COMPLETE REVISION
April 2013

Process Industry Practices
Machinery

PIP REEC001
Compressor Selection Guidelines

PUBLISHING HISTORY

March 1999	*Issued*
February 2007	*Complete Revision*
April 2013	*Complete Revision*

Not printed with State funds

Process Industry Practices
Machinery

PIP REEC001
Compressor Selection Guidelines

Table of Contents

Tables

Figures

1. Introduction

1.1 Purpose

This Practice provides guidelines for the selection of compressors.

1.2 Scope

This Practice describes the overall guidelines for selecting an appropriate type of compressor for a specific process service. Guidelines are also provided for lubrication and seal systems.

2. References

Applicable parts of the following Practices and references should be considered an integral part of this Practice. The edition in effect on the date of contract award should be used, except as otherwise noted. Short titles are used herein where appropriate.

2.1 Process Industry Practices (PIP)

- PIP REIE686/API RP686 - *Recommended Practices for Machinery Installation and Installation Design*
- PIP REIE686A - *Recommended Practice for Machinery Installation and Installation Design (Supplement to PIP REIE686/API RP686)*
- PIP RESE003 - *Application of General Purpose Non-Lubricated Flexible Couplings*

2.2 Industry Codes and Standards

- American Petroleum Institute (API)
  - API 611 - *General-Purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services*
  - API 612 - *Special-Purpose Steam Turbines for Petroleum, Chemical, and Gas Industry Services*
  - API 613 - *Special-Purpose Gear Units for Petroleum, Chemical, and Gas Industry Services*
  - API 614 - *Lubrication, Shaft-Sealing, and Control-Oil Systems for Special-Purpose Applications*
  - API 617 - *Centrifugal Compressors for Petroleum, Chemical, and Gas Service Industries*
  - API 618 - *Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services*
  - API 619 - *Rotary-Type Positive Displacement for Petroleum, Chemical, and Gas Industry Services*
  - API 670 - *Vibration, Axial-Position, and Bearing-Temperature Monitoring Systems*
  - API 671 - *Special-Purpose Couplings for Refinery Service*

- API 672 - *Packaged, Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical, and Gas Industry Services*
- API 677 - *General-Purpose Gear Units for Petroleum, Chemical, and Gas Industry Services*

- American Society of Mechanical Engineering (ASME)
  - ASME B19.3 - *Safety Standard for Compressors for Process Industries*
  - ASME PTC-10 - *Performance Test Codes - Compressors and Exhausters*
- Association of German Engineers (VDI)
  - VDI 2045 - *Acceptance and Performance Test of Dynamic and Positive Displacement Compressors*
- Gas Processors Suppliers Association
  - *Engineering Data Book*

2.3 Other References

- *Compressor Application Engineering*, Volume 1, Compression Equipment, Gulf Publishing Co., Pierre Pichot
- *Compressor Application Engineering*, Volume 2, Drivers for Rotating Equipment, Gulf Publishing Co., Pierre Pichot
- *Compressors: Selection and Sizing*, 2nd ed., Gulf Publishing Co., Royce N. Brown
- *Leak-Free Pumps & Compressors*, Elsevier Advanced Technology, Gerhard Vetter

3. Nomenclature and Definitions

absolute pressure: Pressure measured from absolute zero (i.e., from an absolute vacuum). Absolute pressure is equal to the algebraic sum of the atmospheric pressure and the gauge pressure.

absolute temperature: Temperature above absolute zero

actual volumetric flow rate, also actual cubic meters per hour (acmh) or actual cubic feet per minute (acfm): The volume throughput per unit time at the compressor inlet

adiabatic compression: Compression process where no heat transfer takes place (process may be irreversible)

capacity: Volume rate of flow of gas compressed and delivered, expressed at inlet conditions

density: Mass of the gas per unit volume

discharge pressure: Pressure at the discharge flange of the compressor

discharge temperature: Gas temperature at the discharge flange of the compressor

displacement: Average volume displaced per unit time by the piston of reciprocating compressors or by vanes, screws and lobes of rotary compressors. If used to indicate size or rating, displacement must be related to a specified speed. For multistage machines, it refers to the first stage cylinder(s) only.

inlet (suction) pressure: Pressure at or near the inlet flange of the compressor. In an air compressor without inlet pipe or duct, absolute inlet pressure is equal to the atmospheric pressure which is determined by the altitude.

intercooling: Removal of heat from the gas between compressor sections

kinetic energy (of a mass): Total work which must be done to bring a mass from a state of rest to a velocity, v. For rectilinear motion, the kinetic energy equals ½ mv^2.

molecular weight: Relative weight of a molecule of a substance referred to that of an atom of Carbon-12 as 12.000

polytropic compression: Reversible compression process which follows a path such that between any two points on the path, the ratio of the reversible work input to the enthalpy rise is constant

polytropic efficiency: Ratio of the polytropic work to the actual work required for the compression process

polytropic head: Reversible work required to compress a unit mass of the gas in a polytropic compression process

pressure (compression) ratio: Ratio of absolute discharge pressure to the absolute inlet pressure

section (of a centrifugal compressor): Group of stages (impellers and associated stationary parts) in series. The term “section” defines all the compression components between inlet and discharge flanges. A compressor case (body) may contain multiple sections if the gas is removed for cooling and returned for further compression.

shaft power: Measured power input to the compressor. The term “shaft power” is expected to replace the more commonly used term “brake horsepower.”

specific volume: Volume occupied by one unit of mass of the gas

stage of compression (in centrifugal compressors): A single impeller with the associated stationary parts. For axial machines, each set of blades (one rotating row followed by one stationary row) represents a stage of compression. In reciprocating compressors, a cylinder, piston, and associated valves and parts comprise a stage. Compressor stages for other types of compressors can be defined in a similar manner.

stonewall (choke) point: Condition at which flow in a centrifugal compressor cannot be increased further. In this condition, sonic flow is reached in some part of the gas path within the compressor.

surge point: Minimum flow point in a centrifugal or axial compressor. If flow is reduced below this point, cyclic variation (and even reversal) of gas flow and discharge pressure occurs.

volumetric efficiency: Ratio of the capacity of the compressor cylinder to the displacement of the cylinder in a reciprocating compressor

4. General

4.1 Compressors are the prime movers of gas and air in process industries.

4.2 Compressors are used to increase the static, or inlet, pressure of the gas and deliver it at the specified discharge pressure and flow rate in a process application. Part of the increase in static pressure is required to overcome frictional resistance in the process.

4.3 Compressors are available in a variety of types, models and sizes, each of which fulfills a given need. The selection should represent the best available configuration to meet specified requirements.

4.4 Mechanical integrity and process safety should be important considerations.

4.5 The process requirements should be established before selecting a compressor system. Selection requires matching compressor capabilities to process requirements.

4.6 A compressor system generally includes:

a. Compressors

b. Lubrication systems

c. Seal systems

d. Drivers

e. Couplings

f. Gearboxes

g. Control systems

h. Baseplate/skid

4.7 All components of a compressor system determine its capacity, reliability and life-cycle costs. Life-cycle costs include initial capital, operating and maintenance costs.

4.8 Field-proven equipment should be selected. However, if justifiable, newly developed equipment may be selected with appropriate technical scrutiny and follow-up.

4.9 If more than one type or size of compressor initially appears suitable for a given application, then a more detailed analysis may be justified to make a final, appropriate selection.

5. Types of Compressors

5.1 General

5.1.1 The two basic categories of compressors are:

a. Dynamic (centrifugal and axial)

b. Positive displacement (reciprocating and rotary types).

5.1.2 Figure 1 shows the various types of compressors related to the two categories.

5.1.3 Tables 1a and 1b and Figures 2a and 2b show typical operating characteristics of the different types of compressors.

5.1.4 Table 2 shows the general advantages and disadvantages of the different types of compressors.

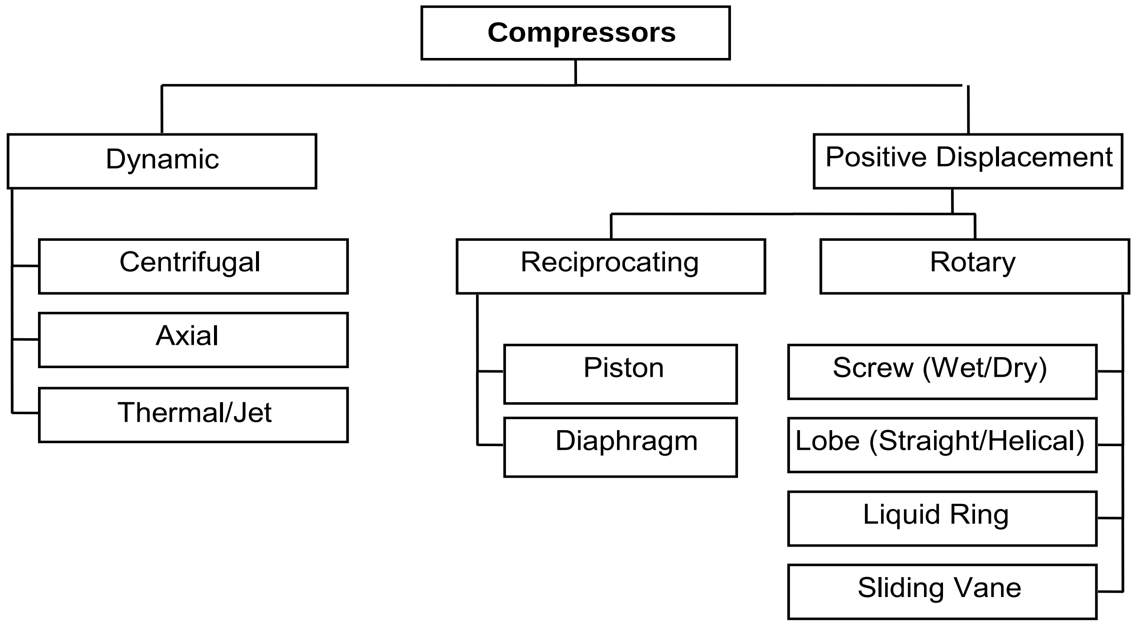


Figure 1. Types of Compressors

Table 1a. Summary of Typical Operating Characteristics of Compressors (Metric)

	Inlet Capacity (acmh, m^3/h)	Maximum Discharge Pressure (bar)	Adiabatic Efficiency (%)	Operating Speed (rpm)	Maximum Power (MW)	Application
Dynamic Compressors						
Centrifugal	170 - 850,000	690	70 – 87	1,800 - 50,000	38	Process gas & air
Integrally Geared Centrifugal Compressors	500 - 500,000	350	80	7,000 - 50,000	60	Process gas & air
Axial	50,000 - 850,000	17	87 - 90+	1,500 - 10,000	75	Mainly air
Positive Displacement Compressors						
Reciprocating (Piston)	20 - 34,000	4,150	80 - 90	200 - 900	15	Air & process gas
Diaphragm	0 – 250	1,400	60 - 70	300 - 500	1.5	Corrosive & hazardous process gas
Rotary Screw (Wet)	100 - 12,000	24	65 - 70	1,500 - 3,600	1.5	Air, refrigeration & process gas
Rotary Screw (Dry)	200 – 100,000	1 – 50	55 – 70	1,000 – 20,000	6	Air & dirty process gas
Rotary Lobe	25 - 50,000	0.3 - 1.7	55 - 65	300 - 4,000	0.4	Pneumatic conveying, process gas & vacuum
Sliding Vane	15 - 5,000	10	40 - 70	400 - 1,800	0.35	Vacuum service & process gas
Liquid Ring	10 - 17,000	5.5 – 10.5	25 - 50	200 - 3,600	0.3	Vacuum service & corrosive process gas

Table 1b. Summary of Typical Operating Characteristics of Compressors (US Units)

	Inlet Capacity (acfm)	Maximum Discharge Pressure (psig)	Efficiency (%)	Operating Speed (rpm)	Maximum Power (HP)	Application
Dynamic Compressors						
Centrifugal	100 - 200,000	10,000	70 – 87	1,800 - 50,000	50,000+	Process gas & air
Integrally Geared Centrifugal Compressor	300 - 30,000	4,000	80	7,000 - 50,000	80,000	Process gas & air
Axial	30,000 - 500,000	250	87 - 90+	1,500 - 10,000	100,000	Mainly air
Positive Displacement Compressors						
Reciprocating (Piston)	10 - 20,000	60,000	80 – 95	200 - 900	20,000	Air & process gas
Diaphragm	0.5 – 150	20,000	60 – 70	300 - 500	2,000	Corrosive & hazardous process gas
Rotary Screw (Wet)	50 - 7,000	350	65 – 70	1,500 - 3,600	2000	Air, refrigeration & process gas
Rotary Screw (Dry)	120 – 58,000	15 – 700	55 – 70	1,000 - 20,000	8,000	Air & dirty process gas
Rotary Lobe	15 - 30,000	5 - 25	55 – 65	300 - 4,000	500	Pneumatic conveying, process gas & vacuum
Sliding Vane	10 - 3,000	150	40 – 70	400 - 1,800	450	Vacuum service & process gas
Liquid Ring	5 - 10,000	80 - 150	25 – 50	200 - 3,600	400	Vacuum service & corrosive process gas

