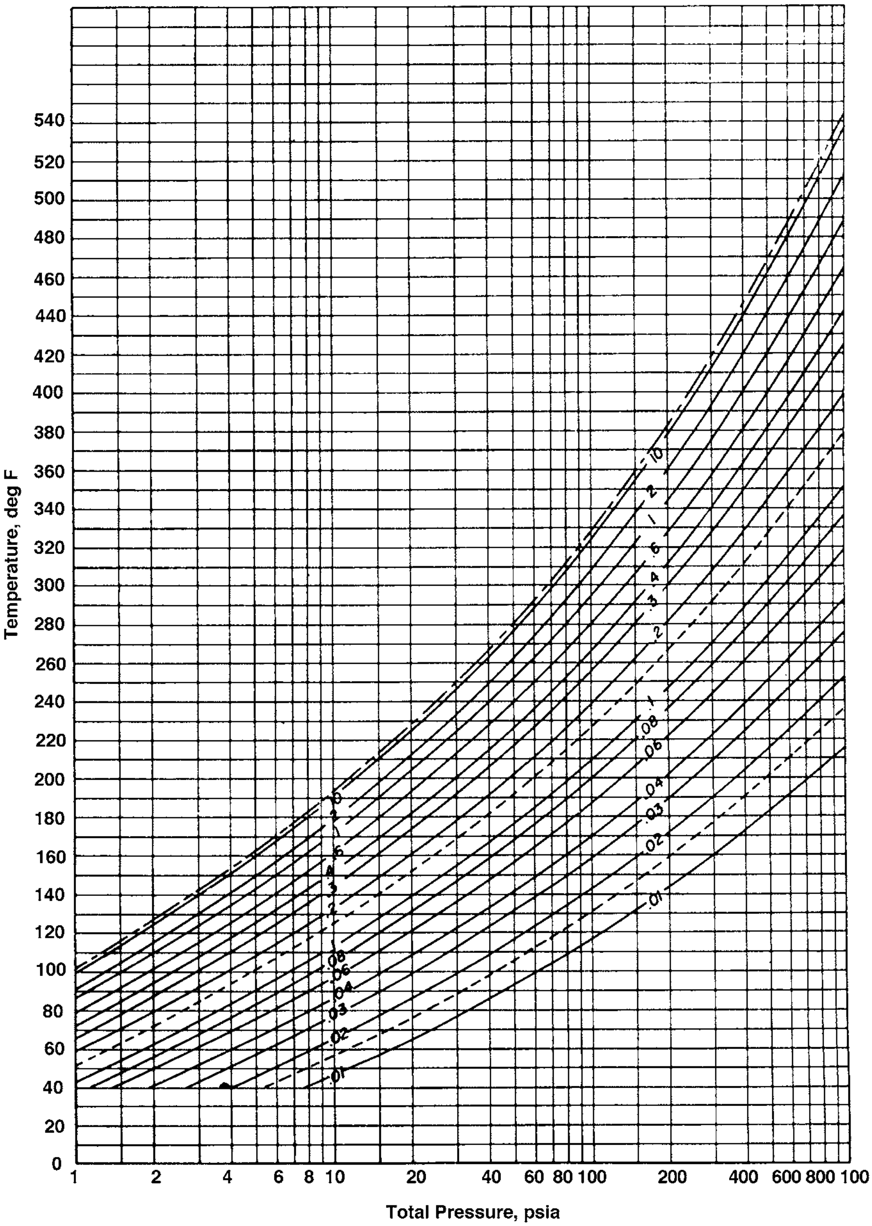

Figure 7.164 Pounds of vapor per pound of dry air at saturation.

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2. Partial pressure of water vapor
   a. Saturation pressure at 100 °F from steam tables:

   $p_{vs} = 0.949$ psia

   b. p_v at 60% RH = 0.60 x 0.949 = 0.569 psia
   (p_a = 14.7 – 0.569 = 14.131 psia)
3. Volume of air and water vapor mixture

   a. $Q_1 = \dfrac{5730 \text{ x } 53.3 \text{ x } 560}{14.131 \text{ x } 144}$ = 84,000 inlet cfm

   b. This may be checked by computing the weight of water vapor in the mixture, and then the volume occupied by that water vapor may he found from its partial pressure.

   $$W_2 = 0.622 \times 5730 \times \frac{0.569}{14.131} = 143 \text{ lb/minute}$$

   $$Q_1 = \frac{143 \times (1544/18.016) \times 560}{0.569 \times 144} = 84{,}000 \text{ inlet cfm}$$

4. Specific humidity = 143/5730 = 0.025 lb of water vapor/lb of dry air
5. Using Fig. 7.163 and specific humidity, one may determine k and MW of the mixture.

   k = 1.393 (at estimated T_{av} = 250°F)

   MW = 28.5 $\qquad R = \dfrac{1544}{MW} = 54.2$
6. Estimated polytropic efficiency from Figure 7.160 at 84,000 cfm is η_p = 75.8%.

7. $\dfrac{n-1}{n} = \dfrac{0.393}{1.393} \times \dfrac{1}{0.758} = 0.372$

8. $r = \dfrac{44.7}{14.7} = 3.04$

C. *Discharge Temperature*

$$T_2 = 560 \text{ x } 3.04^{0.372} = 560 \text{ X } 1.512$$
$$= 848R \ \ (= 388°\text{F})$$

D. Polytropic Head

$$H_p = \frac{RT_1\left[r^{(n-1)/n} - 1\right]}{(n-1)/n} = 560 \times 54.2 \times \frac{1.512 - 1}{0.372}$$
$$= 41{,}800 \text{ ft-lb/lb}$$

E.
Horsepower

1. $\text{ghp} = \dfrac{84{,}000 \text{ x } 14.7 \text{ x } (1.512\text{-}1)/0.372}{229 \text{ x } 0.758}$

   $= 9790$

2. Bhp = 9790 + 30 = 9820

Supercompressibility

Most gases show some departure from the simple perfect gas law when considered over a wide range of pressures and temperatures, and many gases show considerable differences even in a comparatively narrow range. If a satisfactory Mollier diagram is available, or, in other words, if all of the physical properties are shown for the particular gas in question, the Mollier diagram should be used to calculate real horsepower and other related items of performance as well as equivalent capacities at various temperatures and pressures. For gases the physical properties of which are only partially known, or not readily available, it is the usual practice to depend on specific information concerning the supercompressibility. A full discussion of this subject is found in Chapter 8 as it applies to all types of compressors.

Effect of Supercompressibility on Dynamic Compressor Calculations

Many gases normally handled in dynamic compressors cannot be treated with sufficient accuracy by perfect gas equations. Supercompressibility factors are shown in Figs. 8.1 to 8.6, and sample calculations involving Z factors appear on pages 649 to 656. For centrifugal compressor calculations of head and horsepower, it is reasonably accurate to assume that the calculations are affected directly by the average of inlet and discharge supercompressibility factors of the gas mixture,

$$\frac{Z_1 - Z_2}{2}$$

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Polytropic formulas for real gases take the form

$$H_p = \frac{Z_1 - Z_2}{2} \times R \times T_1 \times \frac{r^{(n-1)/n} - 1}{(n-1)/n}$$

$$\text{ghp} = \frac{Q_1 \times p_1 \times \left(\dfrac{Z_1 + Z_2}{2Z_1} \right) \times \dfrac{r^{(n-1)/n} - 1}{(n-1)/n}}{229 \times \eta_p}$$

The correction for supercompressibility in the horsepower formula differs from the head correction since inlet volume in the power formula has already been corrected for inlet supercompressibility. Since discharge supercompressibility is a function of discharge temperature, it will be necessary to estimate discharge temperature before calculating head and horsepower. It is to be understood that the final temperature of the gas after compression through a pressure range may be affected by the supercompressibility of the gas. For the purpose of estimating final temperatures necessary for the design of heat-exchanger apparatus, it is satisfactory to use figures based on the assumption that perfect gas laws may be used in most applications. This will result in figures that are slightly conservative. Sample compressor calculations for gas follow.

