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Revision Memo

11/03	Throughout document: Process Compression Stage changed to Compressor Section Gas Seal Max Pressure from 1450 to 3000 psi GP/API titles revised. Reworded to show increasing use of Integrally Geared Compressors in process service. Included mixed flow impellers in Impeller Design paragraphs. Clarified origination of Figure 16. Placed emphasis on use of gas seals over oil seals. Gas Seal Standards changed to reflect API & GP revisions. Clarified gas seal paragraphs to reflect API & GP revisions. Reworded Parallel Operation paragraph slightly. Changed testing types per PTC10 revision. Corrected required and optional tests per API. Table 1: Changed per new gas seal limits. Figure 14: Added Mixed Flow Impeller. Figure 17: Process Compression Stage to Compressor Section Figure 19-22: Used MW for mole weight rather than M. Figure 29: Unefficient replaced with inefficient.
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SCOPE

This section presents aspects of centrifugal compressors relating to compression service design. Background material is included to acquaint the compression service designer with the fundamentals of centrifugal compressors, and detail design considerations are provided to aid with the selection of specific design features of centrifugal compressors and associated auxiliary systems.

REFERENCES

DESIGN PRACTICES

(Besides other subsections of this section).

Section XV-B Minimizing the Risk of Fire, Explosion or Accident

Section XVIII Guidelines and Considerations for Industrial Hygiene

GLOBAL PRACTICES

- ▶ GP 03-03-03 Flushing and Drain Piping for Centrifugal and Rotary Gas Compressors.
- GP 03-03-10 Strainers for Mechanical Equipment.
- ▶ GP 10-03-01 Axial and Centrifugal Compressors and Expander-Compressors
- GP 10-03-02 Packaged, Integrally Geared Centrifugal Air Compressors.
- GP 10-14-01 Lube and Seal Oil Systems for Machinery.
- GP 15-01-02 Instrumentation for Compressors and Drivers.
- GP 15-07-02 Protective Systems
- GP 15-11-01 Control Panels

OTHER REFERENCES

- ▶ API-614 Lubrication, Shaft-Sealing, and Control-Oil Systems and Auxiliaries for Petroleum, Chemical and Gas Industry Services.
- ▶ API-617 Axial and Centrifugal Compressors and Expander-Compressors for Petroleum, Chemical and Gas Industry Services.
- API-672 Packaged, Integrally Geared Centrifugal Air Compressors for Petroleum, Chemical and Gas Industry Services.
- ▶ ASME-PTC10 Performance Test Code on Compressors and Exhausters.
- CPEE 241 CompCalc, PC Program for Compressor Network Simulation and Design.
- Exxon Engineering Standard EXES 10-15-1B, Dry Gas Seals Purchase.
- Hallock, Donald C., Centrifugal Compressors - The Cause of the Curve, Air and Gas Engineering, 1 (1), January 1968.
- Hampel, John, What Control Engineers Need to Know About Anti Surge Protection, Controls Expo'84, May 24 1984.
- Staroselsky, N., and Ladin, L., Improved Surge Control for Centrifugal Compressors, Chemical Engineering, May 1979.
- Staroselsky, N., and Carter D., Protecting Multicase Compressors, Hydrocarbon Processing, March 1990.

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DEFINITIONS

Compressor Stage - A compressor stage consists of a set of inlet guide vanes, a single impeller, a diffuser and a return channel. Compressor stages are employed singly, in series and in parallel within the same centrifugal compressor body.

♦ **Compressor Section** - A compressor section is the compression step between two adjacent pressure levels in a process system. The compressor section may consist of a single compressor stage or multiple compressor stages arranged in series between which there is no intercooling, loss or gain in flow, and whose rotational speeds are in fixed relationship (ratio) to each other. The compressor section is also referred to as "process compression stage" by process personnel.

Maximum Allowable Working Pressure - This is the maximum continuous pressure for which the manufacturer has designed the compressor when it is operating at its maximum allowable temperature.

Head - Work necessary to raise the pressure of a gas.

Settling Out Pressure - This is the maximum pressure level which can be reached in the compressor following a compressor trip and before the pressure is vented.

Sealing Pressure - The pressure of the compressed gas against which the seal must operate.

Maximum Dynamic Sealing Pressure - The maximum pressure for which the compressor shaft seal should be designed with the compressor rotor in operation.

Maximum Static Sealing Pressure - The maximum pressure for which the compressor shaft seals should be designed with the compressor rotor stationary.

Note: API compressor data sheets specify pressures in psi, bar, and kPa. In this section, bar is the metric unit used for pressure.

BACKGROUND

GENERAL

Centrifugal compressors handle more compression service flow in hydrocarbon processing plants, than all other compressor types combined. They are applied in approximately the same numbers as reciprocating compressors, but each machine handles typically between 10 to 50 times as much gas volume as does each reciprocating compressor. Despite this difference in volume flow handling capability, the centrifugal machine is a compact unit; and due to its high level of reliability, is normally installed as a single unspared machine. Services where it is fairly common to find a centrifugal compressor installed include FLEXICRACKING unit light ends gas, POWERFORMING unit recycle gas, process plant refrigeration and gas pipeline transmission.

SERVICEABILITY AND SPARING

See Design Practice XI Section C, **Compression Machinery Serviceability and Sparing**, for reliability factor information and recommended sparing practices.

OPERATING PRINCIPLES

Compressor can be classified as either dynamic or positive displacement machines. The basic principal of any type of dynamic compressor is the conversion of gas velocity to pressure. Centrifugal compressors belong to the dynamic category.