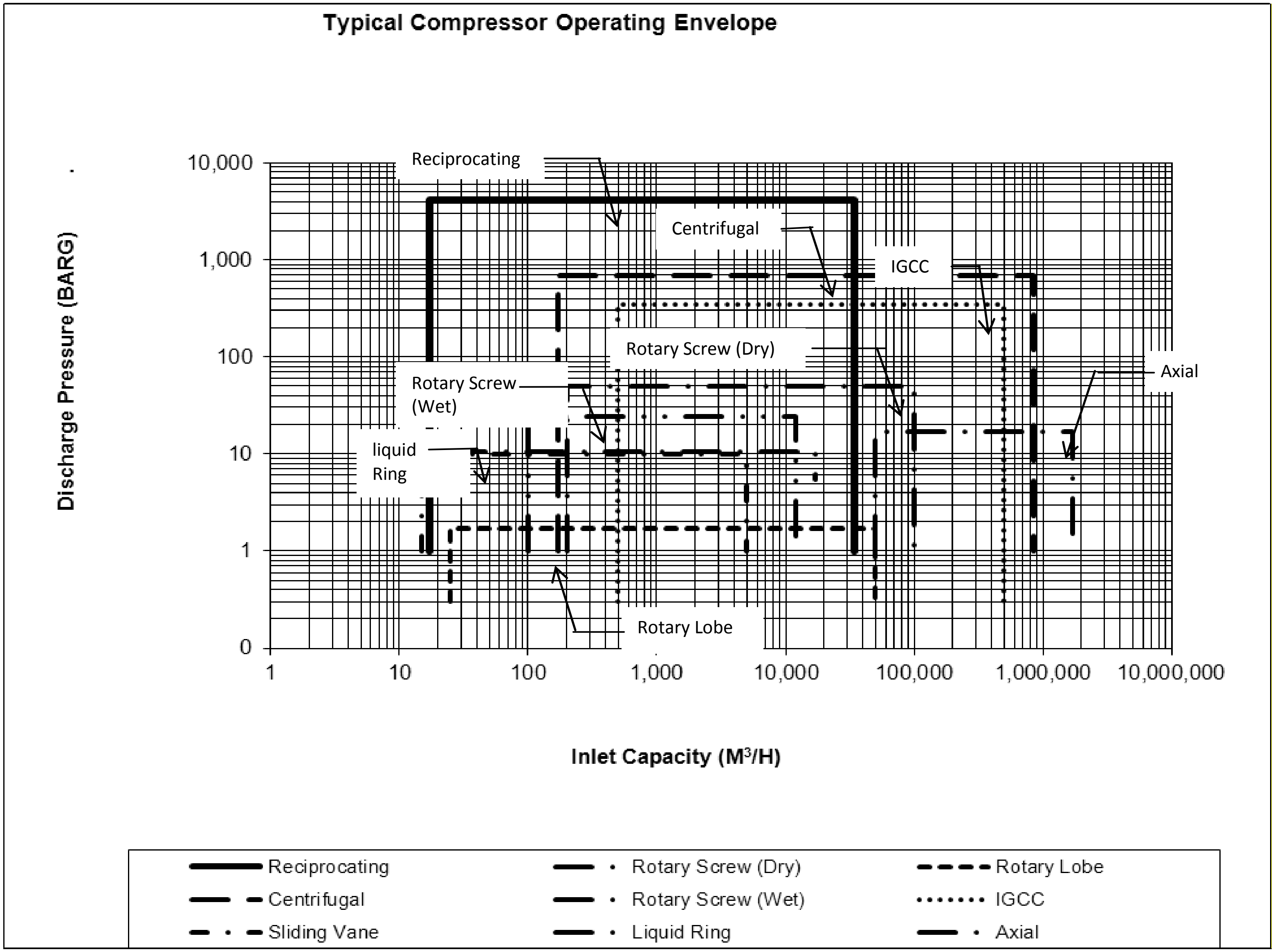
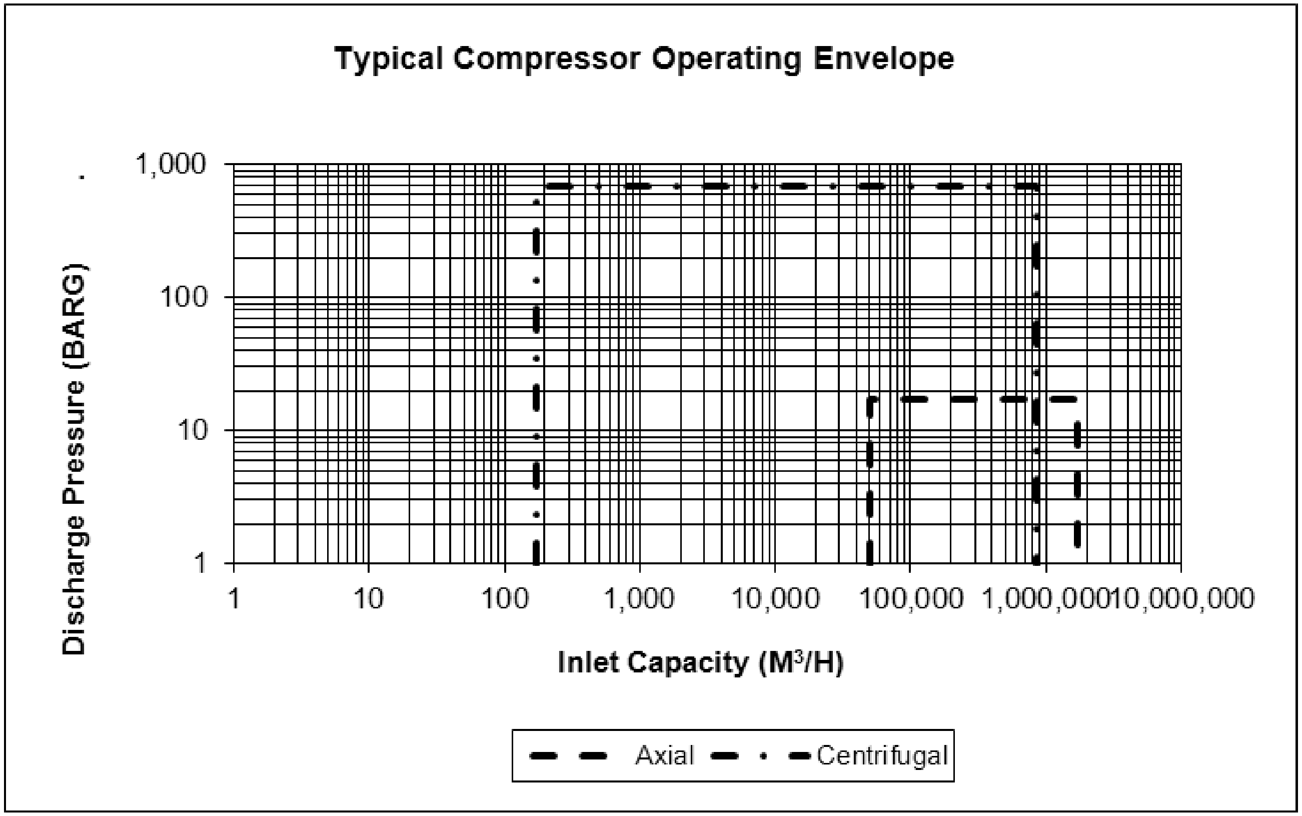
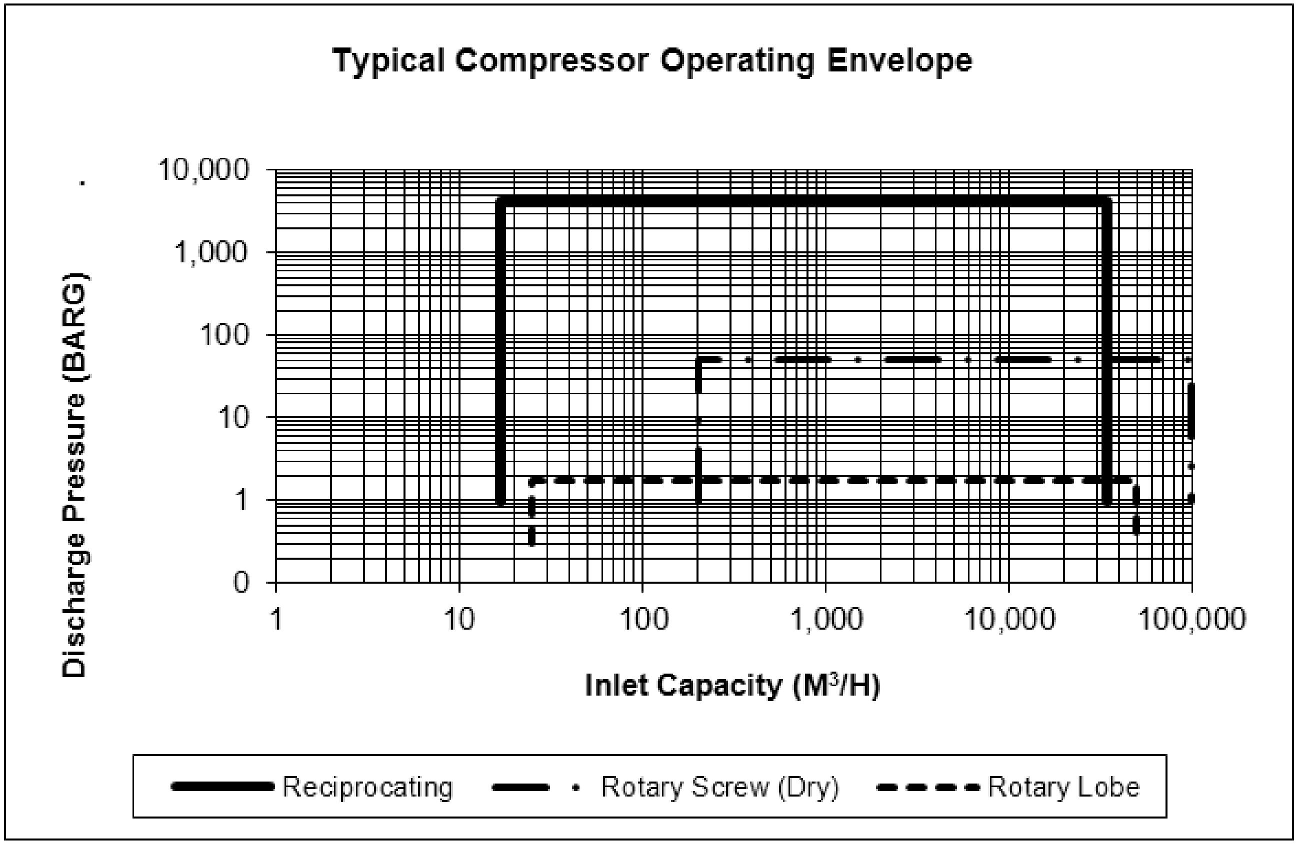


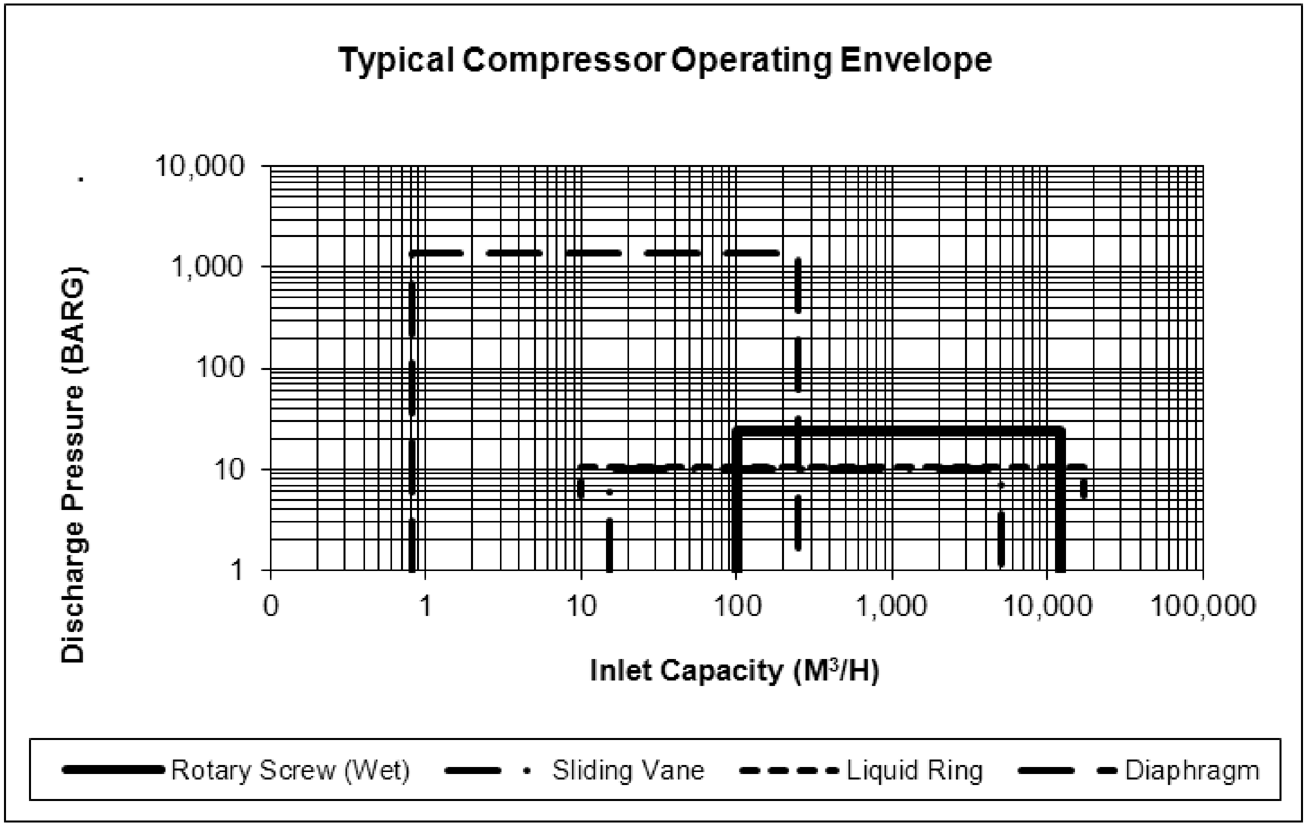
Figure 2a. Typical Operating Envelopes of Compressors (Metric)