A. Gas: typical natural gas (Table 8.1)
Inlet pressure, 250 psia
Inlet temperature, 100 °F
Inlet capacity, measured at 14.7 psia and 60 °F, 102,000 scfm (cmh)
Discharge pressure, 593 psia

B. Preliminary Calculations

1. MW and k may be calculated from the gas analysis table, Table 8.1.

$$MW = 19.75 \qquad \left(R = \frac{1544}{MW} = 78.3 \right)$$

2. $r = \dfrac{p_2}{p_1} = \dfrac{593}{250} + 2.37$

3. Estimated discharge temperature:

$$\eta_p = 75\%$$

$$\frac{n-1}{n} = \frac{1.24-1}{1.24} \times \frac{1}{0.75} = 0.258$$

$$T_2 = 560 \times (2.37)^{0.258} = 560 \times 125$$

$$= 700°\text{F absolute or } 240°\text{F}$$

4. Inlet and discharge supercompressibility factors may be found using gas analysis, given conditions, and estimated discharge temperature (see Chapter 8 for calculations).

$$Z_1 \text{ at } 250 \text{ psia, } 100°\text{F} = 0.97$$
$$Z_2 \text{ at } 593 \text{ psia, } 240\ °\text{F} = 0.97$$

5. Weight flow:

$$W = \frac{102{,}000 \times 14.7 \times 144}{520 \times 78.3} = 5320 \text{ lb/minute}$$

6. Inlet capacity: $Q = \dfrac{WZ_1RT_1}{p_1 \times 144}$

$$Q_1 = \frac{5320 \times 0.97 \times 78.3 \times 560}{250 \times 144}$$

$$= 6270 \text{ inlet cfm}$$

7. Estimated polytropic efficiency (Fig. 7.160):
$$\eta p = 72.4\%$$

C. Discharge Temperature

$$T_2 = T_1\left[r^{(n-1)/n}\right] = 560\left(2.37^{0.268}\right) = 560 \times 1.26$$
$$= 706R = 246°\text{F}$$

D. Polytropic Head

$$H_p = Z_{av} RT_1 \frac{r^{(n-1)/n} - 1}{(n-1)/n} = 0.97 \times 78.3 \times 560 \times \frac{0.26}{0.268}$$
$$= 41.200 \text{ ft-lb/lb}$$

E. Horsepower

1. $$\text{ghp} = \frac{Q_1 \times p_1 \times \dfrac{Z_1 + Z_2}{2Z_1} \times \dfrac{r^{(n-1)/n} - 1}{(n-1)/n}}{229 \times 0.724}$$

$$= \frac{6270 \times 250 \times 1.0 \times (0.26/0.268)}{229 \times 0.724}$$

$$= 9170 \text{ hp}$$

2. Bhp = ghp + mechanical losses (bearings and seals)
   9170 + 30 + 20 = 9220hp

The conditions have been selected for the preceding example so that the weight flow in pounds per minute and head in foot-pounds per pound are almost the same as for the sample dry air calculation. This results in approximately the same horsepower for the two examples, despite the great differences in other conditions and gas characteristics.

INSTALLATION OF COMPRESSORS AND THEIR DRIVERS

General

Proper installation is the most important requisite for satisfactory operation of high-speed rotary machinery. Although such machinery will operate under somewhat adverse conditions, it is always advisable to provide proper operating conditions if the machinery is to give maximum reliability at minimum operating cost.

Plant Layout

Machinery foundations, piping, electric wiring, and all necessary auxiliary equipment must be carefully arranged in the plant layout. Even though space is limited, skillful placing of the equipment will result in better installation, more reliable operation of the equipment, and a minimum of maintenance. Machinery that is easy to install, operate, and maintain will generally get better attention from the operating and maintenance personnel.

In planning a plant layout, the following are some of the points which should be kept in mind.

Crane facilities. Heavy machinery can best be handled with overhead crane facilities. This is true not only during installation but also during periodic maintenance. If crane facilities cannot be provided, some other arrangement must be made for handling heavy units.

Space. Ample space should be provided to permit easy handling during erecting. Floor space should be provided in the vicinity of heavy machinery where the top half may be placed during periodic inspection of the rotating element and internal parts. The designer must note carefully and make provision for clearance limitations specified on the outline drawing.

Accessibility. Machinery should be installed where it is easily accessible for observation and maintenance. In elevated locations or in pits, there should be stairways, catwalks, and the like. An internal combustion engine, of course, must never be operated in a pit in which carbon monoxide could accumulate.

Operating convenience. The equipment should be arranged so as to provide maximum accessibility to parts that require observation or attention during operation. Auxiliary equipment, such as oil coolers, gage boards, and pumps, should be located where they do not interfere with the routine inspection of the equipment.

Cleanliness. Rotating equipment and auxiliaries should be installed in clean locations. Equipment must not be subjected to unnecessary hazards from dirt or moisture. Outdoor installations require particular attention, and such requirements should be carefully noted in the specifications so that proper provisions may be made in designing the machinery.

Piping and wiring. The equipment should be so located as to permit a minimum of piping and wiring, and compromise, if necessary, should be made only in favor of ease and convenience in operating the equipment.

Instrument location. Instruments, mounted on gage boards or elsewhere, should be located within easy view of the operator when starting the machine.

Storage. If machinery is to be stored for even a short period, it should be properly flushed to protect it against corrosion. Machinery should always be stored in a clean, dry place. This is particularly important with respect to electric motors. In case of doubt, the machinery manufacturer may be consulted regarding proper handling.

Foundations

While the foundation for high-speed rotating machines need not be as massive as that for reciprocating machines, it must be sufficient to provide a permanently rigid, nonwarping support for the machinery. Foundations should be properly designed to avoid possible resonance with running frequencies of the compressor and drive train. To meet these requirements, all conditions surrounding the foundation should be uniform; that is, the foundation should rest entirely on natural rock or entirely on solid earth, but never on a combination of both. Foundations supported on piling must have a heavy, continuous mat over the piling. A noncontinuous mat may settle unevenly and result in misalignment of the machinery.

The temperature surrounding a foundation should likewise be uniform. If one section of a foundation is subjected to a substantially lower temperature, (e.g., open windows or doors in cold weather) than another section, there can be possible distortion and resulting misalignment of the machinery. An outline drawing of a rotating machine may include a suggested foundation arrangement, but this should not be construed to be an actual design. To design a foundation effectively and arrange the piping properly requires an intimate knowledge of local conditions with which the machinery builder cannot be familiar. Moreover, these local conditions may present problems beyond the scope of the machinery builder's experience. The user

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must necessarily take full responsibility for an adequate foundation. In general, it is desirable to have the foundation designed by a competent structural designer who has had experience with foundations for heavy machinery.

The following are some of the points that should be kept in mind:

1. The foundation must provide a permanently rigid, nonwarping support for the machinery.
2. The foundation substructure should rest on a uniform footing, entirely on bedrock or entirely on solid earth.
3. The temperature surrounding the foundation should be substantially uniform. If it is variable, the variations should be essentially the same for the entire foundation.
4. If more than one machine is to be installed in a given location, each should have a completely independent foundation supported from bedrock or solid earth. The foundations should be entirely free from building walls or other parts of the building that might transmit resonant vibration. Operating platforms must also be isolated from the machinery foundations.
5. If the foundation substructure rests on bedrock and if it is imperative that no resonant vibration be transmitted to adjacent structures, a vibration-damping material should be interposed between the substructure and the bedrock.
6. Where foundations must be supported from floor beams, whether there be one or more machines so supported, a vibration-damping material should be interposed between the beams and each foundation.
7. Where the foundation substructure rests on piling, the piling should be covered with a heavy, continuous mat.

Adequate and satisfactory foundations may be made of reinforced concrete, structural steel, or combinations of the two. Masonry, brickwork, or concrete foundations are permissible if the equipment is set at substantially ground level on a firm support and where the foundation structure is primarily necessary to provide mass. (This is not always the case. Structural integrity or rigidity may be equally important, or even more important, including some cases where there is a potential vibration problem as when dowel pins are overlooked and a pedestal breaks free from a mat.)

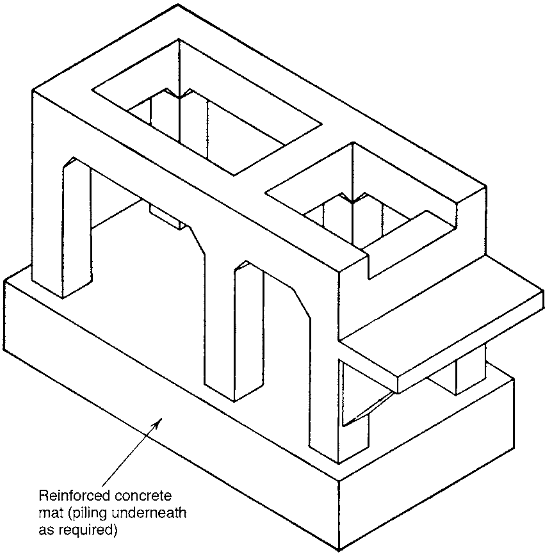

Figure 7.165 Typical reinforced concrete foundation suitable for any centrifugal compressor with either a base plate or soleplates.

Figure 7.165 shows a typical reinforced-concrete foundation structure suitable for rotating machinery of any size. This type of foundation structure permits the use of a base plate or soleplates, as preferred or dictated by the type of installation.

Figure 7.166 shows a structural-steel foundation with a continuous base plate spanning the six columns. This arrangement requires a substantial base plate. Each column must have a milled end pad, with all columns machined to equal length and with each end square with the axis of the column. The base plate is bolted and doweled to each column. Shims are normally provided between the top of the column and the bottom of the base plate to take care of slight inconsistencies. As an alternative, jack-screws may be provided at the bottom of each column to adjust the elevation of the base plate before grouting.

