Selecting a Centrifugal Compressor

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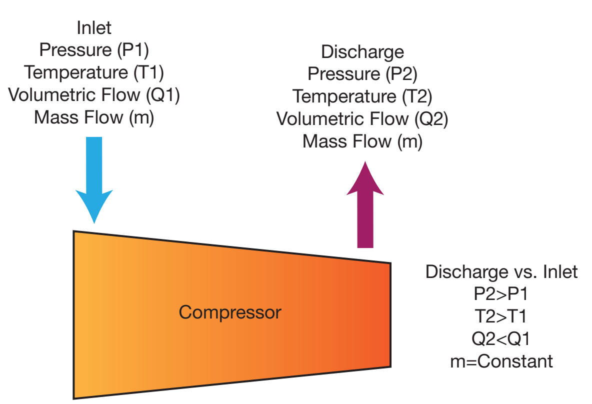
End users must specify certain performance requirements when requesting a quote for a new centrifugal compressor. Understand your process, as well as the advantages and disadvantages of each centrifugal compressor configuration, in order to choose the optimal centrifugal compressor for your application.

Centrifugal compressors, also called radial compressors, are critical equipment in a wide variety of applications in the chemical process industries (CPI). As their name suggests, their primary purpose is to compress a fluid (a gas or gas/liquid mixture) into a smaller volume while simultaneously increasing the pressure and temperature of the fluid. In other words, compressors accept a mass of gas at some initial pressure and temperature and raise that gas to a higher pressure and temperature (Figure 1). At the higher discharge pressure and temperature, the gas density is also higher, so the mass of gas occupies a smaller volume — *i.e.,* the gas is compressed.

Of the numerous technologies that can achieve compression, this article focuses on centrifugal compressors. It explores the various types of centrifugal compressors, provides valuable information on impellers, and explains basic centrifugal compressor sizing. (Reference 1 provides information on other types of compressors, such as positive-displacement, axial, and others.)



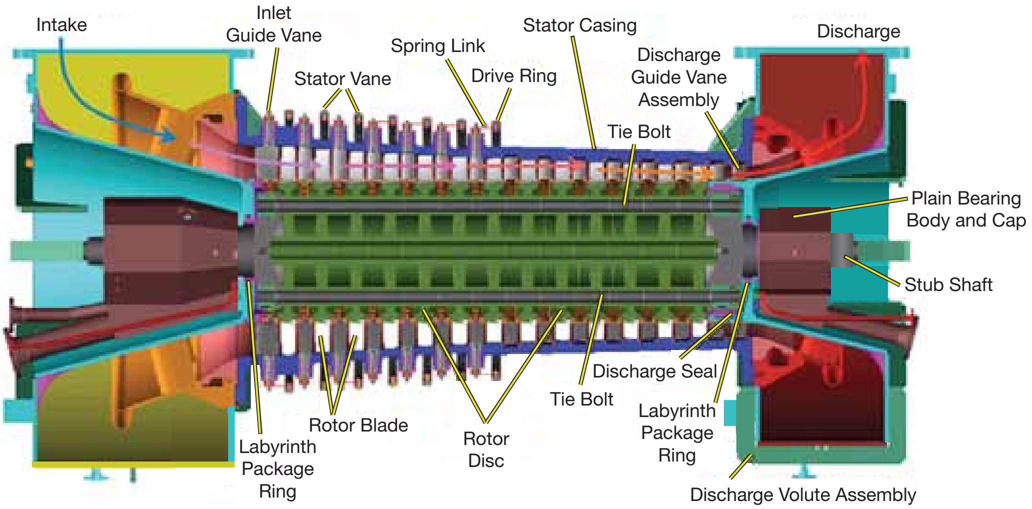
▲ **Figure 1.** Compressors accept a mass of fluid at an initial pressure and temperature, and raise it to a higher pressure and temperature, thereby compressing its volume.

Turbocompressors

Centrifugal compressors are members of a class of technologies called dynamic compressors, or turbocompressors. Axial compressors are also part of this class of turbomachines. Axial and centrifugal compressors draw their names from the primary direction in which the flow moves within the compressor. Axial compressors (Figure 2) handle much higher flowrates than centrifugal compressors, but generate lower pressure ratios. Modern centrifugal compressors accommodate lower flowrates than axial compressors but are capable of generating much higher pressure ratios.

In turbocompressors, the increase in pressure and reduction in volume is accomplished by adding kinetic energy to the fluid stream (*i.e.,* adding velocity pressure) and then converting that kinetic energy into potential energy in the form of static pressure. In centrifugal compressors, kinetic energy is added by impellers. The number of impellers in a compressor depends on how large a compression or pressure increase is needed for the process. As a result, compressors can have one or as many as 10 (or more) impellers.

The conversion of the velocity pressure to static pressure occurs in downstream stationary components, such as diffusers, return channels, and/or volutes. The type of stationary component(s) in compressors depends on the style of centrifugal compressor being considered. The role of each of these components is discussed in this article.



▶ **Figure 2.** In an axial compressor, flow moves in an axial direction (from left to right in this diagram), rather than in a circular direction as in centrifugal compressors.