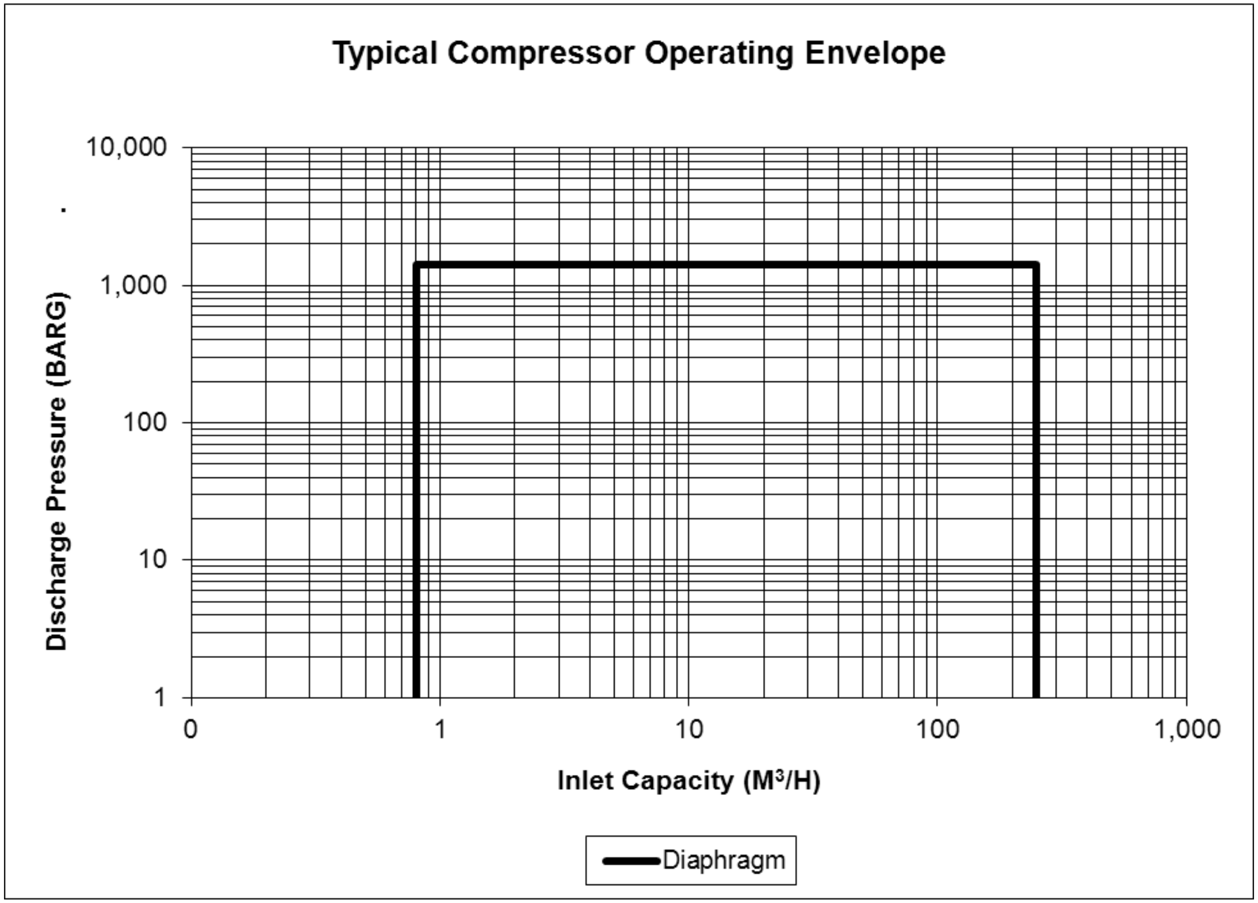
Typical Operating Envelope of Centrifugal and Axial Compressors (Metric)



Typical Operating Envelope of Reciprocating, Rotary Screw (Dry), and Rotary Lobe (Metric)



Typical Operating Envelop of Rotary Screw (Wet), Sliding vane, and Liquid Ring Compressors (Metric)



Typical Operating Envelope for Diaphragm Compressors (Metric)

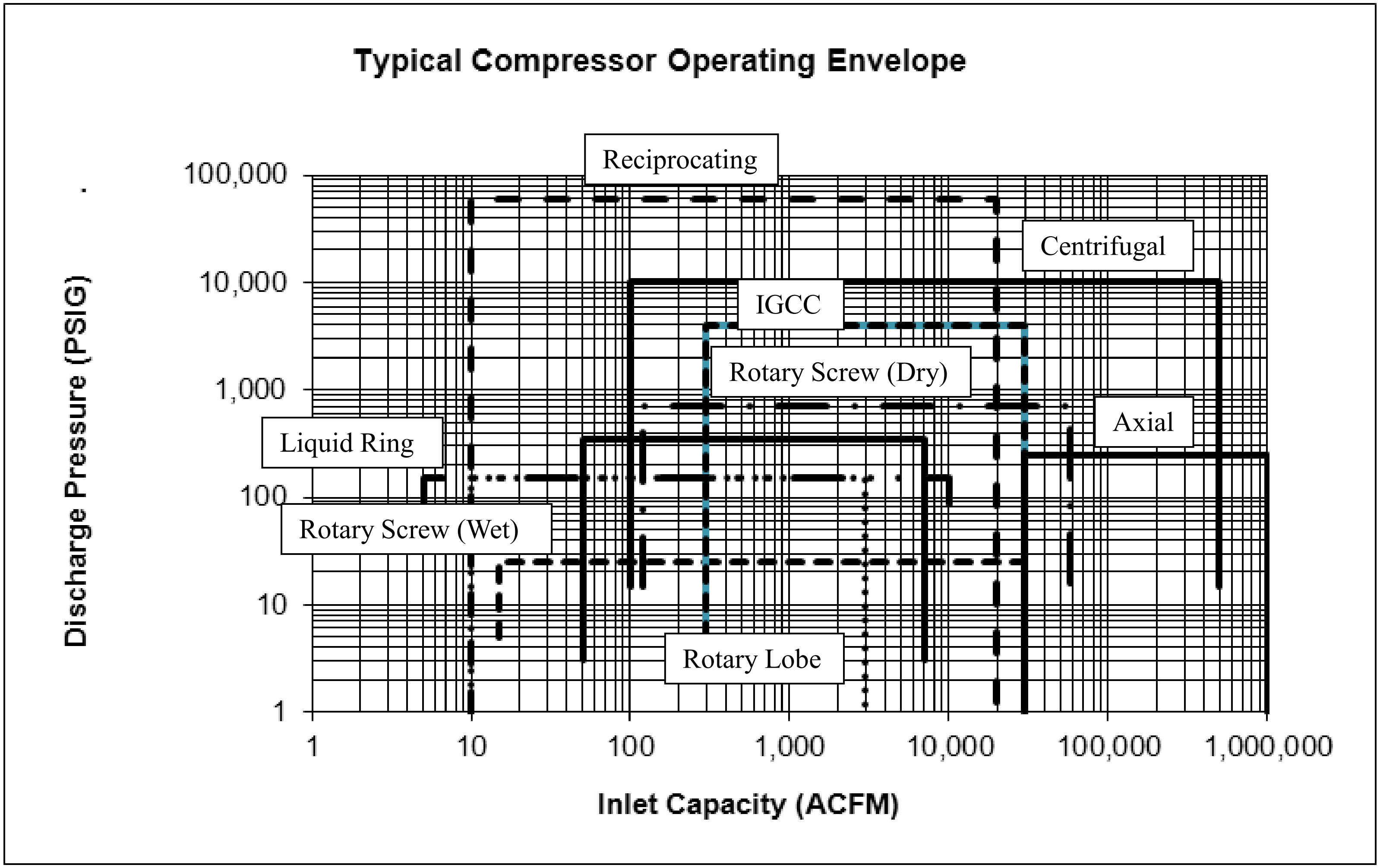
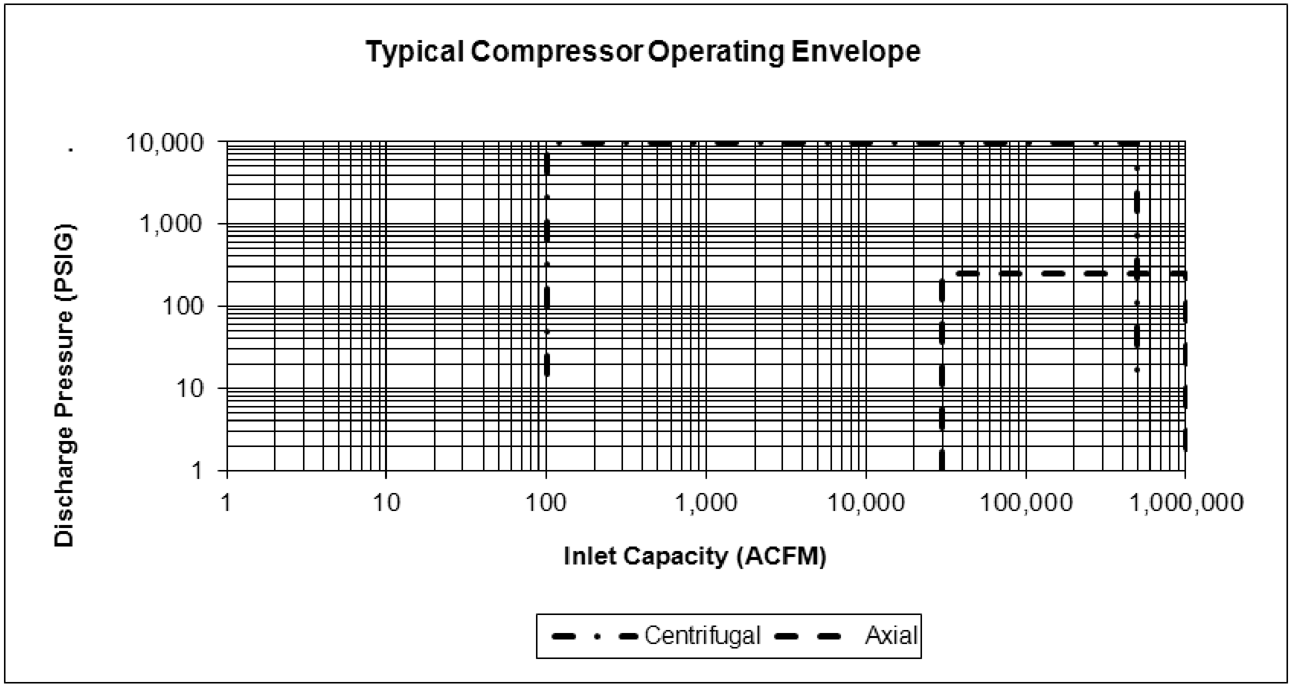
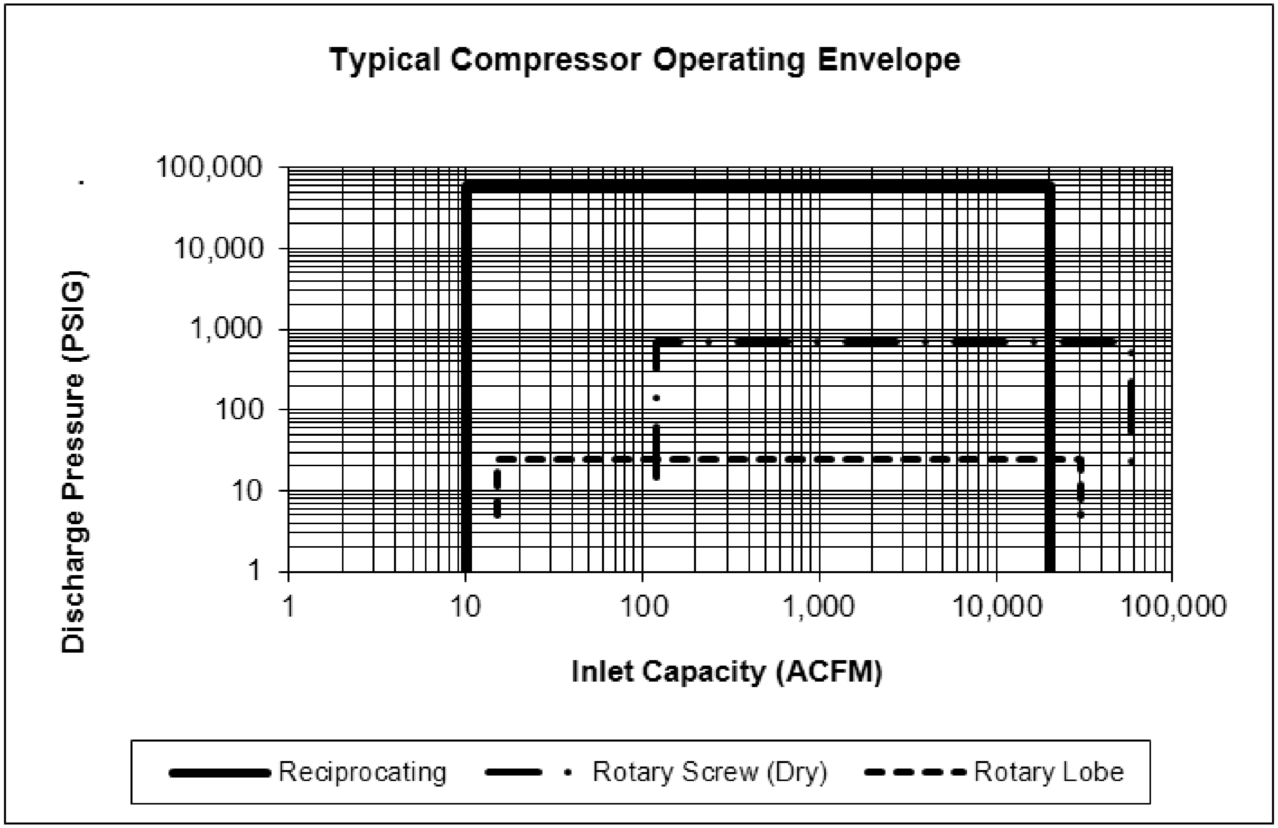