The operating principals of the centrifugal compressor are best understood by following the path of the fluid stream through the compressor. See [Figures 1 through 5](#). In Figure 1, gas enters the compressor through the inlet nozzle connected to the plant pipework. From the inlet nozzle, it travels into an annular chamber called the inlet volute where it is directed towards the center by guide vanes and baffles. Referring to [Figure 2](#), the gas leaves the inlet volute and enters the eye of the rotating impeller in an axial direction. In the impeller, the stream turns perpendicular to the shaft axis and energy is transferred to the gas by the rotating impeller blades which increase the gas stream velocity. Upon leaving the impeller, the gas enters a circular chamber called a diffuser where it loses velocity and increases in pressure. See [Figure 3](#). The gas stream is turned through 180 degrees in the return bend at the end of the diffuser section and then move radially inward in another annular passage known as the return channel. Vanes are installed in this passage to straighten the spiral gas flow and thus obtain a radial outlet from the return channel. These vanes are often referred to as de-swirl vanes. Once it leaves the return channel, the gas is turned to provide an axial entry to the next impeller and the compression process is repeated. In some designs a further set of vanes are provided at the impeller inlet(s) to modify the angle of gas entry. These are referred to as inlet guide vanes. The gas leaving the last impeller flows into a diffuser and then into an annular chamber called a discharge volute. See [Figures 4 and 5](#). The discharge volute collects the gas stream from the diffuser and guides it to the discharge nozzle.

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COMPRESSOR PERFORMANCE CURVES

Curves that define the performance of a centrifugal compressor in terms of head or discharge pressure versus flow rate are often called compressor characteristic curves. The manufacturers normally provide curves for each compressor section.

These curves are produced by combining the characteristics of the individual compressor stages (wheel curves). [Figure 6](#) shows a typical set of performance curves for an 8 stage centrifugal compressor.

HEAD CAPACITY CHARACTERISTIC CURVES

The three important components of the head capacity curve are slope, choke (or stonewall), and surge. See [Figure 7](#) for an illustration of head capacity curve terminology.

Curve Slope - The slope of the head capacity curve for a compressor stage is determined by impeller blade geometry and the characteristics of the compressed gas. The curve slope for a compressor section is further influenced by rotational speed and the number of compressor stages that make up the compressor section. Speed and gas characteristics affect the curve slope due to their impact on gas volume reduction for a single compressor stage. These changes are compounded as the gas moves from one compressor stage to the next.

The change in curve shape with changing gas characteristics is significant when attempting to predict compressor performance over a range of gas mixtures and compressor speeds. Performance curves generated for one gas mixture cannot be used for a significantly different gas mixture at the same compressor speed. Similarly, for a fixed gas composition, performance curves cannot be used to predict performance at a significantly different speed.

A POWERFORMING unit recycle gas machine is a typical example where changes in gas composition occur. Normal operation is on hydrogen rich process gas where molecular weight is in the 7-10 range while during the catalyst regeneration cycle, the molecular weight is typically 30. The low MW curve will be flatter and at any given flow will produce less head than the high MW curve. The low MW curve will, however, have a wider operating range from surge to choke. A similar effect is seen with changing speed where curve slope becomes steeper and operating range reduces as speed increases. Changes in curve shape and operating range will become more significant as the number of impellers in a compressor section increases.

Choke - For every compressor curve there is a maximum flow limit often referred to as choke or stonewall. This maximum flow limit is approached as the velocity of the gas stream relative to the impeller blades nears the sonic velocity of the gas mixture being compressed. For gases with heavier molecular weights (low sonic velocity) the region approaching choke is identifiable by a rapid increase in the slope of the head capacity curve.

Surge - The minimum flow limit of a compressor stage is determined by the unstable cyclic phenomenon known as surge. Operating a compressor in surge can result in serious damage. Surge occurs when a compressor loses its ability to develop a pressure equal to that existing at its exit, and a reversal of gas flow takes place. The reversal of flow continues until the pressure at the exit drops to a level where the compressor can again establish forward flow. Having reached this point, the compressor continues to move the gas stream forward until it again reaches the point where the exit pressure exceeds the pressure that the compressor is capable of producing, and another flow reversal occurs. This cycle continues to repeat itself until the flow to the compressor is increased or the pressure ratio is decreased to a point where compressor can break the cycle and operate in the stable region of the head capacity curve.

Stability and Turndown - The term "stability" is used to define the available operating range from the Normal duty point to surge flow. Stability is defined in percentage terms as being $100\% - (\text{Surge Volume Flow} / \text{Normal Volume Flow} \times 100)$, when operating at design inlet temperature and pressure and with design gas composition.

Another term used to define the operating range of a machine is turndown which, unlike stability, takes into account the effects of control methods such as suction throttling or speed control. This term is better related to mass flow variation than volume flow especially when a suction throttle valve is employed. Turndown with respect to Normal flow is defined as $100\% - (\text{Surge Mass Flow} / \text{Normal Mass Flow} \times 100)$, when operating at design inlet temperature and with design gas composition.

Head Capacity Curves for Variable Speed Machines - A typical variable speed performance curve is shown in [Figure 8](#). By variation of speed, a wide range of operating conditions are achievable. The performance of the centrifugal compressor at speeds other than design can be approximated by the fan laws.

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FAN LAWS

Centrifugal compressor performance can be estimated for points other than the normal flow and speed point by use of the fan laws. These laws reflect the basic operating principles of dynamic machines that volume capacity is proportional to the peripheral blade speed, head is proportional to the square of the peripheral blade speed and power to cube. To use the laws, compressor performance curves must be expressed in terms of head versus inlet volume flow.

$$\frac{Q_2}{Q_1} = \frac{N_2}{N_1}$$

$$\frac{H_2}{H_1} = \left(\frac{N_2}{N_1} \right)^2$$

$$\frac{GHP_2}{GHP_1} = \left(\frac{N_2}{N_1} \right)^3$$

The fan laws produce approximate results which are reasonable in the 80 to 105% speed range. The accuracy of the predicted performance decreases as the speed change increases. The accuracy of these predictions also decreases with heavier gases as the effects of volume reduction ratios become more pronounced.

CENTRIFUGAL COMPRESSOR TYPES COMMONLY APPLIED

Multi-Stage Single Shaft - The most common type of centrifugal compressor found in process plant applications is the multi-stage single shaft compressor. This category is divided into two groups based on casing joint orientation. These are axial split (Figure 9) or radial split machines (Figure 10). Axial split machines have casing joints that are parallel to the shaft axis and are often specified to have all process connections located in the lower half of the casing. This permits the removal of the upper casing half for maintenance purposes without disturbing the process piping. Radial split machines have casing joints that are transverse to the shaft centerline and may have process nozzle connections located in any position around the circumference of the casing. Maintenance of radial split machines requires removal of the compressor internal assembly (bundle) from the non-drive end of the compressor. Sufficient plot space must be provided for this operation. Radial split multi-stage machines are often referred to as barrel compressors.