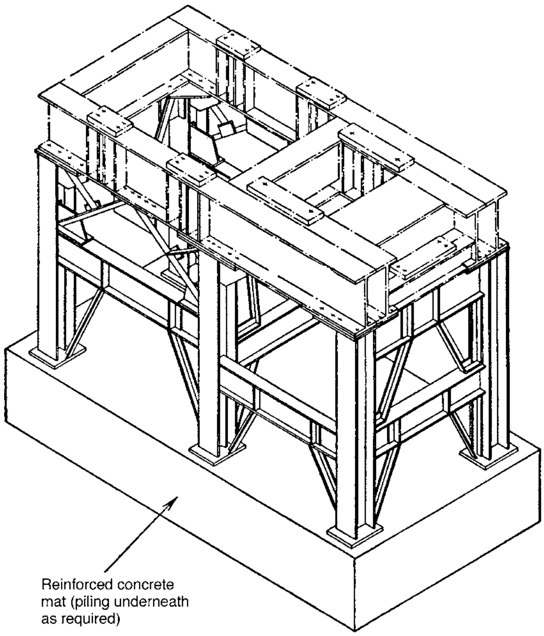

Figure 7.166 Typical structural steel foundation suitable for any centrifugal compressor having a substantial continuous base plate for spanning the six columns.

In addition to the cautions described, the following empirical design suggestions may be of interest:

1. The vertical loading may be calculated by adding not less than 50% to the total dead weight of the machinery.
2. The lateral loading may be taken as 25% of the total dead weight of the machinery, unless local earthquake requirements increase that figure.
3. The longitudinal loading may be taken as 10% of the total dead weight of the machinery.
4. All columns must be designed for the same unit stress to avoid distortion.
5. Center-line loading on vertical columns should be provided wherever possible. It is recognized that eccentric loading is unavoidable at times.
6. Total maximum vertical deflection of horizontal members should not exceed 0.015 in. for machinery operating at 3600 rpm. Deflection should be somewhat less for higher speeds and may be slightly more for lower speeds.
7. Concrete grout with a consistency barely permitting flow should be used for grouting soleplates or for getting base plates to reinforced-concrete structures.

8. If a structural-steel foundation includes a structural-steel deck on which the base plate is to be mounted, there may be need for metal grout to adjust for machining inconsistencies. In such cases, type metal with about 5 percent antimony may be used.

Piping

The reader is referred to Chapter 4 for a discussion of piping requirements, and particularly in Tables 4.5 through 4.9 and the related text.

TESTS

A number of different shop tests are performed on compressors in accordance with applicable standards. These cover hydrostatic, mechanical, and performance tests. The reader is also referred to Chapter 8 for further discussion of these tests.

Hydrostatic Tests

Compressor parts subject to pressure in operation are tested hydrostatically to 150% of the maximum pressure that can exist in the compressor or the casing section, or both, under the most severe operating conditions. Compressors having a maximum operating pressure not exceeding 50 psig are tested in accordance with manufacturer's standards. The compressor data sheet lists the actual hydrostatic test pressure used. The compressor parts subjected to hydrostatic test are maintained at test pressure for a minimum of 30 minutes to permit complete examination. Pressure vessels, filters, coolers, and so on, that are part of the lubricating and seal oil systems are hydrostatically tested in accordance with applicable codes or, where no codes apply, in accordance with manufacturer's standards.

Mechanical Tests

Each compressor is given a mechanical running test in accordance with manufacturer's standards to check general operation, vibration, and performance. Where practical, a turbine-driven compressor is short-time tested to 110% of the maximum continuous speed and at 100% for motor-driven units. Vibration measurements are taken throughout the operating speed range. Auxiliary equipment is tested in accordance with manufacturer's standards and, where practical, is used in the mechanical running test of the compressor. Noise-level tests, when required, should be made on the machine at the site and recorded in accordance with the *ISO 2151, Acoustics-Noise Test Code for Compressors and Vacuum Pumps – Engineering Method (Grade 2)*. Where practical, critical speeds of the compressor are determined during the running test. The compressor is designed so that its critical speeds shall not be detrimental to its satisfactory operation. The critical speeds of the driver shall be compatible with the critical speeds of the compressor, and the combination shall be suitable for the operating speed range.

Performance Tests

Each compressor is tested over its operating range to determine its performance. The test procedures used are in accordance with the latest edition of the ASME Power Test Code or any other applicable code that may be specified. Adequate readings are taken during the test, including discharge pressure, inlet pressure at compressor inlet flange, inlet temperature, relative humidity or ambient, discharge pressure at compressor discharge flange, compressor speeds, and brake horsepower. Readings are taken over the operating range and particularly at the specified operating point. The compressor manufacturer provides a test report, if specified, for each compressor showing the relative test data.

Witness Tests

Witness tests may be performed on any or all of the specific tests outlined previously.

INSPECTION AND MAINTENANCE OF DYNAMIC-TYPE COMPRESSORS

The purpose of an inspection and maintenance program for a piece of rotating machinery is to assure a higher unit availability. The investment in this program must begin with management decisions during the planning stages of a production facility and be continued throughout the life of the machine. Properly operating the compressor is an essential step in properly maintaining the unit, which is the responsibility of the user. Some of today's processes are very complex and impose undefined demands on the compressor in use. The maintenance program should be preplanned, but it should remain flexible enough to reflect additional needs as they develop throughout the operating life of the unit.

The design of a new compressor plant should go beyond the basic requirements of installation and operation. A well-engineered system must allow adequate space for dismantling, with provision for lifting and removal of assemblies and components to a laydown area. Employees must be available and be properly trained well in advance to ensure proper operation, as well as proper inspection and maintenance of the new unit. Proper tooling must be secured, along with spare parts and expendable maintenance materials in order to reduce the inspection time to a minimum.

Personnel

Certain skilled personnel are required to maintain an installation adequately. These personnel must be capable of:

1. Obtaining and analyzing maintenance data that will reflect the condition of the compressor before and after an inspection.
2. Scheduling the maintenance and inspection program based on operating data analyzing, including spare parts and materials.
3. Dismantling and reassembling the unit properly following the manufacturer's recommendations and adhering to sound mechanical practices.
4. Performing required nondestructive material tests, such as magnetic particle inspection and dye penetrant check to assure the soundness of critical components.

Maintenance Data

Specific data for maintenance evaluation should be included in the routine process data-monitoring system. Then these data should have guidelines established to call attention to a deteriorating performance level. Given the proper guidance, the plant operator can become an important link in the cycle of maintenance data evaluation. His or her alertness to changes in the data values will assist system maintenance supervision personnel in scheduling the proper corrective action. Some general maintenance data are common to all compressor installations, and these items can be used as a beginning for data collection. Additional data may be required if there are any unique process or application demands. The user of the equipment is generally in the best position to evaluate any process problems that may be experienced; therefore, the user must make the final choice on data procurement.

The following are examples of maintenance data that can be obtained by operating personnel and applied in the maintenance program. The listing is not a complete guide as objectives are different in each installation. However, once maintenance objectives are defined, the necessary data can be selected:

1. Load-bearing babbitt temperature from an embedded thermocouple or resistance temperature detector (RTD) can signal bearing distress by a gradual or sudden increase in temperature from previously established steady-state values.
2. Thrust bearing shoe temperature from a thermocouple or RTD embedded in the shoe near the babbitt surface can signal significant changes in thrust loading. Changes in thrust loading may come from internal or external causes. Damage to the compressor balance piston seals that reduces their effectiveness could reflect further damage within the compressor rotating element. Increased coupling friction caused by deteriorating coupling

teeth or a lack of lubricant can impose thrust loads high enough to exceed a thrust bearing capability;

3. Gas intercooler temperatures will reflect deteriorating cooler performance due to fouling from cooling water or process gas. The cooler approach temperature should not change over a period of time for a given set of cooling water and gas flow conditions.
4. The pressure differential across the lubricating system oil filter is directly related to the foreign material collected on the filter element. Foreign material can be forced through some elements if the differential pressure exceeds a safe limit.

The operator's contribution to the success of the maintenance program is not limited to written data. The operator can be of invaluable service by developing an alert sense toward noises and a keen eye for abnormal conditions. For instance, sound level or pitch changes are often forecasters of impending damage. Loose bolts on a stationary part can be retightened, if caught in time, to keep the part or bolt from falling into a rotating member.

Certain periodic data must be obtained by specially trained personnel for complete maintenance data evaluation. Here, requirements for special equipment and skills may direct the use of outside consultant for economic reasons. Regardless, the importance of this type of data to the successful program cannot be overestimated. The following are examples of this type of data.