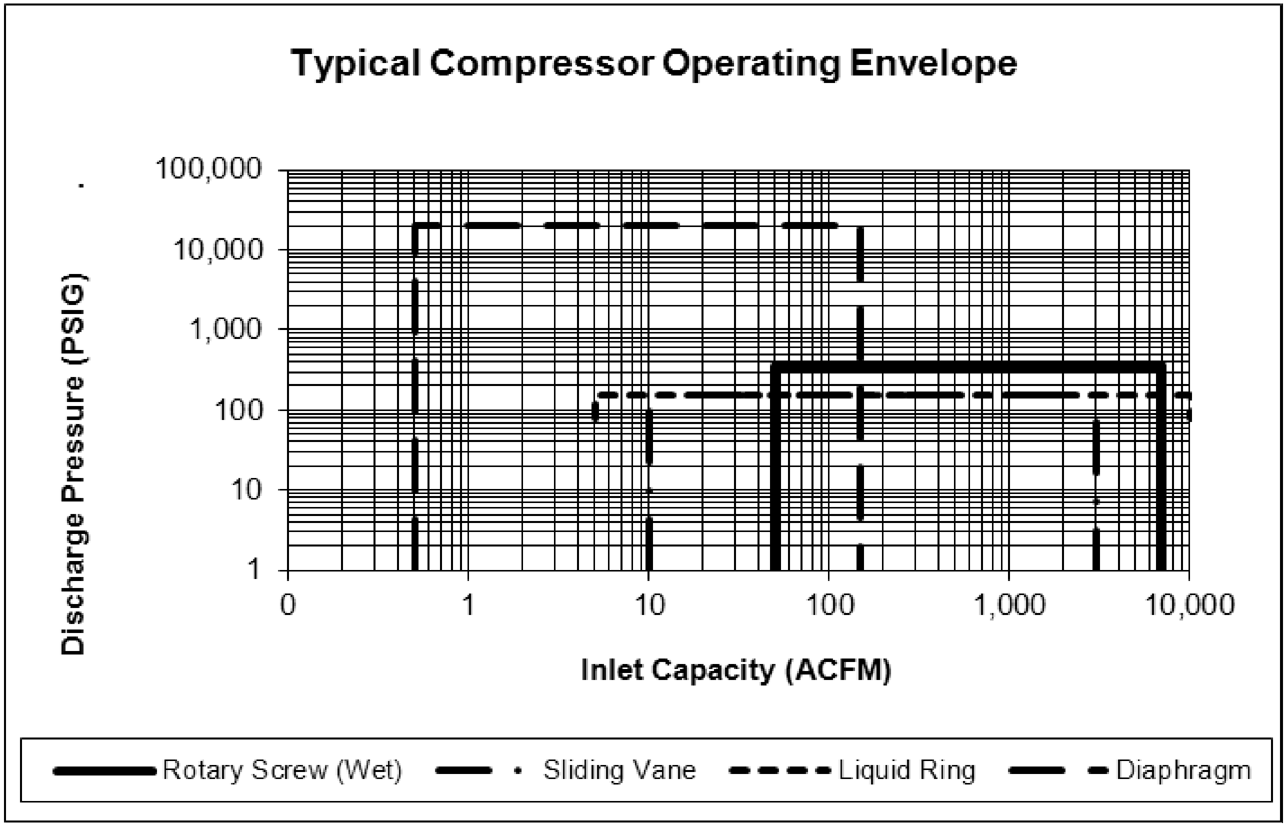
Figure 2b. Typical Operating Envelopes of Compressors (U.S. Units)



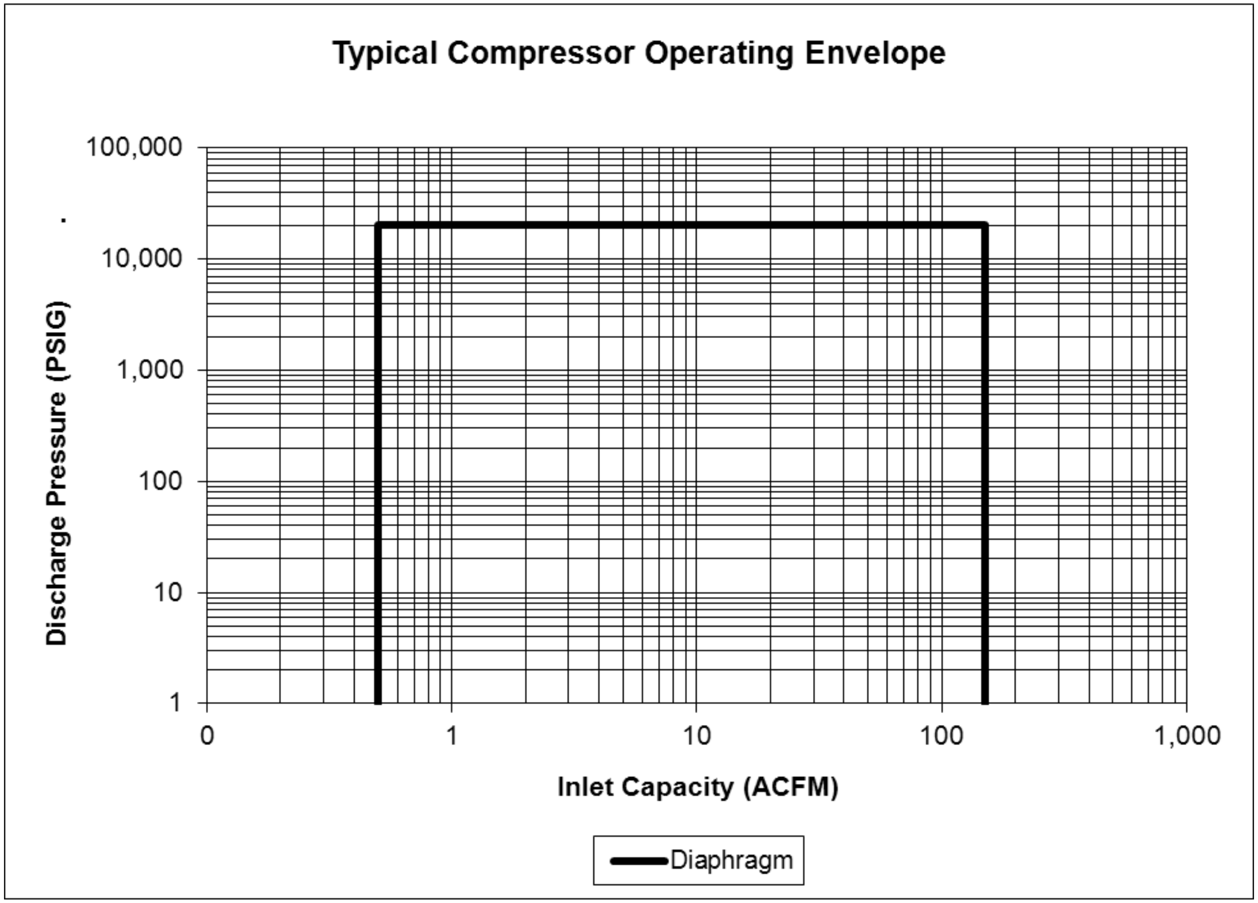
Typical Operating Envelope of Centrifugal and Axial Compressors (US Units)



Typical Operating Envelope of Reciprocating, Rotary Screw (Dry), and Rotary Lobe Compressors (US Units)



Typical Operating Envelop of Rotary Screw (Wet), Sliding Vane, and Liquid Ring Compressors (US Units)



Typical Operating Envelope of Diaphragm Compressors (US Units)

Table 2. Advantages and Disadvantages of Different Compressors

	Advantages	Disadvantages
Dynamic Compressors		
Centrifugal	• Wide operating range • High reliability and low maintenance	• Instability at reduced flow • Sensitive to changes in gas composition • Susceptible to rotor-dynamics problems • Sensitive to liquids in the gas stream
Integrally Geared Centrifugal	• Relatively inexpensive method of obtaining a high compression ration • High efficiency at best efficiency point	• Limited operating flow range • High efficiency is limited to flow rates near the best efficiency point • Sensitive to liquids in the gas stream
Axial	• High capacity for a given size and high efficiency • Heavy duty and low maintenance	• Low compression ratios • Limited turndown
Thermal/Jet	• No moving parts and low maintenance • High pressure ratio	• Very low efficiency • Narrow range of application
Positive Displacement Compressors	• More tolerant of changes in gas composition than dynamic compressors.	
Reciprocating (Piston)	• Wide pressure ratios • High efficiency	• Heavy foundations required due to unbalanced forces • Flow pulsation can cause vibration and structural problems • High maintenance compared to dynamic compressors • Sensitive to liquids in the gas stream
Diaphragm	• Very high pressure • Available in special materials • No moving seals • Low flow	• Limited capacity range • Periodic replacement of diaphragms required • Flow pulsation problems
Screw	• Wide range of applications • Wet screw has high efficiency and high pressure ratio • Dry screw insensitive to changes in gas composition and can handle dirty gases	• Noisy • Wet screw not suitable for corrosive or dirty gases

Table 2. Advantages and Disadvantages of Different Compressors (Continued)

	Advantages	Disadvantages
Positive Displacement Compressors		
Lobe	• Simple in design and construction • Low cost	• Limited operating range and pressure ratio • Capacity control limited to suction throttling
Sliding Vane	• Simple in design • High single-stage pressure ratio • Able to tolerate small quantities of liquids in the process gas	• Generally unsuitable for process gases • Low reliability
Liquid Ring	• High vacuum capability • Able to tolerate small quantities of liquids in the process gas • High single-stage pressure ratio • High reliability	• Sealing liquid/process gas compatibility required • Sealing liquid separation equipment required • Limited suction pressure

5.2 Dynamic Compressors

5.2.1 General

5.2.1.1 Dynamic compressors function by imparting velocity to a gas, and subsequently converting the kinetic energy of the moving gas to an increase in static pressure.

5.2.1.2 Two primary types of dynamic compressors are centrifugal and axial. A combination of the two types may be used to satisfy a given set of process requirements.

5.2.2 Centrifugal Compressors

5.2.2.1 Centrifugal compressors impart velocity to the gas through rotating impellers.

5.2.2.2 The gas is introduced at the eye of the impeller and discharged radially at the outer circumference (impeller tip) at a higher velocity and kinetic energy.