It is normally left to the manufacturers to select casing joint orientation based on operating experience. Typically for inlet flows up to 10,000 ICFM (17,000 inlet m³/h) and pressures below 600 psig (41 barg), axial split casings are used. Radial split machines would normally be used at higher pressures within this flow range. For higher flow rates, the axial split machine is typically limited to pressures of 300 psig (21 barg). These guidelines reflect the fact that, as casing size or internal pressures increase, it is more difficult to seal an axially oriented joint.

For services where the partial pressure of hydrogen at the casing maximum allowable working pressure (MAWP) exceeds 200 psig (14.0 barg), API-617 requires that a radially split casing be used.

Single Stage Overhung - For services with high volumetric flows and low head requirements, single stage overhung machines can be employed. See Figure 11. These machines use a cantilevered rotor design and have an axial inlet nozzle and a radial discharge nozzle. They are available in direct drive or integrally geared configurations.

High Speed Single Stage Vertical - These compressors are similar to vertical in line high speed pumps. See Figure 12. They operate at speeds of up to 33,000 RPM. Their range of application is determined by driver size and discharge pressure limitations. For further information refer to Design Practice XI Section B, Compressor Type Selection.

- ♦ **High Speed Integrally Geared** - Typically used in plant or instrument air services, they are becoming increasingly popular for use in process service. These machines employ a central gear unit with multiple high speed shafts located around the gear wheel periphery. The compressor impellers can be mounted on both ends of these shafts or have a single impeller per shaft. See Figure 13. For air services, interstage coolers and separators are typically provided with the machine as a package. GP 10-03-02 covers the requirements for this machine type in utility and instrument air service. GP 10-03-01 contains the requirements for this machine type in process service.

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CENTRIFUGAL COMPRESSOR DESIGN FEATURES

- ◆ **Impeller Design** - Closed or semi-open type impellers are commonly used in process gas applications. The semi-open impeller has no cover. See Figure 14. Impellers can be further classified as two dimensional (without inducers), or three dimensional (with inducer). The blade leading edge in a two dimensional impeller does not start until the gas flow in the impeller is moving radially. For the three dimensional impeller, the blades are extended such that the gas stream encounters the leading edge while either moving in the axial direction or while it is in the process of turning perpendicular to the compressor shaft. Adding an inducer section to the blades increases the head capability of an impeller by permitting operation at higher speeds and tends to steepen the compressor stage performance curve. Even higher flows are being obtained through the use of mixed flow impellers. In these impellers, the flow at the discharge is not perpendicular to the shaft but has an axial component.

Most impellers used in process plant applications are the closed, two or three dimensional type with backward leaning blades. Impellers for high flow or heavy gas services are typically the three dimensional, closed, backward leaning blade type. For very high flow, head or speed applications impellers are typically the semi-open, three dimensional design.

Rotor Support - Conventional multi stage centrifugal compressors are arranged with the impellers between bearings. The bearing brackets are attached to the casing which is nominally center line supported. Movement of the rotor center line, as the casing temperature changes from ambient to operating levels, is therefore minimized and coupling alignment is better preserved. Single stage overhung compressor rotors are supported from a single bearing housing containing a thrust bearing and a pair of radial bearings. The impeller is cantilevered beyond the pair of radial bearings.

Balance Pistons and Associated Leakage - Multistage centrifugal compressors develop a considerable thrust load from the discharge end toward the inlet due to the pressure difference across each of the impellers. To minimize the load that must be carried by the thrust bearing, a cylinder (balance piston) is installed on the shaft at the discharge end of the rotor. The balance piston has discharge pressure applied to its inboard side and inlet pressure to its outboard side. Careful selection of the balance piston diameter results in a thrust force approximately equal and opposite to that created by the impellers. Inlet pressure is maintained outboard of the balance piston by providing a labyrinth seal at the piston and piping the chamber on its outboard side to the compressor inlet. This pipe is referred to as the balance piston line. Sizing and cleanliness of this line is critical to proper compressor operation. See Figure 15. An alternative means of limiting thrust bearing loading is to arrange the compressor stages in two axially opposed groups (i.e., back to back), with the objective of balancing the thrust, although this typically does not eliminate the need for a balance piston. This construction method is frequently employed on very high pressure services and on intercooled compressors with two compressor sections installed within a single casing.

Seals - Figure 15 shows some of the various sealing locations in a centrifugal compressor. In addition to the main shaft seals located at the point where the compressor shaft exits the pressure containing casing, there are seals located at the balance piston, division walls (in multi-section casings), impeller eyes, and impeller hubs. For seals other than the main shaft seals, labyrinths of varying designs are normally used. In these locations, the prime objective is to maintain differential pressures and restrict rather than completely eliminate flow. The function of the main shaft seals is either to reduce or eliminate the outward leakage of process gas and/or to prevent inward leakage of air.

BASIC DESIGN CONSIDERATIONS

COMPRESSOR SECTION

Determining the number of compressor sections required for any service, is a choice that is influenced by compressor discharge temperature, compressor head generating capability, and the need for process connections at intermediate pressure levels.

Discharge Temperature - The factors which limit the allowable discharge temperature are as follows:

- **Gas Fouling** - Centrifugal compressor performance is sensitive to flow passage restrictions; therefore, fouling of the compressor internals due to polymerization of gas must be avoided. This limits the allowable discharge temperature to approximately 250°F (120°C) in acetylene, diolefin and olefin rich streams. See Section XI-A, Fouling Tendency.
- **Material and Construction Limitations** - The materials used for parts such as labyrinths and O-rings, limit the compressor casing maximum allowable discharge temperature. Large axially split designs and multi-nozzle cast casings are susceptible to distortion as the casing temperature rise increases. Casing distortion can cause casing joint leaks and alter critical compressor internal clearances. For these reasons, the maximum discharge temperature of a compressor section should be limited to 350°F (175°C). For applications in excess of this, machinery specialists should be consulted. For guidance in determining centrifugal compressor discharge temperatures, see Design Practice XI Section D, Compression Calculations.