Vibration Analysis

A meaningful vibration analysis must be accomplished with a good-quality analyzer. The instrument capability should be broad enough to include all vibration components from the lowest to the highest possible excitation frequency. Where speed increasers are employed, gear tooth mesh frequencies should also be considered in the spectrum. The analyzer should have provisions to determine the phase relationship of any unbalance for correction purposes. Vibration analysis is an effective tool in identifying the cause of unbalance in rotating elements, for example, erosion or unequal buildup of foreign material from the process gas. Many impending mechanical problems can be identified in their early stages by the keen vibration analyst. Loose rotating element components, misalignment, and coupling lockup are but a few that can be mentioned. Certain dynamic problems are not necessarily brought to light by vibration analysis, so this technique should not be used solely, to the exclusion of other techniques. Flexure at the hub of a large rotor due to the gyroscopic effect of the rotor is one example. Certain other vibrations have sudden onset, with little or no warning.

Lubricating and Seal System Oil Analysis

Oil contamination due to water, foreign particles, or reactions with the product gas have been blamed for many system failures. Early detection of this condition can avoid later failures. Samples of the lubricant should be taken frequently from the reservoir and thoroughly analyzed. The oil supplier usually furnishes a complete analysis service for any collected samples. He must be considered the authority on the proper lubricant additives for the particular application.

Mechanical Alignment of Components

Mechanical alignment of the entire compressor train must be periodically verified at the normal operating conditions of the compressor. Repeatable, accurate measurements must be taken to determine the actual thermal growth of each component in the train. The measurements must be corrected to reflect the actual movement of each coupling. Ambient temperature and process data should be recorded for future reference. Alignment corrections must be made where initial cold alignment offset does not fully compensate for actual thermal growth. The periodic alignment check should include a thorough inspection of all piping connected to the compressor for possible strains on the nozzles that could cause distortion or loss of alignment.

Cooling-water Chemical Analysis

Proper chemical treatment of cooling water can control the rate of fouling of heat-exchanger tubes. The appropriate treatment must be determined by a chemical analysis.

Compressor Performance Evaluation

Deteriorating compressor performance can be observed by periodic recording of key process data. A reduction in capacity may indicate material buildup within the rotating element or some internal damage. To be significant, comparative data should be obtained at the same process conditions. Some calculated corrections can be made for minor deviations from the original test conditions in the final comparison of performance.

The importance of proper maintenance data collection and analysis immediately after a unit inspection cannot be overemphasized. The post-inspection data will reveal the effectiveness of the work accomplished and set new criteria for future maintenance data evaluation.

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Scheduling

The maintenance inspection timetable must be tailored to each individual installation to account for any unique process demands. The object is to inspect and correct abnormalities in the compressor before serious equipment damage occurs. The original schedule to meet this objective may be altered by experience and plant economics. For instance, the cost of downtime along with the reconditioning costs versus longer operating periods with possible increased reconditioning costs must be considered. It may be more economical to schedule inspections farther apart and replace major assemblies rather than smaller components more frequently. For example, the actual outage time span can be greatly reduced if a complete spare rotating element is on hand. Replacing it rather than reconditioning the existing element during the inspection may be preferable. The used element can probably be economically reconditioned to design specifications and returned to storage for the next scheduled outage. A further advantage to the major component replacement system is that it reduces the strain on repair shop facilities during the outage time.

The inspection schedule will also determine the type of spare parts, tooling, and special personnel required. The manufacturer's spare-parts list should always be considered as a guide for stock levels, tempered by the unique process demands and the objective of the inspection program. The alert planner will make use of his or her operating experience with a certain unit and begin developing packaged kits of parts including such small items as lock rings, O-rings, and gaskets. Frequently, it is the small item that extends the outage time. In a time of high labor costs, it may prove far more economical to expand the spare-part inventory on certain items when costs of repair and extended outage time are subtracted from the cost of the parts.

Dismantling, Inspection and Reassembling

The compressor manufacturer's recommended dismantling and reassembly procedures should always be followed closely. It is a wise investment to use a qualified serviceperson from the manufacturer's service organization for technical supervision during the inspection. It is the responsibility of the user, however, to properly supervise employees and assure the quality of the work required for a successful maintenance program.

It may not be economically feasible for all users of dynamic-type compressors to support a maintenance organization capable of accomplishing their maintenance objectives. Here, the manager must evaluate outside services that are available. Some manufacturers have well-established, factory-certified shops in local areas with complete capability and experience in servicing their compressors. Some have field service groups that can provide in-plant maintenance service. Factory design engineering services including field engineering services are also available from some manufacturers. Factory repair service also provides certain other advantages, since stock parts and special materials are readily available along with special engineering services and test facilities.

Accurate records and photographs of equipment in the as-found condition are invaluable tools in future inspection scheduling. The location of all damaged parts, internal rubs, erosion, or corrosion should be accurately recorded. Accurate records of all measurements on each component of the assembly should be tabulated each time the compressor is inspected. The first entry should always be the design-specified value for comparison with later measurements so that the wear trend can be easily evaluated. Every spare part received from the factory should be correctly identified and dimensionally checked, or it will not have been properly prepared for storage. The agonizing disappointment of removing a critical part from storage and finding it damaged from improper storage, or dimensionally incorrect, will eventually be experienced by the too casual storekeeper.

Accurate set-back measurements should always be recorded upon disassembly for use as checkpoints during reassembly. Field measurements should always be compared with the manufacturer's specified values. All internal clearances should be restored within the specified tolerance to assure the design performance level after inspection.

Proper cleaning and protection of parts is extremely important to prevent damage to critical areas. For example, the mechanical balance of a rotating element can be seriously impaired by improper cleaning methods, thereby jeopardizing the success of the entire inspection and maintenance program.

High operating speeds required in dynamic compressors demand extremely close tolerance and high-quality finishes in certain critical areas. Bearing journals and seal surfaces are ground to a superfinish to ensure proper seal and bearing life. Shaft fillets are highly polished to reduce stress concentration in critical areas. These surfaces must be protected during the dismantling and reassembly of a unit to prevent serious consequences later in the life of the compressor.

Nondestructive Material Tests

Highly stressed parts must undergo proper nondestructive material tests after cleaning to ensure their mechanical soundness. The use of one of the proven inspection methods such as dye penetrant, wet magnaglo, or magnaflux is recommended. Items such as impellers, blading, gear teeth in speed increasers, coupling hubs and sleeves, shaft keyways and fillets, and shrunk-on sleeves are examples of components that may require material tests.

Corrective Engineering

During the early life of the compressor, certain components may require attention more frequently than the rest of the assembly. Frequent part replacement may be due to a process demand that was undefined during the compressor design stage. The troublesome part can usually be redesigned to better withstand the demands imposed on it, thereby increasing its life expectancy in line with the remainder of the assembly. This maintenance step may involve the manufacturer's design engineer, who becomes a part of a corrective engineering team along with the maintenance engineer and operating engineer. The user's investment in corrective engineering will be small in comparison to the increased reliability and production throughout the life of the compressor.

Maintenance Through Operation

The intricacy of a compressor installation is usually dictated by the process demands. The equipment operator's responsibility also increases with the system complexity. For instance, an installation handling a dangerous gas would require a more elaborate seal system than a unit handling air. The operator must have a thorough knowledge of the system's limitations. It is the operator's responsibility to be thoroughly familiar with the design intent for each compressor unit and adhere to sound operating practices. The manufacturer's instruction book is available for this purpose as well as for instructing the maintenance organization in the upkeep of the unit. The operator should be familiar with the unit operating requirements well in advance of start-up. Casual or unplanned operating practice can never be justified and will eventually lead to trouble. Compressor reliability is linked as closely to good operating practice as to good maintenance procedures.

Instrumentation and Protective Devices

Instrumentation and protective devices frequently represent a significant share of the initial investment in a new compressor system. The importance of this equipment cannot be overemphasized. However, it is only as dependable as its calibration. Maintenance of these devices is usually the responsibility of a separate group from mechanical maintenance and, therefore, they are frequently neglected. The simple pressure gage is a valuable source of data when reliable, but becomes worthless if its calibration cannot be trusted. Instruments must be recalibrated at regular intervals and checked whenever calibration is doubtful. The recalibration and testing of instruments should be scheduled along with other compressor maintenance.

Protective devices used for alarm or trip functions must be kept in calibration and properly tested to ensure their functional reliability. Each of these devices should have a test arrangement to prove its integrity with the compressor on stream and without risking an accidental shutdown.

Many costly plant outages could have been avoided in the past if proper attention had been given to the protective devices on a piece of rotating machinery.

Preventive Maintenance

There is no single, established approach to an effective maintenance program for every compressor installation. The importance of tailoring the maintenance program objectives to each unique compressor installation has already been emphasized. The following preventive maintenance schedule is not intended as a complete guide, but only to suggest how to devise a plan for the particular installation in question. The items selected and the schedule itself must be tailored to the installation and carried out with common sense and good judgement.