5.2.2.3 The gas then passes through a stationary diffuser where its velocity is reduced, and its kinetic energy is converted to static pressure. Part of the static pressure rise occurs in the impeller and part in the diffuser.

5.2.2.4 Impeller diameter and width, rotational speed and the angle of the vanes that make up the impeller are important parameters in the design of centrifugal compressors.

5.2.2.5 One or more impellers (stages) may be required to obtain the desired discharge pressure.

5.2.2.6 The temperature of the gas increases as it is being compressed.

5.2.2.7 If the discharge temperature is likely to increase beyond the maximum allowable temperature before the desired discharge pressure is achieved, the gas must be cooled before it can be compressed further.

5.2.2.8 Heat exchangers, called intercoolers, may be required to cool the gas. Typically, these intercoolers are installed between compressor sections (one or more stages typically make up a compressor section).

5.2.2.9 Maximum allowable temperature should be determined by:

a. Design or materials of the compressor

b. Avoiding excessive temperatures in the process

c. Fouling

d. Polymerization of the process

5.2.2.10 A seal system contains the process gas within the compressor as well as prevents any contamination of the gas by bearing lubricants.

5.2.2.11 These compressors can be designed to be non-lubricating if the process gas is required to be oil-free.

5.2.2.12 Cross-sectional views of a typical single-stage compressor, an integrally geared centrifugal compressor, and a typical multi-stage compressor are shown in Figures 3, 4, and 5 respectively.

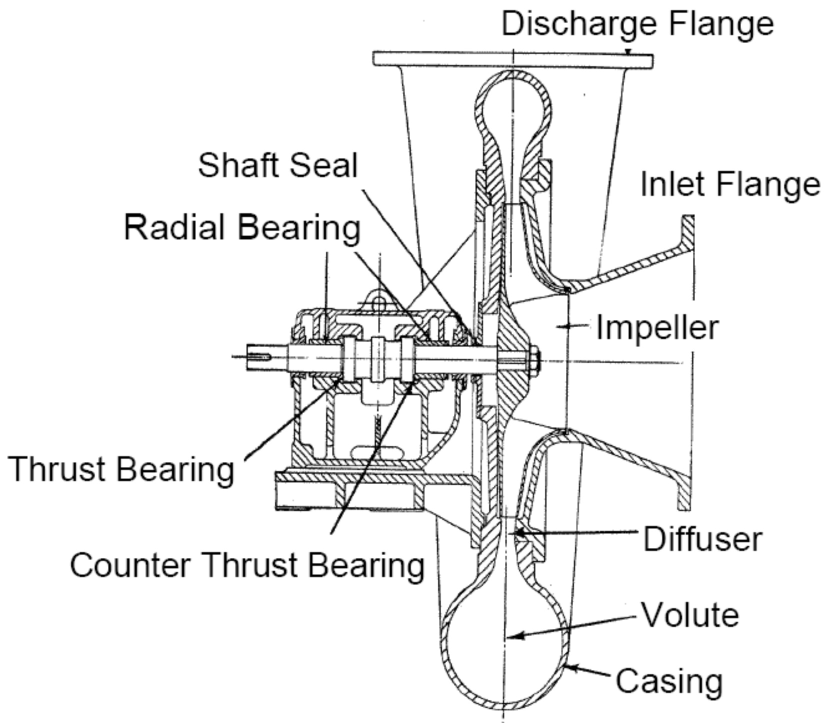


Figure 3. Typical Single-Stage Compressor

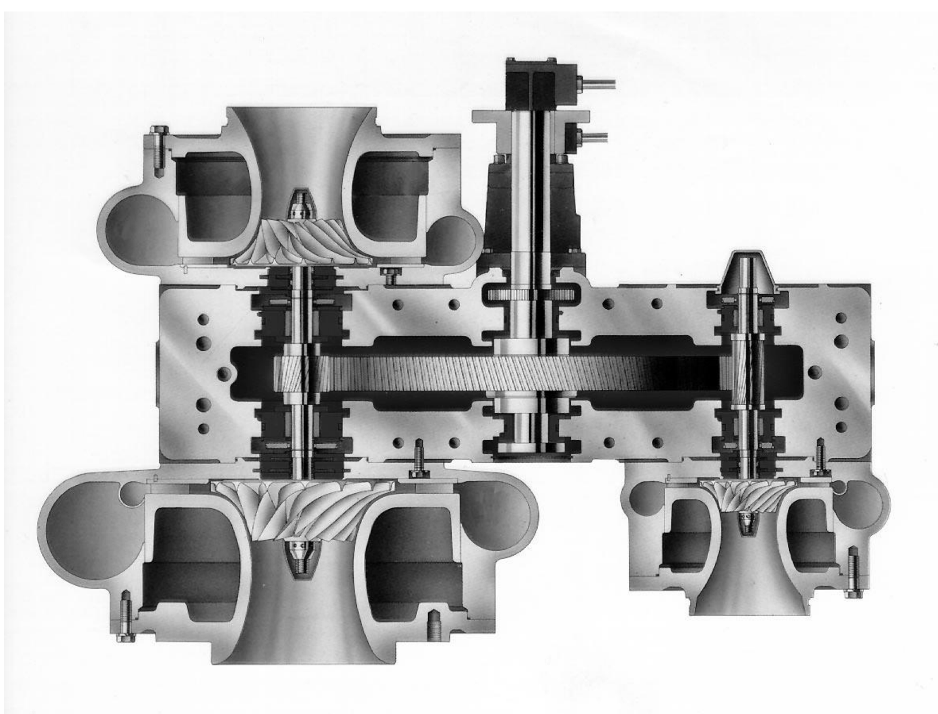


Figure 4. Typical Integrally Geared Centrifugal Compressor

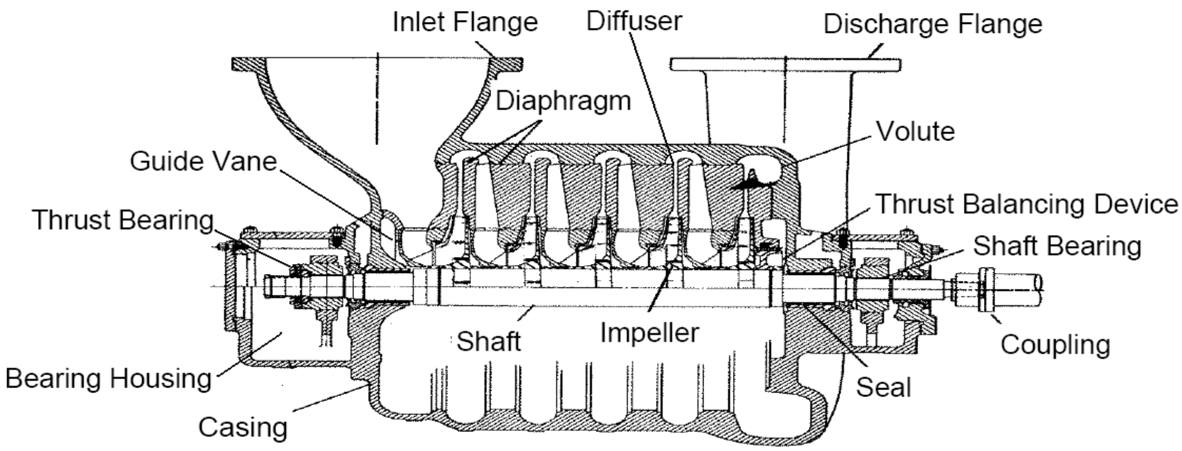


Figure 5. Typical Multi-Stage Compressor

5.2.3 Axial Compressors

5.2.3.1 Gas moves axially along the compressor shaft (parallel to the machine axis) through alternating rows of rotating and stationary blades. Each set of blades (one rotating row followed by one stationary row) represents a stage of compression.

5.2.3.2 The rotating blades impart velocity to the gas; the stationary blades slow the gas and direct it into the next row of rotating blades.

5.2.3.3 As the gas passes through each stage, its pressure and temperature are further increased until it finally exits the compressor at the required pressure and associated temperature.

5.2.3.4 Axial compressors require more stages to develop the same pressure rise as centrifugal compressors, but can handle very high flows at very high efficiency.

5.2.3.5 Axial compressor design does not lend itself to intercooling, and attainable discharge pressures are generally limited by temperature and bearing span (which limits the number of blade rows).

5.2.3.6 A typical axial compressor cross-sectional view is shown in Figure 6.

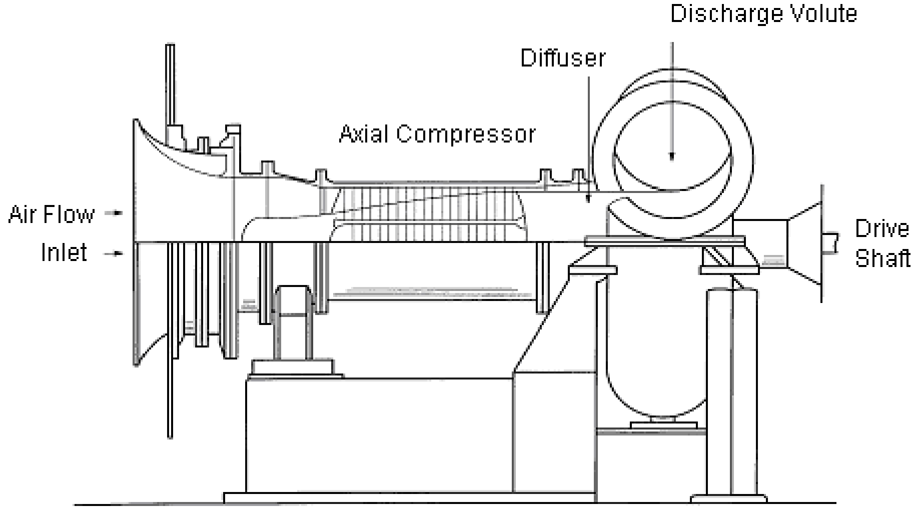


Figure 6. Typical Axial Compressor

5.3 Positive Displacement Compressors

5.3.1 General

5.3.1.1 Positive displacement compressors include reciprocating (piston and diaphragm) and rotary (screw, lobe, sliding vane and liquid ring) compressors.

5.3.1.2 Positive displacement compressors function by enclosing an initial volume of gas and reducing that volume mechanically thereby increasing its pressure.

5.3.1.3 Volume reduction is accomplished by one of the following:

a. Piston reciprocating in a cylinder

b. Reciprocating flexing of a diaphragm

c. Eccentric rotation of a volume sealed by sliding vanes

d. Matched rotating lobes in a volume cavity

e. Matched male and female helical screws

f. Rotor operating in a sealing liquid in an eccentric casing

5.3.2 Reciprocating Positive Displacement

5.3.2.1 Reciprocating positive displacement compressors have a piston in a cylinder or a diaphragm operating in a shaped cavity to compress the gas.

5.3.2.2 Reciprocating positive displacement compressors have suction and discharge valves that control the gas flow.

5.3.2.3 Reciprocating positive displacement compressors require special process and piping design considerations because of the pulsating nature of the flow.

5.3.2.4 Other considerations may include:

a. Foundation design

b. Compressor valve speed

c. Piston rod loads

d. Intercooling and safety relief valves.