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BASIC DESIGN CONSIDERATIONS (Cont)

Specially designed, single stage overhung machines are commonly used for hot regeneration gas service up to 500°F (260°C) with little modification other than to seals and have been applied up to temperatures of 800°F (425°C).

- ♦ **Head Capability** - [Figure 16](#) gives typical values of head that can be obtained from an individual compressor stage as a function of gas molecular weight. While the impeller design sets the head capability, the molecular weight limits head that can be obtained. A Mach number at the impeller discharge of one is a practical limit for most compressor designs. Since heavier mole weight gases have lower sonic velocities, the tip speed of impellers in these services is lower restricting the head produced.

A compressor stage consists of just one impeller and diffuser combination. Limits on shaft length place a practical limit on the number of compressor stages that can be installed in a single casing. Most designs limit the number of compressor stages in a single casing to 10; however, for casings with multiple compressor sections (intercooling or side streams), this number should be reduced by one for each additional inlet or outlet nozzle required. For marginal applications where these guidelines suggest more than a single casing is necessary, the possibility of using a single casing should be reviewed with a machinery specialist.

Single stage overhung machines with gearing can produce heads up to 40,000 ft (12,200 m) as can the individual stages of the integrally geared centrifugal air compressors. The vertical in-line high speed machines can produce heads of up to 30,000 ft (9,200 m).

Process Requirements - Process requirements influence the number of compressor sections. The need for side streams in a refrigeration process and the use of two separation stages in a production facility are typical examples.

Compression Stage and Casing Arrangements - [Figure 17](#) illustrates some common casing and compressor section arrangements.

COMPRESSOR DUTY DEFINITION

Compressor Operating Conditions - Centrifugal compressor performance is particularly sensitive to variations in operating conditions. All required operating conditions must therefore be given in the Design Specification. See [Design Practice XI Section A, Operating Conditions](#).

The centrifugal compressor is designed to produce a given head at a given inlet volume flow rate. Failure to completely define all operating conditions can result in the purchase of a compressor with too little head generating capability and insufficient discharge pressure. For example, consider the performance curve for a fixed speed compressor section shown in [Figure 18](#). Curve AB is for a machine designed to operate on air with an inlet pressure of 14.5 psia (1.0 bara), inlet temperature of 100°F (38°C), a dry air MW = 29 and a specific heat ratio (k) of 1.4. Normal inlet volume flow, discharge pressure and power are shown as being the 100% case. If the inlet temperature is reduced to 40°F (4.5°C) and all other conditions remain the same the discharge pressure will increase to approximately 106% of Normal. The reduced inlet temperature will also result in a power increase, see Curve 2. If inlet pressure is reduced to 12 psia (0.8 bara) and all other variables are held per the AB base case, the discharge pressure and power requirements will fall by the amount shown by Curve 3. The next change will be to raise the gas MW to 40 with all other variables as the base case. This will have the impact as defined by Curve 4 where significant increases in discharge pressure and power requirements can be seen. Finally the specific heat ratio of the gas will be reduced to 1.1 which will change the machine performance per Curve 5. This shows a small increase in discharge pressure but no impact on compressor power requirements.

For evaluation of the impact of gas property changes on machine performance for a specific case, the calculation methods defined in [Design Practice XI Section D, Compression Calculations](#), should be used.

Specifying Curve Shape

- **Pressure Ratio Rise** - To ensure that a centrifugal compressor will be controllable and will operate stably within its required operating range, the head-capacity curve of the compressor must rise continuously from Normal flow to the flow rate at which surge begins. There is no rational method available for defining the minimum curve rise necessary to make the machine controllable for a given installation. It is, however, possible to define the minimum curve rise that a well designed compressor can reasonably be expected to provide. The most convenient method of specifying curve rise is in terms of percent increase in pressure ratio (rr), e.g., a machine that produces a pressure ratio of 3.0 at Normal flow and 3.3 at surge flow has an increase in pressure ratio of 10%.

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BASIC DESIGN CONSIDERATIONS (Cont)

The pressure rise capability of a compressor is dependent on the number of compressor stages and the sonic velocity of the gas steam at inlet conditions. [Figures 19 & 20](#) provide a graphical relationship between pressure ratio, gas sonic velocity and the number of compressor stages. Sonic velocity can be obtained by either using the equations given in [Figures 19 & 20](#) or by using the gas properties option in the **EMRE PC Program for Compressor Network Simulation and Design, CompCalc**. The number of stages can be estimated using [Figure 16](#) in conjunction with the head requirement for the service. All well designed compressors can normally exceed the values produced by using this technique; therefore, the Design Specification should quote the figure as the minimum acceptable rise in pressure ratio. **GP 10-03-01** makes reference to this requirement and states that values will be specified by the Purchaser.

- Stability Range** - The Design Specification should state the required compressor inlet volume capacity at the surge point as a specified percentage of normal capacity at normal speed and normal (unthrottled) inlet conditions.
 Specifying a reasonable stability range will discourage the vendor from sacrificing operating flexibility and efficiency by placing the normal operating point close to surge. [Figure 21](#) can be used to establish the maximum percentage value of Normal flow at which surge should occur. The number of compressor stages can be estimated using [Figure 16](#). Well designed compressors should have surge points at flow rates no higher than defined by this method; therefore, the Design Specification should quote the figure as the maximum acceptable volumetric capacity at the surge point. **GP 10-03-01** also makes reference to this requirement.
- Choke** - To prevent curve shape selections that place the normal operating point in the region of the curve where head drops dramatically with increasing flow, **GP 10-03-01** specifies that the head capability of the machine at 115% of Normal flow should not be less than 85% of that produced at Normal flow. The ability of a compressor manufacturer to meet this requirement is dependent on the molecular weight of the gas stream. It should be recognized that it is not always possible to meet this requirement for high molecular weight gases; however, the intent is to ensure that the manufacturer selects compressor stages loaded at or near their best efficiency flow point.