A. Daily Operator Maintenance Instructions

1. Check the lubricating oil and seal oil reservoirs for possible water accumulation by draining a small sample from the reservoir low-point drains.
2. Check the lubricating and seal system oil filters for excessive pressure differential.
3. Verify that the oil levels in seals and lubricating oil reservoirs are within a safe operating range.
4. Check the operation of all process cooler and separator traps and seal system sour oil traps by observing the liquid levels in sight glasses or blowing off trap bypass valves to drain funnels.
5. Review all supervisory and process instruments such as those indicating oil pressures and temperatures, vibration, process pressures, and temperatures to ascertain that no unexplained deviations have occurred.
6. Listen for noise level and pitch changes around compressors, gears, and drivers.
7. Inspect visually for oil, gas, or water leaks and loose parts.
8. Check the differential pressures across intake gas filters, intercoolers, aftercoolers, and interstage separators for excessive differential that could signal plugging or other deterioration.
9. Observe the level in the seal oil drain sight flows and the bearing oil drain sight flows for abnormal level changes and check regularly to establish that the connections have not become clogged.

B. Weekly Operator Maintenance

1. Verify the calibration and operation of all protective alarm and trip devices through actual test. Test lockout arrangements should be provided on each device to allow safe test with the compressor on stream.

C. Monthly Maintenance

1. Make a vibration survey of each bearing housing, including shaft readings where possible. The data at each location should include unfiltered or total wave and rotational or filtered wave at rotating frequency components. If these two values do not agree, a thorough investigation, including a frequency search, should be made to determine the cause. Any significant increase in vibration should be

noted and corrected at the earliest possible moment to prevent permanent bearing or compressor damage.

2. Test the performance of all intercoolers, aftercoolers, and oil coolers to evaluate their efficiency. The rate of deterioration will determine the cleaning schedule.
3. Lubricate the linkage, pins, and slide bars of all control valves and valve positioners or guide vane positioners.
4. Obtain oil samples from lubricating and seal oil reservoirs for analysis by the lubricant supplier.

D. Major Maintenance Shutdown

1. Coupling Inspections

   Gear Type

   a. Dismantle, and remove all grease, taking note of the condition of the grease. If significant separation has occurred, the lubricant supplier should be consulted for further grease recommendations. The coupling grease should always be checked within the first month of initial operation of a new unit to verify that the lubricant has not separated or otherwise deteriorated. This practice should be followed until a lubricant is selected. The amount of sludge buildup in continuous oil-lubricated couplings may indicate the need for better or additional oil filtration at the coupling spray nozzles. Check the spray nozzle pattern on reassembly.
   b. Clean all hubs and sleeves thoroughly and inspect gear teeth for abnormal wear and broken or cracked teeth. The hub and sleeve teeth and hub keyway should be given a thorough magnetic particle or dye check inspection for evidence of cracks.
   c. Repack and replace the proper type and amount of grease using new gaskets or O-rings, where applicable. Follow the coupling supplier's recommended bolt torque values and bolt tightening sequence.

   Non-lubricated Type

   a. Inspect the flexible hinge member for cracked or damaged disks in the disk pack or diaphragm.
   b. Inspect coupling hubs and spacers at all high stress points for cracks using the magnetic particle or dye check method.
2. Verify the Alignment of All Couplings
   a. The cold offset alignment should be verified by actual thermal growth measurements at operating conditions. This procedure should be repeated whenever the process conditions are changed significantly. The cold offset may require changes due to foundation settling or shifting.

b. The ambient temperature and the location of the dial indicator, driver, or driven shaft must be recorded on all alignment records for future reference.
c. A 12-inch diameter face plate temporarily bolted to each coupling hub will assist greatly in improving angular alignment accuracy on small-diameter, high-speed couplings.

3. Clean and Inspect All Journal Bearings
   a Remove and inspect each bearing for signs of babbitt damage.
   b. Measure and record the bearing-to-shaft clearance following the manufacturer's recommended procedure. Replace any bearings found to have clearances exceeding the manufacturer's specifications.
   c. Inspect the shoe-to-retainer contact point on all pivoted pad bearings for signs of fretting or excessive wear that could hamper the shoe pivot freedom.
   d. Thermocouple or RTD detector lead wires embedded in shoes must be installed to allow complete freedom of shoe movement.
   e. Inspect bearing seal rings, if used. Garter springs or other retainer springs should be replaced if weakened by worn spots.
4. Clean and Inspect Each Thrust Bearing Assembly
   Note: Record an axial set-back measurement to locate the rotating element in the casing before disturbing the thrust bearing. Check the manufacturer's drawing for reference dimensions. Always keep forward and rear thrust assemblies and related shims separated. These assemblies must not be interchanged.
   a. Inspect shoes for signs of loose or damaged babbitt.
   b. Inspect hardened contact buttons on self-aligning bearing shoe backs and leveling plates for flat spots or fretting. Slight wear spots should be removed by light stoning to restore the button crown or leveling plate curvatures.
   c. Check the thrust collar radial and face runout if a removable collar is used. Carefully follow the manufacturer's instructions for reassembly of the thrust collar if it is removed.
   d. Readjust the thrust bearing clearance to the proper value.
   Note: Forward and rear thrust shims must be adjusted together to keep the rotating element axial position in its correct relationship with the casing.
5. Inspect all oil baffles for signs of rub, plugged drain-back holes, or chipped touch points. Reinstall with specified radial and axial clearances.
6. Remove the casing cover for an internal inspection if performance level or maintenance schedule dictates this step.
   a. Properly clean and inspect all impellers or blading for erosion or corrosion. All highly stressed parts should be given a magnetic particle inspection or dye checked for cracks. Casing diffusers or stator blading should be cleaned and inspected to meet the same requirements as the rotating speed.

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b. Interstage seals should be replaced if damaged or eroded to such an extent as to allow clearances to exceed specified values.
c. Rotating elements should be checked for balance if increase in vibration has been noted. The cause of any such lack of balance should be in-vestigated, such as loss of a nut or accumulation of sludge.

7. Lubrication and Seal Systems
    a. Drain and clean each system reservoir. Use squeegees or synthetic sponges for internal cleaning. No rags of any type should be used, since lint may get into the lubricating and seal systems.
    b. Centrifuge the oil before returning it to the reservoir or replace it with new oil. Replacement oil should be carefully strained to keep foreign material that may be present in the oil drums out of the system.
    c. Clean or replace lubricating and seal oil supply filters.
    d. Remove tube bundles from oil coolers and thoroughly clean the water and oil sides. Test the tube bundle of the cooler hydrostatically to its rated pressure before reassembling the cooler.
    e. Inspect all system controls and regulating valves for foreign material, sticking pistons or valve stems, and so on.
    f. Inspect the lubricating and seal oil pumps for abnormal wear at pump element bearings, shaft seals, and couplings.
8. Process Check Valves
    a. Remove and inspect each valve for wear at hinge pins, disk guide pins, disk return springs, and seals. All worn parts should be replaced.
9. Process Expansion Joints
    a. Inspect the internal bellows surface for pitting and erosion that could lead to failure. If lined, inspect the liner.
    b. Adjust pipe support hangers, if required, to position the expansion joint properly in its cold position.
10. Speed Increasers
    a. Inspect pinion and gear bearings for damage or wear and proper clearance. Carefully check the bottom of the gear base for metallic particles.
    b. Check the gear tooth contact pattern. Check gear teeth for signs of abnormal tooth wear and unequal tooth loading.
    c. Gear and pinion teeth should be subjected to thorough inspection by magnetic particle process or dye check.
    d. Clean all mesh spray nozzles and all internal oil supply passages. Make sure that the nozzles are properly secured on reassembly.
    e. Inspect all splash pans for possible fatigue cracks or loose fasteners.

11. Instrumentation and Protective Devices
    a. Check the calibration of all instruments used for monitoring operations or for obtaining maintenance data.
    b. Check the calibration and functioning of all alarm and trip devices.
12. Main Driver
    a. Follow the driver manufacturer's recommended inspection and Introduction to Gas Drying

INTRODUCTION TO GAS DRYING

Reference to Similarities to Air Drying

Compressed air is the most utilitarian compressed gas and, as such, has received the most attention of all the gases. Purification techniques have been perfected and specifications have been clearly defined for compressed air. Although other gases are seldom employed as a utility as compressed air is, there is a definite need for them to be dried. These gases are often raw materials for a process and must be reduced in water content to meet process requirements. An example is shown in Fig. 7.167. In addition, water, especially in liquid form, can present the same problems during the transmission of gas to its point of use in a process system, as would be expected in a compressed-air system.

The drying of compressed gases other than air may require specialized attention due to the particular properties of the gas. The technology, however, and, in fact, the equipment are often quite similar to that used for drying compressed air.

Most gases, unless the system temperature and pressure conditions are unusually high or low, behave like ideal gases, having the same relationship for volume, pressure, temperature, and molecular weight. For this reason, air and many other gases can be treated with the same type of equipment to reduce the water content.

Although the equipment used for drying air and other gases may be similar and utilize the same operating mechanisms, the size, choice of materials, and other specifications may be decidedly different because of the specific properties of the gases. These properties include specific gravity, specific heat, viscosity, thermal conductivity, explosive characteristics, toxicity, corrosion, and others. Also, in some cases a particular gas may have an effect on certain materials of construction and even on the properties of the water contained in the system because of their solubility in liquid water or their chemical reaction with water. These and other properties of a particular gas may have to be considered in the design of gas-drying equipment.