5.3.2.5 Figure 7 shows a piston-type compressor and Figure 8 shows a diaphragm-type compressor.

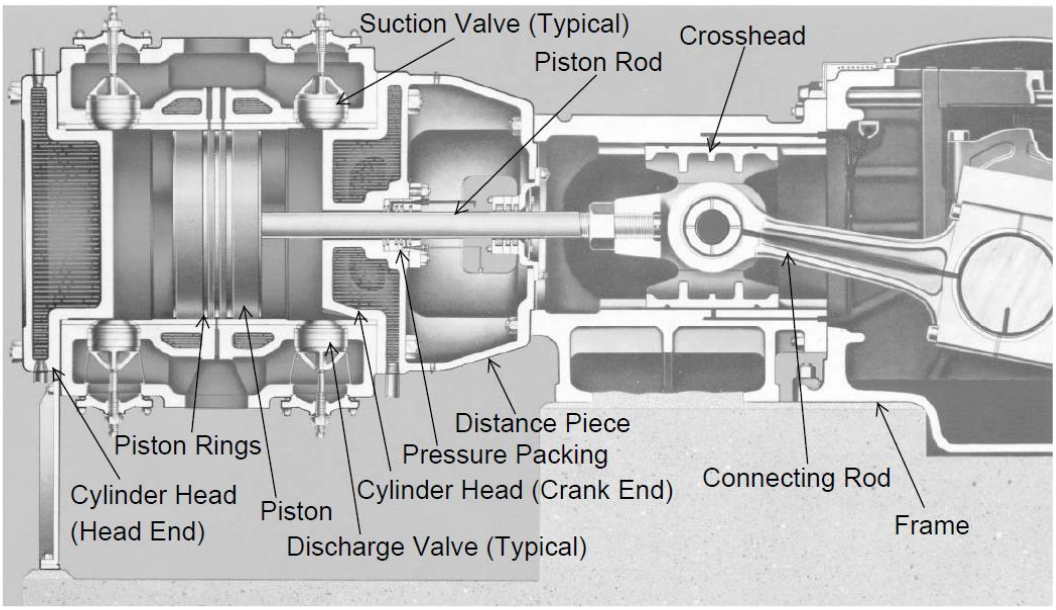


Figure 7. Typical Piston-Type Reciprocating Positive Displacement Compressor

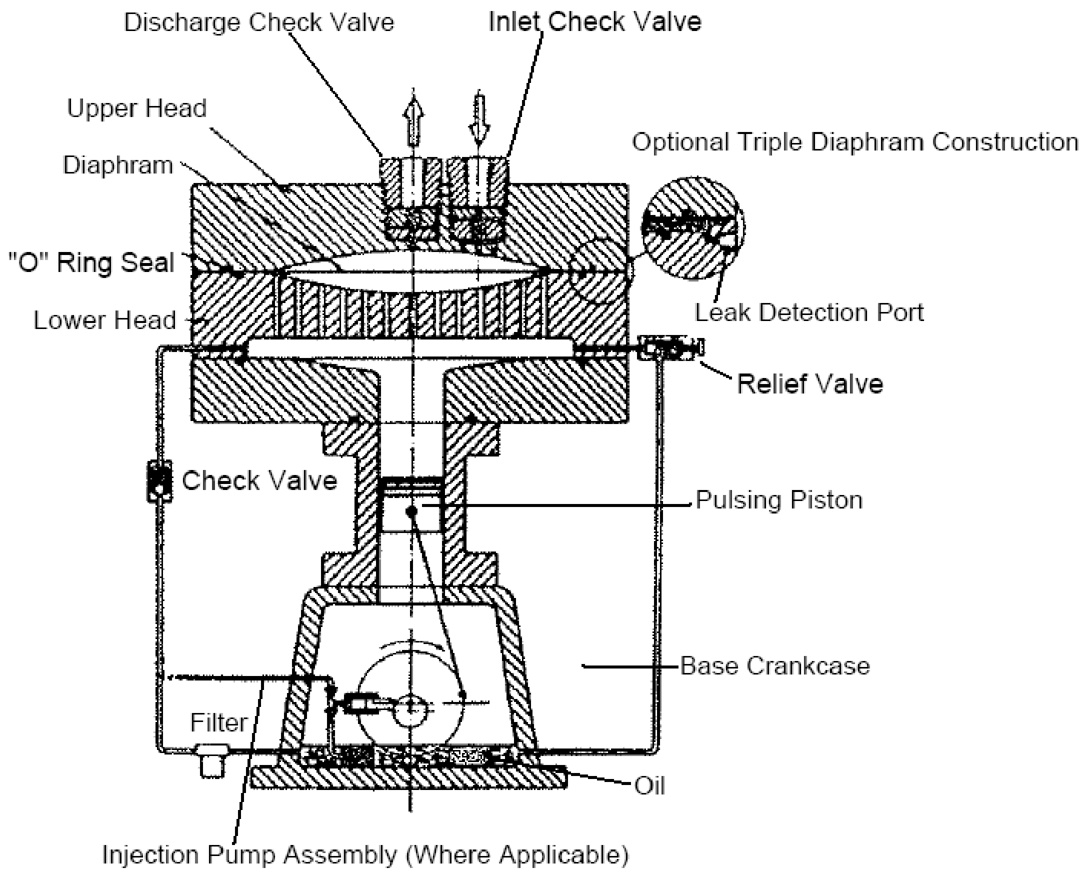


Figure 8. Typical Diaphragm-Type Reciprocating Positive Displacement Compressor

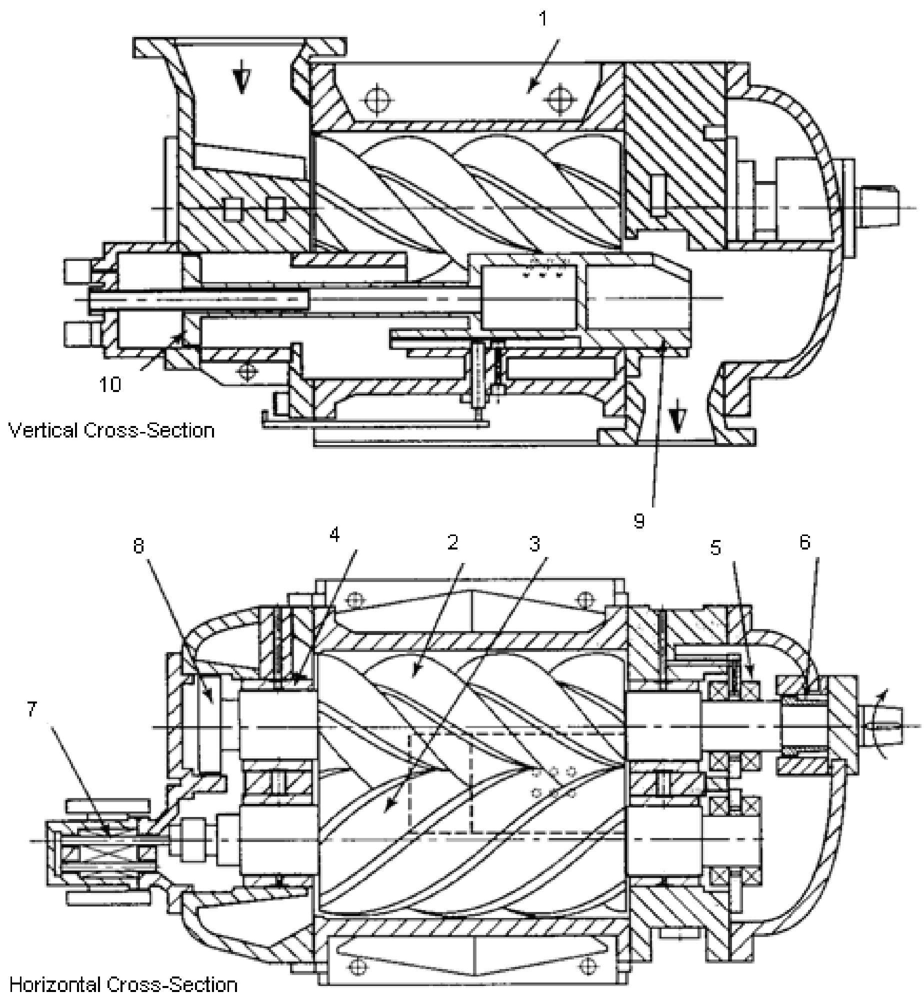
5.3.3 Rotary

5.3.3.1 Rotary compressors generally have no suction or discharge valves, and use suction and discharge ports that are alternately exposed and covered by the rotating elements or sliding elements.

5.3.3.2 Screw, rotary lobe, liquid-ring and sliding vane-type compressors fall into this category.

5.3.3.3 Most of the rotary machines are specialized with limited applications.

5.3.3.4 A typical flooded screw-type rotary compressor, cross-sectional view is shown in Figure 9.



Major Components:
1. Housing
2. Male Rotor
3. Female Rotor
4. Radial Bearings
5. Axial Bearings
6. Mechanical Seal
7. Oil Pump
8. Hydraulic Thrust Compensating Piston
9. Capacity Control Slide Valve
10. Double Acting Hydraulic Piston

Figure 9. Typical Flooded Screw-Type Rotary Compressor

6. Basic Principles of Compression

6.1 General

6.1.1 The basic principles underlying the compression of gases in centrifugal and reciprocating compressors are discussed briefly below. These two types cover the majority of compressors currently in operation. Discussion about other types of compressors can be found in the references listed in Section 2.

6.1.2 Typically, the compression of gases is based upon the following:

a. Boyle’s Law

b. Charles’ Law

c. Avagadro’s Law

d. Ideal Gas Law

e. Bernoulli’s Equation

6.1.3 The thermodynamics of gas compression is derived from these laws above.

6.1.4 Few gases used in the process industry, such as air, follow ideal gas relationships. Therefore, real gas equations of state should be used to characterize the behavior of most process gases.

6.2 Centrifugal Compressors

6.2.1 In a centrifugal compressor, gas compression is most accurately modeled by a polytropic path defined by the relationship:

$$PV^n = \text{constant}$$

Where:

P = Absolute Pressure

V = Specific Volume

n = Polytropic Exponent

6.2.2 The above relationship is depicted on a pressure-volume diagram shown in Figure 10. An adiabatic process is also shown.

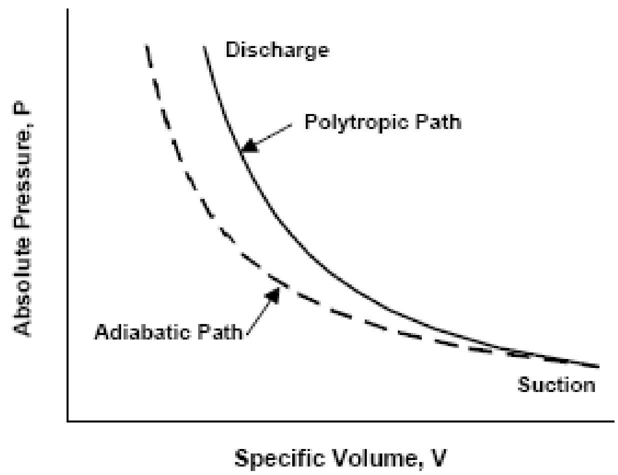


Figure 10. Pressure-Volume Diagram for Compression Processes

6.2.3 The compression of gas from suction to discharge conditions may be accomplished using a single impeller or multiple impellers depending upon the pressure rise required, the need to improve efficiency, and to limit pressure and temperature differentials. In either case, the polytropic curve in the figure represents the pressure-volume path that the gas follows as it flows through the compressor.

6.2.4 A section of a compressor may consist of multiple impellers. As the gas is compressed, its temperature increases. The gas may have to be cooled (using gas intercoolers) between compressor stages or compressor sections to keep its final discharge temperature below the maximum allowable temperature.

6.2.5 The efficiency of compression depends on the aerodynamic design and operating condition of the impellers and the stationary gas passages and elements. This efficiency can be defined as follows:

$$\text{Polytropic Efficiency, } \eta_{\mathrm{p}} = \frac{\text{Reversible Work Input}}{\text{Enthalpy Rise}}$$

6.2.6 The enthalpy rise of a gas may be obtained from a Mollier chart of the gas being compressed if it is a single component. Frequently, however, process gases are a mixture of several hydrocarbon gas elements. Computer programs are available to calculate required thermodynamic properties of gas mixtures.

6.2.7 The polytropic exponent is not a constant during the compression cycle. However, for most practical applications, it can be assumed to be a constant. Therefore, along a polytropic path, the ratio of the reversible work input to the enthalpy rise is constant.

6.2.8 The exponent n is related to the ratio k of specific heats Cp/Cv and the polytropic efficiency ηp approximately as follows:

$$\frac{n-1}{n} \cong \frac{k-1}{k} \times \frac{1}{\eta_p}$$

6.2.9 The work input per unit mass is typically referred to as polytropic head and can be evaluated as follows:

$$\text{Polytropic Head, } H_p = \frac{Z_{avg}\ R\ T_s}{MW(n-1)/n}\left\{\left(\frac{P_d}{P_s}\right)^{\frac{n-1}{n}} - 1\right\}$$

Where:

Z_{avg} = Average Compressibility Factor of the gas

R = Universal Gas Constant(8314 J/kg-°K) = (1545 ft-lbf/lbm-°R)

MW = Gas Molecular Weight

T = Absolute Temperature of the Gas

P = Absolute Gas Pressure

s, d = Subscripts that refer to suction/inlet and discharge conditions, respectively

6.2.10 The gas power can be calculated using the following relationship:

$$\text{Gas Power} = C\ \frac{wH_p}{\eta_p}$$

Where:

C = a constant depending on units used = 1/3600 for SI units (1/33,000 for U.S. Customary Units)

w = Mass Flow Rate

6.2.11 The shaft power is the sum of the gas power and frictional losses in bearings, seals and gearing (if present).

6.2.12 Typically, the adiabatic process should be suitable to define the compression of gases such as air that exhibit ideal gas behavior. In all other cases, the polytropic process should be used. The following should be noted:

a. Adiabatic Head is less than Polytropic Head

b. Adiabatic Efficiency is less than Polytropic Efficiency

6.2.13 The performance of centrifugal compressors is typically represented by head-flow curves and efficiency-flow curves. A typical example of these curves is shown in Figure 11.

6.2.13.1 Some manufacturers may also provide curves of pressure ratio and shaft power versus volumetric flow rate.

6.2.13.2 The end points of the curves represent two important limits of centrifugal compressors: surge and stonewall or choke point, defined as follows:

a. Surge is characterized by cyclic variation (and even reversal) of gas flow and discharge pressure, and it occurs if the flow is reduced below the surge point. It is typically accompanied by abnormal noise and vibration and it can lead to significant damage if the compressor's operating condition is not changed quickly to increase the flow.

b. A stonewall or choke point condition is encountered if the gas flow reaches sonic conditions somewhere in the compressor passages. In this condition, flow through the compressor cannot be increased further.