Manufacturers Performance Curves - **API-617** requires the vendor to provide compressor section characteristic curves showing discharge pressure, power, polytropic head, and polytropic efficiency versus inlet volumetric flow. In addition to this the manufacturer must provide curves for variable speed machines at minimum speed, 80, 90, 100 and, 105 % of rated speed. Curves can also be produced to show pressure ratio and suction pressure versus inlet volume flow or all the above variables versus mass flow. If any curves, not requested by **API-617**, are needed they should be requested in the Design Specification.

When reviewing curves for parameters such as discharge pressure, inlet pressure or mass flow, care must be taken. These curves are dependent upon an assumed set of inlet or discharge conditions and the reviewer must ensure that these conditions are consistent with the analysis being performed. This is particularly true for curves of sections other than the first of a multi-section machine or curves said to define the combined performance of all compressor sections as the performance of the interstage equipment is assumed.

INTERSTAGE DIAPHRAGMS

When a centrifugal compressor casing with intermediate main process connections is specified, the Design Specification must provide the maximum and minimum pressures at each connection considering all operating, startup, shutdown and trip conditions. This data is used by the manufacturer to design the compressor internals to withstand the maximum differential pressures imposed.

To illustrate, consider a compressor casing specified to have more than one compressor section with the stages intercooled. The casing will have an internal wall or diaphragm separating the compressor sections. Depending upon the piping and system arrangement, it may be possible to subject the diaphragm to higher differential pressures than the compressor manufacturer anticipated. On shutdown, it is possible that the pressure in the low pressure section will settle out near to its normal inlet pressure. The pressure in the high pressure section may settle out at a pressure between its normal inlet and discharge pressures. The result would be a higher than normal differential pressure across the diaphragm separating the compressor sections. This condition should be clearly specified.

MAIN PROCESS CONNECTION ORIENTATION

The Design Specification should identify the orientation of the compressor main process connections. The specified location will be dependent to a large extent on the type of machine foundation employed, i.e., either on an elevated table or at grade.

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BASIC DESIGN CONSIDERATIONS (Cont)

Axially split compressors may be specified with inlet and discharge connections located in either the top or bottom half of the casing. Connections in the bottom half of the compressor offer the advantage that the process piping need not be disturbed when the casing top half is removed for maintenance. The largest connections in the casing top half normally point straight up and those in the bottom half, straight down. However, horizontally oriented primary and side stream connections are available on many models.

For a radially split multi-stage (barrel) type compressors, the connections can be located anywhere on the circumference of the machine but not in the ends of the machine. Single stage radially split compressors have nozzles oriented as shown in [Figure 11](#).

If a compressor is specified with more than one compressor section in a single casing, it is possible that physical space available may prevent the manufacturer from providing a machine with all main process nozzles oriented in the same direction.

SHAFT SEALS

Main Shaft Seal Types - The compressor main shaft seals are located at either end of the compressor shaft adjacent to the machine bearing housings. [See Figure 15](#). The seal will be balanced to compressor inlet pressure or if a buffer gas supply is used they will be subjected to the supply pressure of the buffer gas. For machines that have the last compression stage located next to the balance piston, the seal at that end of the machine will still be subjected to inlet pressure; however, the temperature at that seal will be close to discharge temperature.

There are many types of seals available, with each manufacturer having their own particular variation of a given seal type. Compressor main shaft seals may be one or a combination of five seal types which are labyrinths, restrictive rings, liquid film oil seals, mechanical (contact) oil seals, and self acting gas seals.

- Labyrinth Seal** - The labyrinth seal is the simplest and the least expensive of all the compressor shaft seal types. Three varieties are used in centrifugal compressors, straight, stepped and interference types. These are illustrated in [Figure 22A, B & C](#). Labyrinth seals utilize a number of shaft or casing mounted teeth with a radial clearance between the teeth and the opposing surface. The teeth act like a series of restriction orifices, with a pressure break down across each tooth. Labyrinth leakage rates are a function of the clearance, differential pressure, and the geometry of the flow path. Stepped and interference labyrinth leakage rates are approximately 50-60% of those for the straight labyrinth design.
- Restrictive Ring Seal** - The restrictive ring seal operates on the same principle as a labyrinth, with the labyrinth teeth being replaced by carbon rings. [See Figure 22D](#). The rings are rub tolerant; however, close clearances are used and replacement due to wear is more frequent than for labyrinth seals. The close clearances employed by the seal also make it susceptible to fouling in dirt laden services. For this reason it should not be used on air compressor duties. Leakage rates are approximately 20% of those for the straight labyrinth seal.
- Liquid Film Oil Seals** - The liquid film oil seal is illustrated in [Figure 23A](#). It contains two bushings arranged back to back. Oil is injected into the cavity between the bushings at a pressure approximately 5 psi (0.35 Bar), above the internal gas pressure. Oil differential pressure is controlled by maintaining an oil level in an overhead tank in which the vapor space at the top of the tank is referenced to the compressor internal gas pressure. There is a small flow of oil through the inner bushing which prevents the gas from flowing outwards. This oil flow exits the seal housing via an oil drain and can either be returned to the compressor seal oil reservoir, or reclaimed, or discarded if contaminated by the process gas. The flow rate is typically 10-15 US gallons per day per seal (35-55 liters/day). The majority of the seal oil flow is across the outer bushing to atmosphere. This oil flow cools the seal and is returned to the seal oil reservoir.
- Mechanical (Contact) Oil Seal** - [See Figure 23B](#). This seal typically contains a pressure breakdown sleeve outboard and a mechanical contact seal on the inboard or process gas side. Seal face contact is maintained by hydraulic pressure and spring forces. Oil is injected at a pressure of 35-50 psi (2.5-3.5 bar) above the internal gas pressure. There is a small flow of oil across the seal faces which prevents the outward flow of gas. This oil flow exits the compressor via an oil drain and can either be returned to the compressor seal oil reservoir, or reclaimed or discarded if contaminated by the process gas. The flow rate is typically 5-10 US gallons (20-35 liters) per day per seal.
- Self Acting Gas Seals (Dry Gas Seals)** - The self acting gas seal does not require any liquid for sealing, lubrication or cooling as it uses the sealed gas to perform these functions. The gas can be either the compressed process gas or an external gas supply. A gas seal consists of a stationary face and a rotating seat etched with a spiral grooved pattern. The rotating grooves raise the pressure of the gas which maintains a small gap between the seat and face of 0.06 to 2 thousandths of an inch (3 to 10 microns). Leakage rates across a single gas seal is very low, typically 2% of an equivalent plain labyrinth seal. The common arrangements for self acting gas seals are as follows:

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1. **Single Gas Seals - See Figure 24** - The single seal consists of one seat and face combination. Filtered gas, typically taken from the compressor discharge line, is injected into a cavity between the seal and seal chamber labyrinth. The majority of the filtered gas flows into the machine across the seal chamber labyrinth, with the remainder leaking across the gas seal to atmosphere.
2. **Double Gas Seal - See Figure 25** - The double seal consists of two seat and face combinations arranged back to back. An external supply of filtered gas is injected between the two seals at a pressure higher than compressor sealing pressure. In most applications, the external gas supply (buffer) is an inert gas such as nitrogen. The filtered gas leaks inboard across the primary seal and outboard across the secondary seal. There is no process gas leakage to atmosphere as only buffer gas leaks across the secondary seal. The secondary seal will also seal against compressor internal gas pressure in the event of a major primary seal failure. A control system is provided to maintain a constant buffer gas to internal process gas differential pressure.

Double gas seals are less expensive than tandem gas seals and have simpler auxiliary systems. The primary application concerns are the availability of a reliable buffer gas source and the impact of buffer gas leakage into the process stream. The buffer gas source must be reliable since the compressor is inoperable without buffer gas for the seals.
3. **Plain Tandem Gas Seal - See Figure 26** - This seal has two seat and face combinations arranged in series. The primary seal is designed to seal against compressor sealing pressure. The secondary seal normally operates with a small differential pressure across it but is also designed for compressor sealing pressure thus providing a backup in case of a primary seal failure. Filtered gas, typically taken from the compressor discharge line, is injected into a cavity between the primary seal and the seal chamber labyrinth. A small amount of the filtered process gas leaks across the primary seal into a cavity between the primary and secondary seals. This cavity is normally connected to a flare or hydrocarbon vent system which is maintained at a pressure that is typically just above atmospheric pressure. A typical secondary seal leakage rate to atmosphere is approximately 0.5 SCFM (14 SI/min.). The secondary seal leakage is normally piped to the atmosphere at a safe location. This seal arrangement will not contaminate a process stream with an external gas supply; however, there is some leakage of the process gas to the atmosphere.
4. **Tandem Gas Seal With Intermediate Labyrinth - See Figure 27** - This arrangement is a variation of the plain tandem design, having an intermediate labyrinth installed in the cavity between the primary and secondary seals. Filtered gas, typically taken from the compressor discharge line, is injected into a cavity between the primary seal and the seal chamber labyrinth. A small amount of the filtered gas leaks across the primary seal into a cavity between the primary seal and an intermediate labyrinth. This cavity is vented to the plant flare or hydrocarbon vent system which typically operates at pressures close to atmospheric. An external supply of filtered buffer gas is injected into the cavity between the intermediate labyrinth and the secondary seal at a pressure that is higher than the flare or vent system pressure. Buffer gas leaks across the intermediate labyrinth and exits to the vent or flare system together with the process gas that has leaked across the primary seal. The leakage across the secondary seal to atmosphere is just buffer gas, which, if nitrogen is used, makes this type of gas seal a zero emissions device. The primary seal is designed to operate with compressor sealing pressure. The secondary seal normally operates with a small differential across it but is also designed for compressor sealing pressure to provide a backup in case of primary seal failure.
5. **Triple Seal Arrangements** - For very high sealing pressures, three self acting dry gas seals can be arranged in series with the first two seals sharing the pressure reduction duty. While some experience does exist with these arrangements the use of a triple seal should not be specified without consultation with a machinery specialist.

Shaft Seal Type Selection - API-617 requires that the Purchaser specify the type of shaft sealing device that should be used. The following recommendations assume that specialists in the areas of industrial hygiene, air quality, and safety will review the service and provide inputs necessary to determine the adequacy of the proposed arrangements. See [Design Practice XVIII Section B, Guidelines and Considerations For Industrial Hygiene](#). Additional factors affecting seal selection and design are, the maximum static and dynamic sealing pressures, the settling out pressure, sealing temperatures and other process considerations. See [Table 1](#) for a summary of the following guidelines.

- **Labyrinth and Restrictive Ring Seals** - Labyrinths are recommended for all air services. Labyrinth and restrictive ring seals may be used only for nitrogen or other inert gas services. For high sealing pressures and valuable gases, it may be possible to justify a single self acting dry gas seal. These recommendations assume that the gas stream is **Respiratory Class R-4** and no fugitive emission limits apply. Typically the maximum temperatures and pressures to which these seal designs should be exposed are 350°F (175°C) and 150 psig (10 barg). For other applications machinery specialists should be consulted.