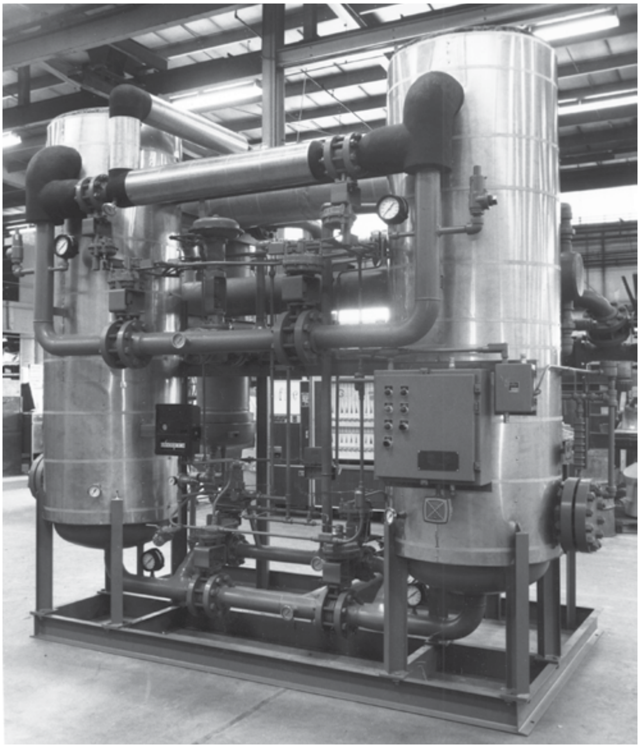

Figure 7.167 A fuel gas dryer for a petrochemical plant.

It is the purpose of this chapter to relate the drying of compressed air to the drying of other compressed gases and to define those areas of difference that require special consideration.

DESIGN CONSIDERATIONS

Contaminant Filtration, Separation, and Disposal

In any properly designed drying application, care should be taken to provide for contaminant filtration, if possible, to help achieve the dew point performance required. A contaminant can be considered as any component of the gas stream that, if not removed, will affect the dryer's performance on either a short- or long-term basis.

Water, in the form of liquid drops, droplets, aerosols or fog, is perhaps the most common contaminant. Dryers are designed to take out or remove water only as true water vapor as it can exist at the specific operating pressures, temperatures, and flow. Proper removal by means of knockout drums, centrifugal separators and coalescers is essential to good drying practices.

When the compressed gas contains oil vapors, aerosols, or condensed liquids, whether from a compressor or, in the case of some of the hydrocarbon gases, as a natural constituent, these should also be removed to minimize contamination of the dryer's desiccant. This is usually accomplished by means of scrubbers, coalescing filters filtering to 0.3 microns or smaller. As in water removal, any liquid droplets or aerosols are best removed by a separator, scrubber, and coalescing filter.

Other contaminants can affect drying performance. These can be constituents of the gas stream that compete with water vapor for its place in the desiccant. Other contaminants can plate out or coat the desiccant so that water vapor cannot be adsorbed into the desiccant's pores or capillaries. Proper selection of the desiccant, as well as the method of regeneration, can usually correct or minimize this condition.

Disposal of contaminants should be made in accordance with local Department of Environmental Resources (DER) requirements, and it is the customer's responsibility to conform to other local regulations as well. Condensate in the form of water can be sent to a drain. Oil condensates should be disposed of by methods prescribed locally, and other contaminants can possibly be disposed of by sending them to a flare or to a hazardous-waste-disposal system.

Design Pressures

Design pressures for gas-drying equipment should follow the ASME Pressure Vessel Code, Section VIII, or Section m if Nuclear Code is required. If the gas being handled is a lethal gas, the ultimate customer must advise the dryer manufacturer if the vessel is to be manufactured to the special ASME requirements for lethal materials. This is the customer's choice, and it depends on the application and insurance company requirements.

Special Materials

Construction materials for units drying gases other than air should be selected as applicable to the particular situation or as requested by the customer. These requirements may include anything from standard screwed brass valves and malleable iron fittings to steel valves, socket weld fittings, or stainless-steel construction through-out. Familiarity with the application will help in the initial quoting stages so that the proper type of construction will be used.

Control Requirements

As with air dryers, the control of gas dryers is largely a question of customer preference as to whether the dryer is to be manual or automatic and whether it is to be controlled by moisture load, time interval, or both. The location of the equipment will dictate whether it must be suitable for indoors, outdoors, hazardous, or explosion-proof locations or a combination of these.

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Leakage Considerations

Particular care should be given to leakage requirements, which may be quite stringent in the case of gases such as hydrogen, helium, natural gas, and toxic or radioactive gases. Selecting the proper safety valve for a pressure vessel is also extremely important, taking into account whether the valve must be gas tight or whether its discharge must be piped to a safe location.

Purge and Purge Exhaust Disposal

Disposal of effluent or purge gas from dehydration systems requires that water be disposed of by effluent or gas purging. Since dehydrated gas may contain noxious or toxic contaminants, these contaminants may be in the effluent from a refrigeration or deliquescent system or in the purge gas of a regenerative system. The type of contaminant will determine what is required by the user to comply with plant, DER, and OSHA requirements. In many cases, the effluent or purge gas contains no contaminants. In such cases, the effluent can be disposed of through a regular sewage system. The purge gas, if valuable, can be recovered by introducing it into the intake side of the compressor once the liberated water is condensed and removed.

Specifications for Gas Quality

Gas quality is dependent on the customer's needs and the ultimate use of the product. Quality should be specified by the customer as to dryness required, what undesirable components must be removed, and what degree of filtration is necessary for desiccant dust removal.

TYPES OF DRYERS FOR COMPRESSED-GAS SERVICE

All types of dryers currently manufactured can be used in some phase of compressed-gas service. The following outlines the use of deliquescent dryers, refrigerant dryers, and adsorption desiccant dryers in compressed-gas industries.

Deliquescent Dryers

A deliquescent dryer can be used in situations where contaminants in the gas stream may be quite high and in hazardous areas where remote operation without electricity may be necessary. Normally, these dryers must be operated at inlet temperatures of 100°F or less. Deliquescent dryers consume no power while operating, do not use a portion of gas for purging, are capable of handling light and heavy hydrocarbons in the gas stream, and do not react with most contaminants. Outlet pressure dew point compensates for varying inlet temperature and pressure conditions. Their only moving parts are those required to discharge effluent. Since deliquescent dryers use a consumable material, periodic addition or replacement of the desiccant is required. Desiccant consumption is high, and dew point depression is minimal at higher inlet temperatures. Environmental conditions or restrictions must be considered, as with other types, in the removal of the solutions drained from the dryer.

Refrigerant Dryers

Refrigerant dryers can be used in compressed-gas service for applications that do not require pressure dew points below 35 to 40°F. Also, refrigerant dryers are often used for bulk water removal from many industrial gases, such as argon, nitrogen, and dissociated ammonia. Air-separation plants utilize refrigeration of the compressed gases to lower the energy consumption of the dual-tower drying system by removing nearly 70 or 80% of the compressed-gas water content.

Refrigerant dryers are also used to condense out heavy hydrocarbon gases such as butane, pentane, and hexane from natural gas without gas compression. To accomplish this, normally a deliquescent, regenerative, or glycol unit must be installed prior to the refrigeration unit to remove water vapor that otherwise could condense out also.

Adsorption Desiccant Dryers

Heat-reactivated dryers. Heat-reactivated, adsorption desiccant dryers are often used for drying almost any of the common gases, such as carbon monoxide, carbon dioxide, argon, oxygen, and nitrogen, as well as hydrocarbon gases such as ethylene, ethane, methane, and natural gases. Most of the above drying applications require an extremely low water content of 1 to 25 ppm by weight. The most common heat-reactivated compressed-gas dryers are discussed next.

Internal Heat-reactivated Dryers. Regeneration of this type of heat-reactivated dryer is accomplished by energizing the heaters and purging, either with part of the gas being dried or by dry air or dry inert gas introduced from an external source. With some gases, purge exhaust disposal must be considered because of safety and environmental restrictions. These internally steam- or electric-heated dual-tower Adsorption desiccant dryers are commonly used for flow rates ranging up to 200 scfm.

External Heat-reactivated Dryers. This type of dryer reactivates the wet tower by using a blower for recirculating a captive volume of the gas in a closed-loop circuit, thus eliminating purge losses. The reactivation heat is supplied by an external steam or electric heater. The heated gas, 350 to 500°F, enters the dryer tower, driving off the adsorbed moisture. The liberated moisture is then carried out of the bed into the water-cooled condenser where the water vapor is condensed and the liquid water is separated. At the end of the reactivation heating period, the desiccant is cooled down to approximately 125°F prior to tower switchover.