6.2.13.3 These performance curves vary typically with operating speed and gas composition (density, molecular weight, etc.).

6.2.14 A gas being compressed in a centrifugal compressor approximately follows the Fan Law which states that:

a. Volumetric flow rate is approximately proportional to the impeller rotating speed

b. Head is approximately proportional to speed squared

c. Shaft power is approximately proportional to speed cubed

6.2.15 The Fan Law relationships govern the behavior of centrifugal compressors as operating parameters change. However, deviation from the Fan Law increases as the number of stages increases.

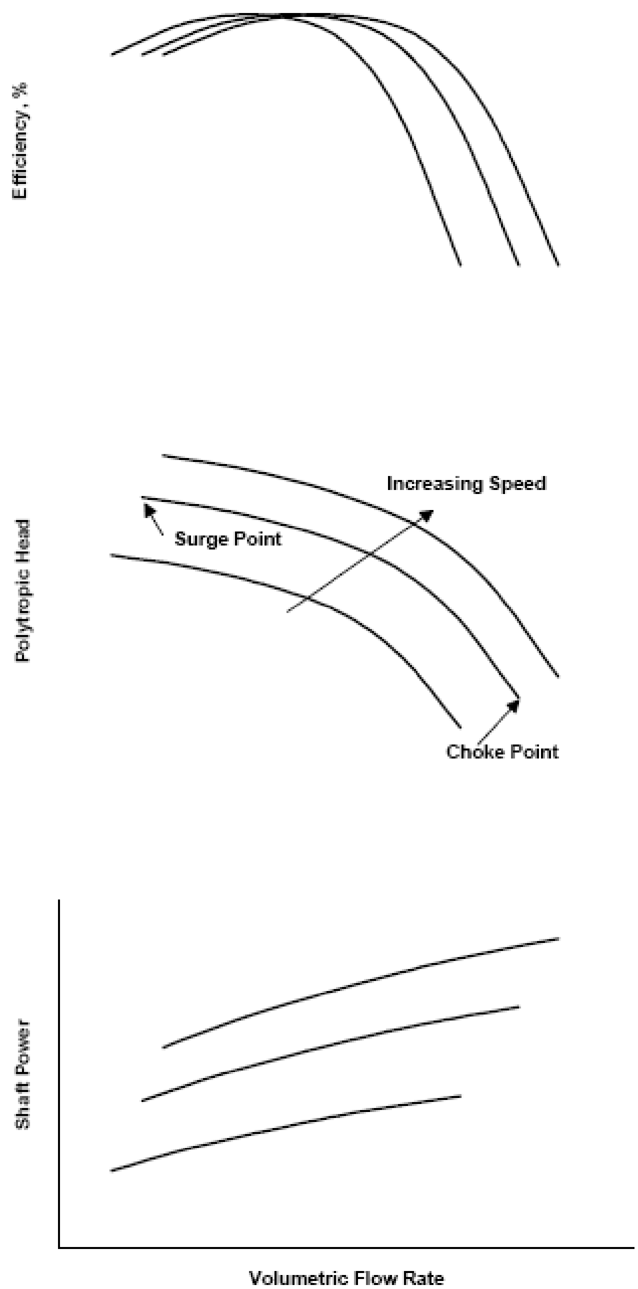


Figure 11. Typical Performance Map of a Centrifugal Compressor

6.3 Reciprocating Compressors

6.3.1 Compression in a reciprocating compressor can be depicted on a pressure volume diagram as shown in Figure 11.

6.3.2 For the most part, the process can be represented as an adiabatic process using the adiabatic exponent k as defined in the following relationship:

$$PV^k = \text{constant}$$

6.3.3 The adiabatic process was also depicted in the pressure-volume diagram shown in Figure 12.

6.3.4 Typically, reciprocating compressors do not have a performance curve like the one for centrifugal units. These compressors can deliver whatever pressure is required to overcome the discharge back pressure unless some material or design limit is reached. Therefore, reciprocating compressors should be provided with discharge relief valves to protect against exceeding safety parameters.

6.3.5 Compression stage ratios should typically be 3:1.

6.3.6 Reciprocating compressors do not have surge and choke limitations which are associated with centrifugal compressors.

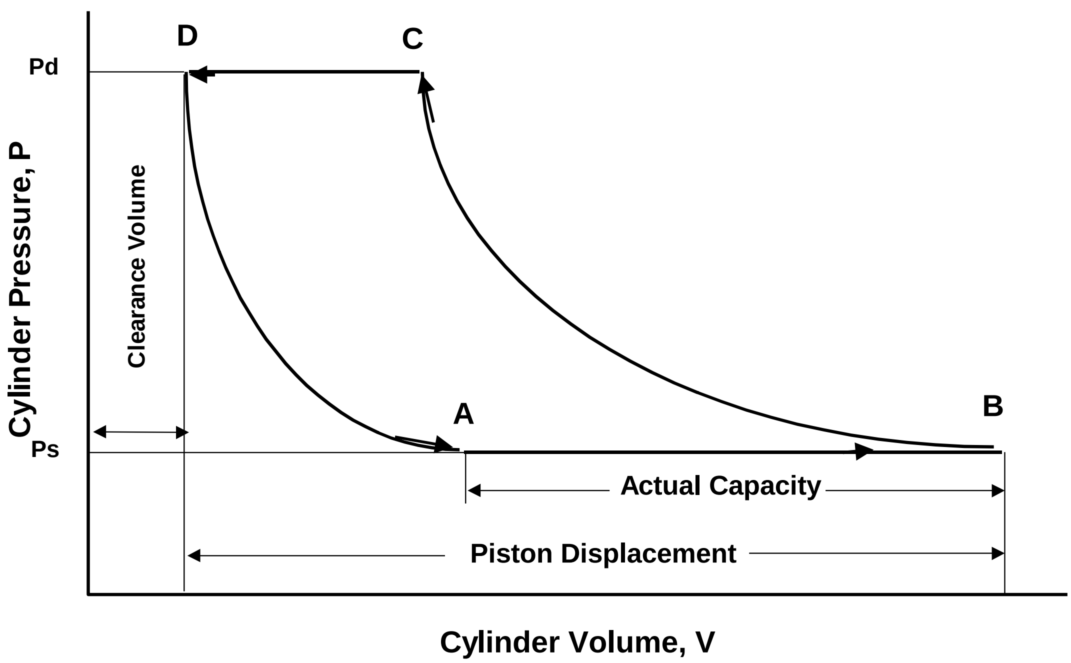


Figure 12. Pressure-Volume Diagram for Reciprocating Compressor

6.3.7 The work required for adiabatic compression is obtained in a manner similar to the polytropic process. However, the exponent n is replaced with the ratio of specific heats, k.

Therefore:

$$\text{Adiabatic Head, } H_{ad} = \frac{Z_{avg}\ R\ T_s}{MW(k-1)/k}\left\{\left(\frac{P_d}{P_s}\right)^{\frac{k-1}{k}} - 1\right\}$$

6.3.8 The volumetric efficiency E_V is defined as the ratio of the gas handled at inlet conditions to the theoretical volume displaced by the compressor expressed as a percent.

$$E_v = \frac{\text{Actual Capacity}}{\text{Piston Displacement}} = 100 - \{(Z_s/Z_d)r_p^{\,1/k} - 1\}c - L$$

Where: r_p = Pressure Ratio based on absolute pressures
c = Clearance Volume (see Figure 11)
L = Effects from internal leakage, gas friction, pressure drop through valves, etc. (approximately 5% for lubricated compressors)

6.3.9 The volumetric efficiency defines the volume flow capability of a cylinder in terms of inlet volume flow rate under a specific set of operating conditions.

6.3.10 It is important to recognize that the volumetric efficiency of a given cylinder configuration (fixed diameter, stroke, speed, internal clearance volume) is not a constant parameter.

a. Volumetric efficiency decreases with increasing pressure ratio across the cylinder.

b. Volumetric efficiency decreases with increasing internal cylinder clearance volume.

6.3.11 The volumetric efficiency (hence, capacity) of a given cylinder configuration can be altered by varying the internal clearance within the cylinder control volume.

6.3.12 The internal clearance within the cylinder control volume can be varied through the use of fixed or variable volume clearance pockets which add internal clearance volume to the cylinder if manually or automatically actuated for capacity control.

6.3.13 The theoretical discharge temperature is given by:

$$T_d = T_s \times \left\{ \frac{P_d}{P_s}^{\left(\frac{k-1}{k}\right)} \right\}$$

k is the ratio of specific heats (C_p/C_v)

6.3.14 Where temperatures and pressures are in absolute units. The compressor equipment manufacturers should be able to provide estimates of predicted discharge temperature and shaft power for given operating conditions. K is the ratio of specific heats.

7. Selection Considerations

7.1 Safety

Safety attributes that should be considered in the selection of compressor systems include:

a. Limiting gas properties (e.g., decomposition, flammability, toxicity). Normal compressor operation should not violate these limiting gas properties.

b. Compatibility of process gas with materials of construction (e.g., H_2S with copper, ethylene oxide with brass, oxygen)

c. Containment, collection and disposal of seal and vent gases

d. Over-pressure protection

e. Process and economic issues listed below should also be evaluated for safety implications

7.2 Process

7.2.1 Key process variables which should be considered in the selection of compressor systems include:

a. Mass flow rate

b. Absolute suction pressure and absolute temperature

c. Absolute discharge pressure and absolute temperature

d. Gas physical properties (e.g., composition, molecular weight, ratio of specific heats, compressibility)

e. Effects of process gas on the compressor system (e.g., corrosion, erosion, fouling, chemical reaction, coking, polymerization, temperature, condensation and liquid removal)

f. Machinery interaction with the process gas (e.g., lubricants, buffer fluids, seal media)

g. Solubility of the compressed gas in the lubricant in flooded screw applications.

h. Startup and shutdown process/mechanical conditions. These conditions can be very different than normal operating conditions and may significantly influence the selection process.

i. Preferred and acceptable methods of capacity controls

7.2.2 Typically, a normal operating point is specified in the selection process. This is the point at which the process is expected to operate. However, it is common to expect variations in operating conditions. It is, therefore, important to establish an expected range of the above variables during normal operating, startup and shutdown conditions.

7.3 Economics

Economic issues which should be considered in the selection of compressor systems include:

a. Life-cycle cost (trade-off between capital, operating cost and maintenance costs over life of equipment)

b. User and vendor capabilities and facilities for maintaining the equipment. This affects availability, maintainability, and mean time between failures. All of these combine to determine the expected equipment reliability

c. Level of sparing dictated by production needs (e.g., warehoused or installed)

d. Standardization of equipment and lubricants

e. Project life expectancy

7.4 Site and Location

7.4.1 The most common, but important location, attributes that should be considered in the selection of compressor systems include:

a. Available utilities

b. Accessibility to services and support

c. Environmental conditions

7.4.2 Depending on the application, other location variables may have to be considered as well.

7.4.3 The relative importance of the location variables should also be established to ensure that the best type and size of compressor is selected.

8. Lubrication Systems

8.1 General

8.1.1 Compressors typically require oil for bearing lubrication and bearing cooling. Oil is supplied by the lubrication system.

8.1.2 Components in a lubrication system should include:

a. Oil reservoir or tank

b. Lube oil pump or pumps

c. Oil filters

d. Oil coolers

e. Associated piping and instrumentation

8.1.3 The cleanliness of the lubrication system is critical to long term reliability of the compressor system.

8.1.4 Lubrication systems downstream of the filter should be constructed of stainless steel.

Note: Lubrication systems constructed completely of stainless steel eliminate many potential contamination issues caused by corrosion of non-stainless steel components.

8.1.5 If carbon steel components are used, provision for cleaning and passivating, such as a flushing jumper, should be included.

8.1.6 Flushing and cleaning of the lubrication system constitute major commissioning activities.

8.2 Centrifugal Compressors

8.2.1 Lubrication oil is required for fluid film and rolling element journal and thrust bearings.

8.2.2 Lubrication oil is supplied from a reservoir by a lube oil pump.

8.2.3 The pump may be compressor shaft-driven or independent. An independent main oil pump and an auxiliary oil pump should be preferred for centrifugal compressors.

8.2.4 Magnetic compressor bearings do not require oil lubrication.

8.3 Reciprocating Compressors

8.3.1 Reciprocating compressor cylinders may be lubricated or non-lubricated (with limits).

8.3.2 Lubricated reciprocating compressor cylinders utilize forced feed mechanical lubricators.

8.3.3 Cylinders of lubricated compressors should be pre-lubricated before starting the compressor.

8.3.4 If contamination of compressed gas by lubrication cannot be tolerated, non-lubricated cylinder designs are available for some applications that use piston rings and rod packing which do not require cylinder lubrication. Non-lubricated cylinder designs are not suitable for all applications.