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- **Liquid Film and Mechanical (Contact) Oil Seals** - Both of these seal types use oil injection to minimize leakage of process gas to the atmosphere and are suitable for sealing gas streams with **Respiratory Classes R-1 through R-4**. For services with sealing pressures up to 650 psig (45 barg) and temperatures to 350°F (175°C), both seal types are applicable and should be specified. The compressor vendor can then choose which to offer based on operating experience. For higher sealing pressures a liquid film oil seal should be specified. Liquid film oil seals have been applied at pressure of 5000 psig (345 barg).
- **Self Acting Gas Seals** - Gas seals may be used for maximum dynamic sealing pressures up to 3000 psig (210 barg), maximum static sealing pressures to 3750 psig (260 barg) and temperatures up to 500°F (260°C). This is the current limit of operating experience for dual seal arrangements. Gas Seals should only be specified above these pressures after consultation with a machinery specialist for verification of experience at contract conditions. If available, gas seals should be used on all new equipment for safety, reliability, contamination and maintenance reasons.
 - For services with sealing pressures of 300 psig (20 barg) to 3000 psig (210 barg) and gases with **Respiratory Classes R-1 through R-4**, specify double gas seals. Be careful to ensure that a reliable buffer gas source is available before specifying the double gas seal.
 - For services with sealing pressures of 360 psig (25 barg) to 3000 psig (210 barg) and gases with **Respiratory Classes R-2 through R-4**, specify plain tandem seals.
 - For services with sealing pressures 435 psig (30 barg) to 3000 psig (210 barg) and gases with **Respiratory Classes R-1 through R-4**, specify tandem gas seals with an Intermediate labyrinth.
 - For services outside the above pressure ranges, consult with a machinery specialist to determine if experience is available at the contract conditions.
- **Gas Seal Standards** - Currently the API and Global Practices do not cover all of the requirements for the specification of gas seals and their auxiliaries. However, **GP 10-03-01** and **API 617** has basic requirements for the gas seal. **GP 10-14-01** and **API 614** discuss specifications for the seal support systems. **GP 15-01-02** specifies auxiliary system schematic and instrumentation requirements. When specifying a gas seal, the designer should consult with a machinery specialist to ensure that the job specification includes all of the required information.

Gas Seal Design Sensitivities

- **Gas Filtration** - Self acting dry gas seals need to be supplied with a stream of dry, filtered gas at the necessary pressure and temperature for desired seal performance. For tandem or single seal designs, this is typically achieved by taking a tapping from the compressor discharge line, prior to any cooler, filtering the gas to 5 micron or better and then controlling the gas pressure relative to compressor sealing pressure. A high level of filtration is necessary because of small operating gaps between the seal faces. Although larger than the operating gap for some applications, a 5 micron level of filtration has generally proven to be sufficient for continued trouble free operation. Any gas stream required to pass across any pair of seal faces must be filtered to this level. This includes buffer gas supplied to either double or tandem seals with intermediate labyrinths.
- **Liquids in the Gas Stream** - Seal performance can be degraded by liquids in the gas stream. They have a tendency to leave deposits on the seal faces or cause the accumulation of dirt particles in grooves provided on the rotating seat. Liquid can enter the primary seal faces from the filtered process gas or the secondary seal faces from lube oil migration along the shaft. Liquids in the filtered process gas stream are typically eliminated by using hot compressor discharge gas and by taking tappings from the top of the discharge line rather than side or bottom. Every effort should be made to eliminate the potential for liquid in the process gas supply by using discharge gas from the final compressor section or by using an external gas supply. If concerns still remain regarding the quality of the available process gas and double seal arrangements are not practical, then coalescers and/or gas heaters should be considered.

To avoid contamination of the secondary seal from lube oil migration, various options are available. They include the use of mechanical devices to limit or restrict oil flow, eductor systems and the use of separation gas injected into a restriction seal (See Figure 26). The latter arrangement eliminates oil flow by providing a positive flow of gas along the shaft in the direction of the bearing housing. Nitrogen is the preferred medium for separation gas for this application, however, the control method employed to regulate the supply of this utility should be sufficiently sophisticated to keep the consumption to a minimum.

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- **Reverse Rotation** - Grooves are etched into the rotating seat of a gas seal. They create a gas pressure between the seal faces in operation; however, they are typically uni-directional and do not provide any pressure profile to separate the seal faces if spun in the reverse direction. This means that if the compressor spins in the reverse direction the seal faces will rub and very quickly fail. It is therefore important to ensure that compressor system design eliminates or minimizes the potential for reverse rotation. Bi-directional seal designs have been developed, but have little operating experience. These designs results in a reduced and less stable operating gap. Bi-directional seals should not be specified without consultation with machinery specialists.
- **Reverse Pressurization** - Gas seals are designed for a pressure differential in one direction. If they are subjected to reverse pressure, the seat and face will touch and cause sudden seal failure. Compressors that have low internal gas pressures to seal are susceptible to reverse pressurization when vent pressure, under upset conditions, exceeds sealing pressure, or if the inlet pressure to the compressor goes sub-atmospheric. The Design Specification must state both normal and maximum operating pressures for the hydrocarbon vent or flare system to which primary seal leakage is routed. If it is possible for sealing pressure to fall below the pressure of the plant vent or flare system, then a note must be added to the Design Specification. This note should require that the sealing chambers be balanced to a pressure higher than compressor suction pressure, e.g., second stage inlet pressure.

Buffer Gas Requirements - The foregoing discussions covered buffer gas requirements for various types of gas seals. Liquid film and mechanical (contact) oil seals may also require buffer gas in special situations. For oil seals, buffer gas is injected into the seal chamber labyrinths located at either end of the machine. See [Figure 15](#). The buffer gas supply is controlled to have a higher pressure than the sealing pressure thus resulting in a flow of buffer gas across the labyrinth into the machine. The buffer gas also flows into the inner seal chamber between the main shaft seal and the seal chamber labyrinth. The seal oil leakage into the inner seal chamber mixes with the buffer gas rather than the process gas thus reducing the possibility of oil contamination by the process gas. Oil seal applications where buffer gas should be provided are as follows:

1. Use in low temperature services to maintain the temperature of the oil in the inner seal chamber above its pour point. This is normally accomplished by taking a warm stream of compressed gas off the operating compressor and an external source of buffer gas is not required.
2. In high temperature service, use a buffer gas to cool the inner seal to prevent seal overheating and oil degradation.
3. Use buffer gas to maintain a positive internal gas pressure at the seals under all operating conditions including negative suction pressure. This buffer gas is normally compressed gas from the operating compressor and no external source of supply is required. However, for compressor operation on air an external nitrogen buffer gas supply is required to avoid the mixing of air and oil in the inner seal chambers.
4. Use buffer gas to prevent compressed gases containing corrosive agents from contacting the oil in the seal and contaminating both the lube and seal oil of compressor trains with combined lube and seal oil systems. This could result in corrosion of oil system components and damage to bearings and seals. See **Combined Lube and Seal Oil Systems** discussion this Section.