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This type of dryer provides pressure dew points of as low as - 40 to - 100°F and reduces the dew point and temperature elevation spikes commonly experienced with internal heat reactivated dryers.

Split Stream Steam or Electric Heat-reactivated Dryers with an External Steam or Electric Heater. In the split-stream type, a throttling valve diverts some gas from the inlet side of the dryer. This gas is heated and then it strips moisture from the adsorbent during the regeneration. The gas is then cooled, water condenses, and is drained from the unit. The gas is then returned to the inlet without purge losses. Split-stream reactivation is commonly used to dry natural gas and many other hydrocarbon gases or other precious gases that cannot be purged to atmosphere. The distinct advantage of the split-stream reactivation versus a closed-loop reactivation is obviously the absence of the reactivation blower; however; pressure loss is incurred to operate the throttling valve and the regeneration system.

Heatless dryers. Heatless dryers are rarely used to dry compressed gases since a large amount of the gas being processed must be purged during the reactivation. It should be pointed out that heatless dryers can be utilized for dry, explosive, flammable, and precious gases, as long as a means of recovering the gas and returning it to the intake side of a compressor is provided. Perhaps better and more efficient compressed-gas drying can be accomplished by utilizing a closed-loop or a split-stream reactivation drying system.

Glycol dehydration. A common method of drying natural gas is glycol dehydration. Glycol dryers can obtain pressure dew points as low as - 40°F. They are generally used where the pressure of the gas is above 100 psig and inlet temperatures are below 120°F. The glycol dryer can he used where no electrical power is available, although a combustible gas must be used to heat the boiler.

APPLICATIONS FOR COMPRESSED-GAS SERVICE

The main reason for removing water from a gas is to limit the adverse effects it has on a system or to aid in its transportation. These adverse effects can be physical, chemical, or electrical, or a combination of these. The drying of a gas, whether at atmospheric or higher pressures, is done for the following reasons:

1. To control the relative humidity in industrial processing and in drying material at limited temperatures.
2. To control dew point and prevent the precipitation of water.
3. To minimize the chemical and corrosive effect of water.
4. To reduce the weight of gas for compression and transportation purposes.

Many industrial gases can be dried by adsorption; a partial list follows:

Acetylene	Freon	Natural gas
Ammonia	Furnace Gas	Nitrogen
Argon	Helium	Oxygen
Carbon dioxide	Hexafluoride	Propane
Chlorine	Hydrogen	Propylene
Cracked gas	Hydrogen chloride	Sulfur dioxide
Ethane	Hydrogen sulfide	Sulfur hexafluoride
Ethylene	Methane	

The degree of dryness and the type of desiccant dryer to be used depend on the application.

Natural Gas Drying

Some natural gas wells furnish gas of very high purity, that is, gas that is almost pure methane. However, most hydrocarbon streams are complex mixtures of hundreds of different compounds. A typical well stream is a high-velocity, turbulent, constantly expanding mixture of gases and hydrocarbons intimately mixed with water vapor, free water, solids, and other contaminants.

Contaminant-removal processes can be divided into two groups: dehydration and purification. The principal reasons for the importance of natural gas dehydration include the following:

1. Liquid water and natural gas can form solid, icelike hydrates that can plug valves, piping, and other parts of the system.
2. Natural gas containing liquid water is corrosive, particularly if it contains CO_2 or H_2S.
3. Water vapor in natural gas pipelines may condense, causing slugging flow conditions.
4. Water vapor increases the volume and decreases the heating value of natural gas, thus leading to reduced line capacity.
5. Dehydration of natural gas prior to cryogenic processing is an absolute requirement to prevent ice formation on low-temperature heat exchangers.
6. To prevent freezing after regulation.

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Of these, the most common reason for dehydration is the prevention of hydrate formation in gas pipelines.

Natural gas hydrates are solid crystalline compounds formed by the chemical combination of natural gas and water under pressure at temperatures considerably above the freezing point of water. In the presence of free water, hydrates will form when the temperature is below a certain point, called the hydrate temperature.

Hydrate formation is often confused with condensation, and the distinction between the two must be clearly understood. Condensation of water from natural gas under pressure occurs when the temperature is at or below the dew point at that pressure. Free water obtained under such conditions is essential to the formation of hydrates, which will occur at or below the hydrate temperature at the same pressure.

During the flow of natural gas, it is necessary to avoid conditions that promote the formation of hydrates. This is essential since hydrates may choke the flow stream, surface lines, and other equipment. Hydrate formation in the flow stream results in a lower value for measured wellhead pressures. In a flow-rate measuring device, hydrate formation results in lower apparent flow rates. Excessive hydrate formation may also completely block flow lines and surface equipment. Thus, the need to prevent hydrate formation is obvious. The easiest way to eliminate hydrate formation is to substantially remove the water from the natural gas stream.

Another important application for desiccant drying is the liquefaction of natural gas. Methane is converted into a liquid in a cryogenic process at - 285°F and atmospheric pressure. There is a 600 to 1 reduction in volume. As a liquid, large volumes of methane can he easily transported and stored. Natural gas companies liquefy and store gas up to 20 million scfd when demand is low and use the stored liquid during periods of high demand. Natural gas found in remote areas can be liquefied and transported to places of demand. Because of the low dew points needed for the cryogenic production of LNG, molecular sieve dryers are used.

Refrigerated gas dryers are also used for the dehumidification of fuel gases, including landfill gas, digester gas, methane and natural gas. Drying to pressure dew points of plus 35 to 40°F will improve the heat value efficiency of the boilers and complete burning of the fuel gases. Dry gas also helps to meet the air quality requirements from the combustion exhaust gases. Dry fuel gas in an internal combustion engine improves the operating efficiency as well as reducing maintenance costs. Landfill and digester gases dried in refrigerated dryers are low cost alternative fuels for internal combustion engines and boilers.

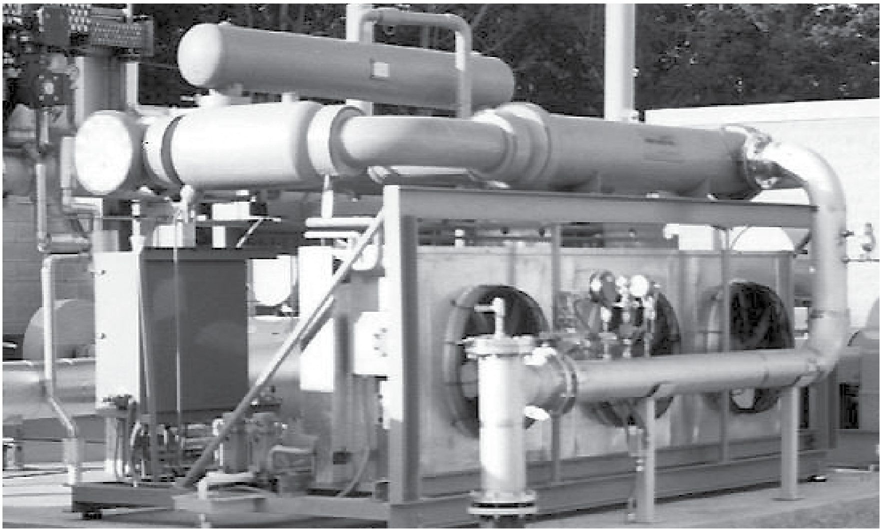

Figure 7.168 Landfill/Digester gas outdoor installation, air-cooled refrigeration system for cooling. Mechanical separate for moisture separation. No-gas loss automatic drains for condensate removal.

Mechanical refrigeration cools and dehumidifies the fuel gas through the dryer. Automatic no-gas loss drainage system helps the operation of the dryer non-hazardous. A package custom designed for outdoor installation offers economical installation.

Chemical Process Industry Drying

Nitrogen usage in the United States is more than 550 billion ft^3/year. Twenty-five percent of this gas is used as blanketing atmospheres for the chemical processing industry. Nitrogen, in most cases, is dried by adsorption. This keeps moisture from causing adverse chemical effects or a possible chemical reaction from a system. These potential chemical effects with water present could include corrosion of pipelines and vessels, poisoning and consumption of valuable catalysts, and inhibition of other reactions, as well as other effects. This is also true of other inert gases.

An example of this is an inert cover gas used for polymer production. A dry inert gas blankets the polymer material being produced. If moisture is present in the gas, the polymer will undergo oxidative degradation and the desired product will not be obtained.

Refrigerants are also dried by adsorption. To operate properly, the moisture content of the refrigerant system must not be allowed to be above a certain maximum. Therefore, it is very important during the manufacturing and assembly that no moisture enter the system. Also, if any moisture enters during the operation of

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the system, it must be removed quickly. Sources of moisture in a system can be leakage at the water-cooled condenser, oxidation of certain hydrocarbons, wet oil, wet refrigerant, decomposition of motor insulation in hermetically sealed units, and low-side leaks, which introduce wet air. The effects of this moisture are valve failure, corrosion, freezing of the expansion valve, copper plating, formation of ice in the evaporator, and chemical damage to insulation or other system materials. Desiccant dryers are used to remove this moisture from the installed system. Desiccant drying is exceptionally well suited for this application, not only because of its water-removal capacity, but also because there is no reaction with the refrigerant, oil, or machine parts. It is used satisfactorily with all refrigerants.