8.3.5 To prevent carry-over of oil from frame crankcase to cylinder, a distance piece should be used between the frame and cylinder. The compressor crankcase also contains the oil for the frame lubrication. A wiper packing is installed between the crankcase and the distance piece to wipe excess crankcase oil from the piston rod.

8.4 Rotary Compressors

8.4.1 Rotary compressors may be lubricated or non-lubricated.

8.4.2 Flooded Screw Compressors

8.4.2.1 Flooded screw compressors use common oil systems for lubricating the rotors, sealing clearances, removing heat of compression and lubricating bearings.

8.4.2.2 A large amount of oil is circulated through the compressor. This oil is in contact with process gas and in addition to providing lubrication; it assists in removing the heat of compression from the process gas. After passing through the compressor, it is separated from the gas and returned to the lubricating points as well as to the compressor inlet.

8.4.2.3 Large flooded screw compressors may have timing gears to maintain rotor to rotor relationship.

8.4.3 Vane compressors should have a minimum amount of oil injected to lubricate the sliding vanes in the cylinder.

8.4.4 Straight Lobe and Dry Screw Compressors

8.4.4.1 Straight lobe and dry screw compressors do not use liquid for sealing the rotor clearances and driving the non-coupled rotor.

8.4.4.2 Lubrication is required for bearings and gears.

8.4.4.3 The compressors are provided with seals which separate the lubrication from the process.

8.4.4.4 The size and complexity of the lubrication and seal oil systems will vary considerably with compressor size, type and application.

8.4.4.5 The rotor-to-rotor relationship is maintained by timing gears on each rotor.

9. Seal Systems

9.1 General

9.1.1 Compressor seals restrict or prevent process gas leaks to the atmosphere or seal fluid leaks into the process stream over a specified operating range, including start-up and shut-down conditions.

9.1.2 Depending on the application, the seals may be of a dry or liquid lubricated type.

9.1.3 In applications where leakage can be tolerated, a labyrinth seal may be used.

9.1.4 Seals may require a buffer gas, seal oil, or both.

9.1.5 Manufacturers provide a variety of seals and sealing systems to meet the intended service.

9.1.6 If seals require higher pressure oil than required by bearings, seal oil requirements may be met by booster pumps or a separate system.

9.1.7 Seal selection also depends on whether the shaft is rotating or a reciprocating (piston rod).

9.1.8 If completely separate seal oil and lubrication systems are required, major items such as coolers, filters, reservoirs and pumps should be provided for each of the seal oil and lubrication systems.

9.1.9 If gas handled by a compressor causes contamination, chemical reaction or any other deterioration of oil, sealing systems should completely isolate gas from lubrication.

9.2 Dry Shaft Seals

9.2.1 Dry shaft seals are available in five general types:

a. Labyrinth

b. Bushing (Carbon Ring)

c. Film-Riding Face Seal

d. Contacting Face Seal

e. Circumferential Seal

9.2.2 A combination of two or more types can be used to achieve the required result.

9.2.3 Labyrinth seals should be used where high leakage rate can be managed or tolerated. Labyrinth seals have no pressure or speed limits and may incorporate buffer gas.

9.2.4 Bushing seals function by inhibiting flow through a close clearance around the shaft. The leakage rate can be relatively high. Bushing seals may be used as a single bushing or a series of bushings and may incorporate buffer gas and vents.

9.2.5 Film-riding face seals (also known as mechanical seals) can seal high pressures and allow high shaft speeds. They are used in single, tandem, or double arrangement and require clean process, flush, or buffer gas. Double seals with buffer gas pressure higher than process gas pressure prevent any leakage of process gas past the seal.

9.2.6 Contacting face seals and circumferential seals are less commonly used in compressors. They have limited shaft speed capability and shorter life.

9.3 Liquid Film Seal (Oil Film Seal)

9.3.1 Liquid film seals should be available in eight general types:

a. Labyrinth

b. Bushing (Carbon Ring)

c. Windback (Reversed Helical Groove Bushing)

d. Restricted Bushing (Trapped Bushing)

e. Film-Riding Face Seal

f. Contacting Face Seal

g. Circumferential Seal

h. Lip Seal

9.3.2 The liquid film seal or oil film seal is particularly applicable to high speed machines.

9.3.3 The actual seal is accomplished by a thin oil film supplied by the seal oil pump to a space between the rotating and stationary seal elements.

9.3.4 This oil contacts process gas and is degassed before returning to the oil reservoir.

9.3.5 Contaminated oil should be reconditioned or discarded.

9.3.6 If handling hazardous, toxic, or emission-regulated gases, the seal also should prevent gas leakage to the atmosphere after the compressor has tripped due to seal oil system failure.

9.3.7 Various devices within the seal support system are available to assure that the compressor seal contains the gas at a standstill, even if no seal oil is being pumped to the seal.

9.3.8 Elevated seal oil tanks can provide for the necessary static differential pressure of the fluid above the sealing pressure for a sufficient time to allow the compressor to be depressurized before the elevated tank oil supply is depleted.

9.4 Packing Glands

9.4.1 Packing glands are another type of sealing system and should be used in reciprocating compressors to control gas leakage from the cylinders.

9.4.2 The packing gland contains a series of segmented packing rings around the piston rod.

9.4.3 A purge gas, such as nitrogen, may be used to provide a positive seal from the atmosphere and improve the venting of the process gas from the sealing area.

9.4.4 The leakage is typically vented to a low pressure vent system or to a low pressure flare header.

9.4.5 Care should be exercised to ensure that the packing gland pressure is compatible with the flare header pressure.

10. Ancillary Equipment

10.1 Ancillary equipment and supporting systems that should be part of the selection and operation of compressors include the following:

a. Drivers

b. Couplings

c. Gearboxes

d. Cooling Systems (inter- and aftercoolers)

e. Pulsation Suppressors

f. Separators (process gas, seal and lube oil)

g. Control Systems including Anti-Surge Systems

h. Monitoring Systems (performance and vibration)

i. Piping Systems including Safety Valves

j. Foundation, Grouting and Mounting Plates

k. Suction Strainers and Filters

l. Silencers

10.2 Discussion of the above topics is beyond the intended scope of this Practice. However, all of them play a very important role in the selection, design, installation and reliable operation of compression systems.

10.3 Figure 13 summarizes the important sub-elements of some of th ancillary equipment items.

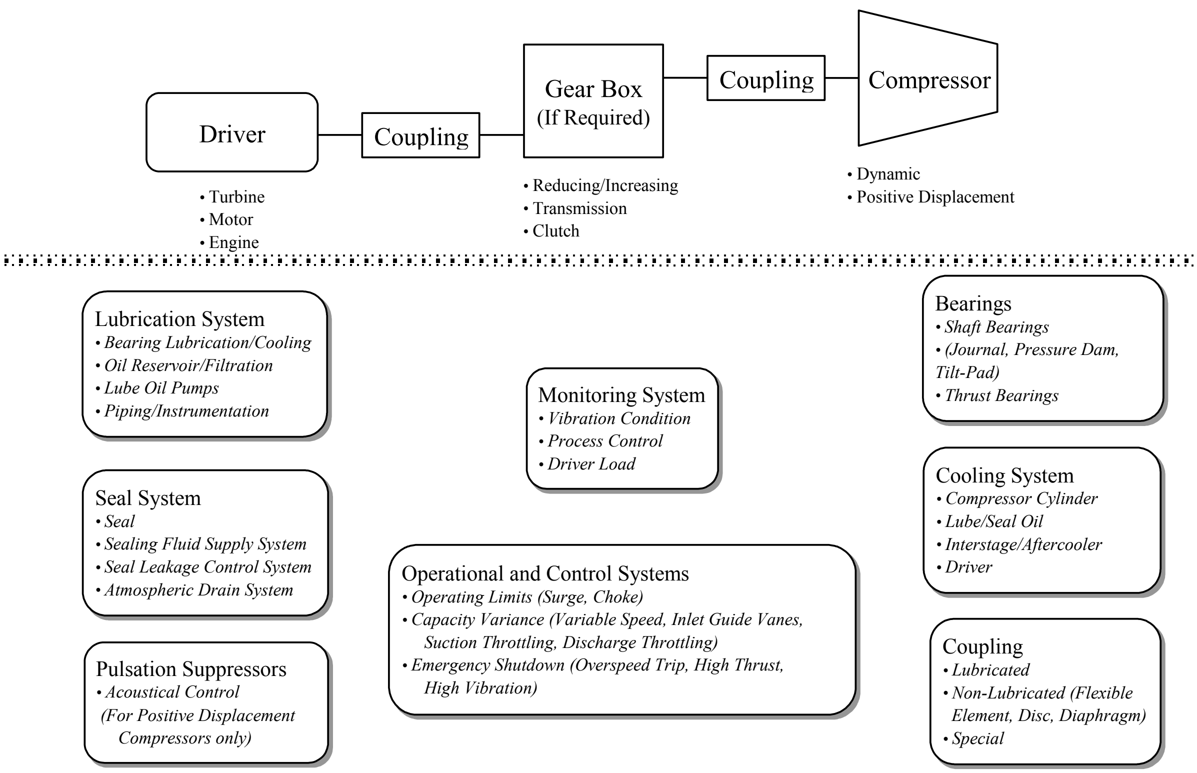


Figure 13. Ancillary and Supporting Systems for Compressor Applications

10.4 Detail specifications and Practices for specific equipment types covered in this guideline may be found in the following list:

10.4.1 Process Industry Practices (PIP)

- PIP REIE686/API RP686 - *Recommended Practices for Machinery Installation and Installation Design*
- PIP REEE003 - *Application of General Purpose Non-Lubricated Flexible Couplings*

10.4.2 Application of Industry Codes and Standards to Select Ancillary Equipment

- American Petroleum Institute (API)

  Note: Selected parts of API data sheets and specifications can be used to purchase non API compliant equipment due to the thoroughness of the documents.

  - *API 611* can be used to develop specific requirements and data sheets for horizontal or vertical turbines that are used to drive equipment that is usually spared, relatively small in size or is in non-critical service. They are generally used where steam conditions will not exceed a pressure of 48 bar (700 pounds per square inch gauge) and a temperature of 400°C (750°F) or where speed will not exceed 6000 revolutions per minute.
  - *API 612* can be used to specify requirements and gives recommendations for the design, materials, fabrication, inspection, testing and preparation for shipment for steam turbines in special-purpose services. It also covers the related lube-oil systems, instrumentation, control systems and auxiliary equipment.
  - *API 613* can be used to specify the minimum requirements for special-purpose, enclosed, precision single- and double-helical one- and two-stage speed increasers and reducers of parallel-shaft design
  - *API 614* can be used to specify the minimum requirements for lubrication systems, oil-type shaft-sealing systems, dry gas face-type shaft-sealing systems, and control oil systems for general or special purpose applications.
  - *API 617* can be used to specify the minimum requirements for axial compressors, single-shaft and integrally geared process centrifugal compressors and expander-compressors.
  - *API 618* can be used to specify the minimum requirements for reciprocating compressors and their drivers for handling process air or gas with either lubricated or non-lubricated cylinders. Compressors covered by this standard are of moderate- to-low speed and in critical services. Also covered are related lubricating systems, controls, instrumentation, intercoolers, aftercoolers, pulsation suppression devices, and other auxiliary equipment.

- *API 619* can be used to specify the minimum requirements for dry and oil-flooded helical lobe rotary compressors used for vacuum or pressure or both.
- *API 670* can be used to specify the minimum requirements for a machinery protection system measuring radial shaft vibration, casing vibration, shaft axial position, shaft rotational speed, piston rod drop, phase reference, overspeed, and critical machinery temperatures.
- *API 671* can be used to specify the minimum requirements for special-purpose couplings that transmit power between the rotating shafts of two pieces of equipment.
- *API 672* can be used to specify the minimum requirements for constant-speed, packaged, general purpose and special duty integrally geared centrifugal air compressors, including their accessories.
- *API 677* can be used to specify the minimum requirements for general-purpose, enclosed single and multistage gear units incorporating parallel shaft helical and right angle spiral bevel gears.

- American Society of Mechanical Engineering (ASME)
  - *ASME B19.3* can be used to determine the requirements for safety devices and protective facilities to prevent compressor accidents as a result of excessive pressure.
  - *ASME PTC-10* can be used to determine the thermodynamic performance of an axial or centrifugal compressor or exhauster doing work on a gas of known or measurable properties under specified conditions.
- Association of German Engineers (VDI)
  - *VDI 2045* can be used for acceptance and performance tests on turbo compressors and displacement compressors.
- Gas Processors Suppliers Association
  - *Engineering Data Book* is a source of basic design information along with data and procedures that can be used to determine operating and design parameters.

10.4.3 Other References

- *Compressors: Selection and Sizing*, 2nd ed., Gulf Publishing Co., Royce N. Brown
- *Compressor Application Engineering*, Volume 1, Compression Equipment, Gulf Publishing Co., Pierre Pichot
- *Compressor Application Engineering*, Volume 2, Drivers for Rotating Equipment, Gulf Publishing Co., Pierre Pichot
- *Leak-Free Pumps & Compressors*, Elsevier Advanced Technology, Gerhard Vetter