Centrifugal compressors with oil shaft seals should never be operated in a closed loop using air or oxygen, unless special safety features such as those described below have been provided. If this is not done, seal oil will leak into the recirculating gas stream and could generate an explosive concentration within the closed loop. See also [Design Practice XV Section B, Minimizing the Risk of Fire, Explosion or Accident](#). Applications of this type must be reviewed with the machinery specialist.

An example is the use of a centrifugal POWERFORMING recycle gas compressor equipped with oil type shaft seals to circulate air for drying out the piping system or for catalyst regeneration. To guard against entry of the oil into the circulating air stream, the compressor should be specified to have nitrogen injected at the inner seal chamber labyrinth. The nitrogen acts as an inert gas barrier between the hot air at the compressor discharge and the seal oil at the inner oil drain chambers, and it provides an inert atmosphere at the inner seal. The buffer also serves as a barrier to keep oil from entering the process piping. The buffer gas system must be instrumented for alarm at low differential pressure, to signal failure of the protection system.

Buffer gas consumption rates will vary with machine size, seal design and service conditions. For an estimate of leakage rates see [Design Practice XI Section P, Compressor Unit Auxiliary and Installation Facilities](#).

API-617 requires that seals be provided with the facility for buffer gas injection but does not stipulate that buffer gas must be used. Buffer gas may be an inert gas such as nitrogen or a sweet hydrocarbon gas taken from a suitable location within the process unit. The Design Specification must give details of buffer gas composition, pressure, and temperature. **GP 10-03-01** requires the compressor manufacturer to supply the buffer gas manifold together with the necessary valves, pressure indicators, flow meters, check valves, throttle valves, differential pressure indicators and alarms, controllers and control valves.

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LUBE OIL, SEAL OIL AND GAS SEAL SYSTEMS

General - The function of these systems is primarily to provide an uninterrupted supply of clean and cooled fluid to the bearings and seals of the compressor, gear and driver. Compressor train reliability depends on the continued operation of these systems. To attain long continuous runs, the key components of these systems are spared so that maintenance can be performed with the compressor train in operation. Oil system pumps are spared with automatic start up of the spare pump provided. Coolers and filters are spared, and are provided with cross-over valving designed to prevent flow interruption during the switching operation. In addition to the use of multiple key components, the systems are equipped with extensive instrumentation which raise alarms and provide shutdown and permissive start functions.

Combined Versus Separate Lube and Seal Oil Systems - For compressors using oil seals, combining lube and seal oil requirements and providing one common system to meet the needs of both bearings and seals offers significant cost savings. These systems can be combined if the process gas stream cannot permanently degrade the oil or if a suitable clean buffer gas supply can be found and injected into the machine sealing area thus preventing process gas and oil contact and hence contamination. Gas streams with a H₂S content less than 100 ppm by volume are suitable for combined oil systems, for higher H₂S content a suitable source of sweet buffer gas must be found or separate systems become necessary. The Design Specification must state whether the lube and seal oil system is to be separate or combined.

Common Oil Systems for Multiple Machine Trains - Combining the oil requirements of several compressor trains into a single oil system can also provide significant savings. The combined system must be designed with sufficient redundancy to avoid any impact on individual machine train availability and should also permit maintenance on one machine train while others remain in operation. The machine trains need to be physically located in close proximity so that oil drain lines returning to the reservoir will have the necessary slope. Drain line runs in excess of 80 feet (25 m) should be avoided. The Design Specification should state when an oil system is intended to serve more than one compressor train.

Oil System Pump Drivers - **GP 10-14-01** requires that the driver for the main oil pump be electric motor driven and the standby steam turbine driven. If another arrangement is required, the Design Specification must so specify. For applications where both pump drivers are electric motors, power supplies should be provided from independent sources.

Gas Seal Systems - The systems provided for the self acting gas seals are smaller and less complex than a seal oil system. Gas seal systems are normally mounted on or adjacent to the compressor base plate and consist primarily of filters and controls. For the gas seal arrangements that use nitrogen as a separation gas between bearing housing and seal assembly and for the tandem seal with intermediate labyrinth, the cost of nitrogen consumption can be high. This can be remedied by the use of instrumentation which will precisely control nitrogen differential pressure to fractions of a psi. As previously noted, Global Practices and API standards contain only the basic requirements for dry gas seal systems; therefore, the Design Specification must be supplemented with the appropriate notes if a gas seal is specified. **GP 15-01-02** specifies auxiliary system schematic and instrumentation requirements. Additional requirements should be developed in conjunction with a machinery specialist.

CONTROLS AND INSTRUMENTATION

General - The Design Specification should provide a description of the compressor process control and surge protection scheme. It should identify the controlled variable, the source of the control signal, and the control method, e.g., speed control, inlet throttling etc. Fail safe modes for control elements should be shown.

The controls and instrumentation associated with a centrifugal compressor train can be grouped into one of three categories. The first group includes all the controls and instrumentation associated with the machine auxiliary and mechanical condition monitoring systems, e.g., vibration and temperature measurement. This type of instrumentation is discussed further in [Design Practice XI Section P, Compressor Unit Auxiliary and Installation Facilities](#), with scope and requirements defined by the Global Practices. The second group is instrumentation for compressor performance control. The final group includes the controls and instrumentation that prevent compressor surge.

Interaction of System Characteristics with Compressor Performance Curves - It is important that the interaction of the compressor performance and system resistance characteristic be appreciated prior to deciding on a control method. This can be done in the design phase by estimating the system discharge pressure requirements over a flow range of 0 to 130% of Normal flow and obtaining typical compressor performance curves generated by either the PC Program for Compressor Network Simulation - CompCalc, or by using the techniques previously discussed for determining curve slope and stability range.

Figure 28 shows a performance curve for a fixed speed compressor with three different system characteristic curves overplotted. A machine operating against a fixed pressure would have a system characteristic defined by the line AB, e.g., an Alkylation plant refrigerant compressor. A compressor discharging into a long run of pipe would have a characteristic that