Carbon dioxide is another gas that is dried by solid adsorption. Over 4 million tons per year are produced. It is recovered from synthesis gas in ammonia production, from refinery production of hydrogen, from fermentation processes, and from natural wells. The main uses of carbon dioxide are refrigeration, beverage carbonation, urea manufacture, and enhanced oil recovery. Carbon dioxide is dried for production purposes. A carbon dioxide gas dryer at a chemical plant is shown in Fig. 7.168. The presence of moisture in carbon dioxide can also cause process line freezing at high pressures, along with corrosion problems.

Other Chemical Process Uses

The unloading of chlorine tank cars and other transfers of chemicals is accomplished by padding the tanks with dried compressed gas. Moisture in contact with many chemicals will cause rapid corrosion, deterioration, polymerization, and oxidation.

Figure 7.169 A carbon dioxide gas dryer operating at 720 psig for a chemical plant.

Oxygen manufacturers require the dew point of the air be as low as possible. If water is present, freeze-ups will occur at the low temperatures necessary for the liquefaction of air.

PETROCHEMICAL PROCESS INDUSTRY

Alkylation

Alkylation is the union of an olefinic with a paraffinic hydrocarbon to obtain high-octane gasoline. The reaction is catalyzed by hydrofluoric or sulfuric acid. The olefin is injected into the paraffinic feed, and the combined streams are contacted with acid.

The paraffin concentration is kept in excess to prevent copolymerization. Acid alkylation is limited to isobutane with propylene, butylene, and pentylene. Ethylene must be combined using $AlCl_3$. Phosphoric acid is used as a catalyst to unite propylene and benzene to form isopropyl benzene. Aluminum chloride and HCI catalyzes ethylene and benzene to ethylbenzene. Alkylations using aluminum chloride and other halogen-activated catalysts must have dry feed streams. Reduction, dilution, or loss of the catalyst will take place if the feed is not dried.

Gas Plant

The C_4 naphthenes and lighter gases from various refinery operations are sent to this section of the refinery. All these gases require dehydration.

Catalytic Re-forming

Catalytic re-forming refers to the octane improvement of straight-run gasoline and cracked refinery naphthas. C_5 and C_6 naphthenes are isomerized and dehydrogenated to aromatics; paraffins are hydrocracked or cyclicized and dehydrogenated to aromatics. Catalytic re-forming is also a source of benzene, toluene, and xylene. Recycled hydrogen, inert gas, and regeneration gases used in this process must be dried by means of solid adsorption to ensure proper process reactions. Hydrogen is also produced in large quantities by catalytic re-forming and must be dried before other refinery uses or sale.

Isomerization

Isomerization is the conversion of normal butane, pentane, and hexane into their respective isomers. It is a fixed-bed, vapor-phase process that is carried out under dry hydrogen atmosphere. This prevents coke deposition and saturates any cracking products. In this process, both the paraffin and hydrogen feeds need to be dried.

Compressed Air & Gas Institute • 1300 Sumner Avenue • Cleveland, OH 44115
Phone: 216/241-7333 • **Fax:** 216/241-0105 • **E-mail:** cagi@cagi.org

Other petrochemical processes, such as hydrocracking and catalytic cracking, utilize adsorption drying. Recycle and makeup hydrogen regeneration gas must be dried for proper reactions.

Metals and the Metallurgical Industry

In the metals industry, a blanketing or protective gas is used in many heat-treatment processes. The drying of this gas is imperative to produce a more uniform metal. A required grade is made with precision, and the furnace in which the heat treating is done works with greater regularity.

Steel is sometimes annealed in a controlled atmosphere prepared by the combustion of natural gas. A gas formed this way is called an exothermic base gas. An exothermic base gas is an inert gas generated when natural gas is burned with a controlled amount of air, a process that produces mostly nitrogen and carbon dioxide with 0.1 to 0.5 percent combustibles and 0 to 0.1 percent maximum oxygen. During the combustion, a considerable amount of water vapor is formed. The gas is cooled and then dried by a desiccant dryer. The gas then blankets the heat treating of the steel to prevent oxidation.

In heat treating or annealing aluminum, an exothennic gas is also used. Too much moisture in the furnace atmosphere can cause oxidation of the alloying constituents. The amount of moisture is extremely critical whenever a metal is exposed to processing. This is true not only for heat treating but also for polishing, carburizing, and welding of titanium, stainless steel, and other alloys.

A hydrogen atmosphere is used in copper-brazing furnaces for annealing highly oxidizable metals. Where even a slight amount of moisture is very detrimental, nickel, nickel steel, and Monel wires must be annealed in these furnaces to avoid discoloration.

Another heat-treating atmosphere used to a large extent is cracked ammonia. Anhydrous ammonia is dissociated into gases, resulting in three parts of hydrogen to one part of nitrogen. Most cracking units are highly efficient, so that the degree of dissociation is usually 99.75 to 99.95%. Since one volume of ammonia yields two volumes of the mixed gas, the ammonia content is then 0.125 to 0.025 percent by volume, respectively, or 2,500 to 5,000 ppm by weight. A molecular sieve dryer is used to remove water and the undissociated ammonia.

A gas of increasing importance in the heat treating of metals is HN_x. This gas has excellent properties in the bright annealing process and is nearly neutral with regard to carburization when treating steels with different carbon contents. Thus, the gas can be used universally. This makes the gas distribution within the workshop easier and enables the purchase of bigger and more economical gas-production plants.

The following is a typical HN_x gas composition:

N_2	= 93% to 96% by volume
H_2	= 7% to 4% by volume
O_2	= 0.0005% by volume
$CH_4 + NO + NO_2 + CO_2 + CO$	= 0.05% by volume
Dew point	= -58°F

HN_x gas is produced by the controlled combustion of fuel gas. This gas must be dried before being used in a furnace.

Electrical Industry

Sulfur hexafluoride (SF_6) is a nontoxic, inert gas that is about six times as heavy as air. Usually, this gas is dried to a pressure dew point of - 60°F or lower. Its main use is as an insulating gas for arc suppression in electrical switch gear, circuit breakers, and so on.

Food and Beverage

Carbon dioxide (CO_2) is obtained as a by-product from air-separation plants and is recovered from the fermentation process in the making of beer, wines, and various alcohols. The carbon dioxide is recovered from these processes, cleaned up, and then dried so that it can be stored as a high-pressure gas or as a solid. Carbon dioxide is reintroduced into various food products for the carbonization of soft drinks, beer, and some wines. As a solid, carbon dioxide is known as dry ice and is used for short-term storage of frozen foods, such as in portable ice cream carriers where mechanical refrigeration is not possible.

Nitrogen is used as a blanketing gas to eliminate oxygen and prolong the storage life of fruits such as apples, grapes, and bananas. It is also used for the blanketing of wines and aseptic packaging of foods such as coffee and potato chips, as well as other packaged foods. Elimination of moisture and oxygen from all these products is essential to retard spoilage.

Semiconductors

Nitrogen is a colorless, odorless, tasteless diatomic gas that is completely inert. One of the more recent applications has been as a purging and blanketing gas used in the production of semiconductor and microprocessor chips in the electronic industry.

Helium, another inert gas, is also used in the semiconductor industry as a blanketing inert atmosphere in the production of electronic parts and the growing of germanium and silicon crystals.

Medical Industry

Oxygen, which is essential to life, is used for resuscitation and inhalation therapy for people with lung disorders such as emphysema, to ease recovery after a major operation, and as a heart stimulant for patients with heart disorders.

Nitrous oxide is a relatively mild anesthetic used mostly in dentistry, dental surgery, and medicine. It is a narcotic in high concentrations and has the side effect of a type of carefree hysteria, from which it has derived its common name, laughing gas.

Medical compressed air is compressed air that has been produced free of any contaminants such as oil, carbon monoxide, dirt, dust, and water vapor. It is used in the preparation of pharmaceuticals, as a breathing aid, and as an atmosphere in hyperbaric chambers where operations are performed at elevated pressures.

Carbon dioxide is used as a purging or blanketing atmosphere in the production of medicines and pharmaceuticals where elimination of oxygen is important. It is also used as an aerosol propellant in medication dispensers.

Helium provides an inert atmosphere necessary in the production of certain types of pharmaceuticals. It is also used, mixed with oxygen as a breathing gas to replace nitrogen, which is normally adsorbed into the bloodstream. In a hyperbaric chamber, this gas is used for the patient's breathing so that decompression will have none of the damaging side effects caused by nitrogen.

Nitrogen, as with most inert gases, is used in the production of pharmaceuticals. As a liquid, it is used in surgery and for the quick freezing of tissues for medical study.