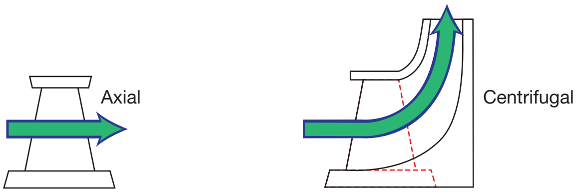
A simple analogy

To help understand the concepts of velocity and dynamic pressure, think about a fan that you might have in your home or office. If you place your hand in front of the fan, you can feel the kinetic energy that the fan blades have added to the air. If you place your hand behind the fan, you can feel movement of the air as it is being drawn into the fan. The suction is caused by a reduction in static pressure due to the acceleration of the air by the fan blades, thereby drawing more air into the fan.

Now imagine that you arrange several fans in a row inside an enclosure to ensure that all of the flow goes in one direction. Imagine how much force you would feel coming out of the last fan in the stack after each fan accelerates the air (*i.e.,* adds more kinetic energy). That is the basic concept behind a compressor — a series of rotating blades adding energy to the gas.

Now suppose that the flow changes direction as it passes through the rotating blades so that it exits the blades traveling radially outward rather than in an axial direction (Figure 3). That is the fundamental difference between the axial compressor's rotor and the centrifugal (or radial) compressor's impeller — the axial rotor discharges flow in the axial direction while the centrifugal impeller discharges in a radial direction.

The impeller adds kinetic energy to the fluid in the same way the blades of a household fan do, although the centrifugal impeller adds more energy to the fluid than can be added with a typical fan blade. Thus, it is possible to achieve much higher pressures with centrifugal impellers. Multistage centrifugal compressors have multiple impellers stacked together and connected by flow passages.



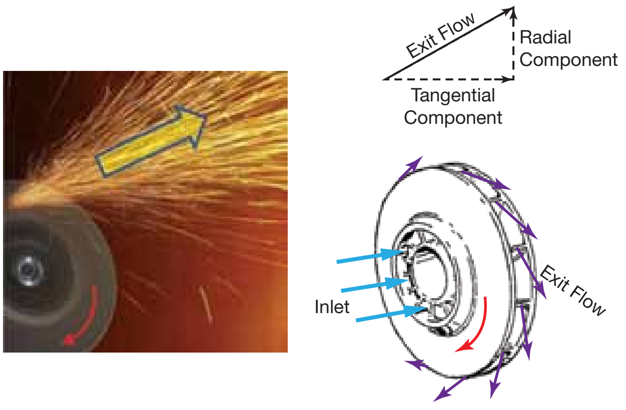
▲ **Figure 3.** Flow exits an axial rotor (left) in an axial direction, while flow from a centrifugal impeller (right) exits radially.

Although centrifugal compressors are sometimes called radial compressors, most of the flow exiting a centrifugal impeller does not travel in a radial direction. Rather, the flow travels to a large extent in a tangential direction.

This motion is characteristic of a rotating disk. Consider the direction that wood dust travels when you are using a disk sander, or that sparks fly when you are using a grinding wheel (Figure 4). Similarly, the fluid that passes through a centrifugal impeller is flung out along a path that has both a radial velocity component and a tangential velocity component.

Impellers — the heart of the centrifugal compressor

The most critical components in any centrifugal compressor, regardless of style, are the impellers. If the impellers do not provide a high efficiency and good overall flow range, it is impossible for the compressor to achieve a



▲ **Figure 4.** Flow exits a centrifugal impeller in the direction of rotation, just as sparks fly from a grinding wheel. The red arrows indicate the direction of rotation.

high efficiency and good flow range.

Impellers are the only rotating aerodynamic components in a centrifugal compressor. They provide 100% of the kinetic energy that is added to the gas and can be responsible for up to 70% of the static pressure rise in a stage.

They are also the most efficient component in a stage. Well-designed impellers can achieve efficiencies in excess of 96%, that is, only 4% of the energy expended is lost. The losses in the stationary hardware reduce the overall stage efficiency from the baseline established by the impeller. Therefore, if the performance of the impellers is poor, the overall compressor performance can only be worse.

Centrifugal compressor impellers can be categorized as shrouded or unshrouded (Figure 5), and their blades as two-dimensional or three-dimensional (Figure 6). The type of impeller chosen for a particular application depends on many considerations, such as required operating speed, desired pressure ratio, desired efficiency, and equipment cost.

The absence of a cover allows unshrouded impellers to operate at higher rotational, or tip, speeds. The pressure ratio generated by an impeller is proportional to the square of the operating speed. Therefore, open (unshrouded) impellers are capable of generating much higher pressure ratios than shrouded impellers. Most shrouded impellers generate pressure ratios of 3:1 or less, whereas unshrouded impellers can reach pressure ratios of 10:1 or higher.

However, unshrouded impellers tend to be less efficient because of the high losses associated with the tip leakage flow (*i.e.,* the flow that leaks over the rotating blades). Tip leakage does not occur in a covered impeller.

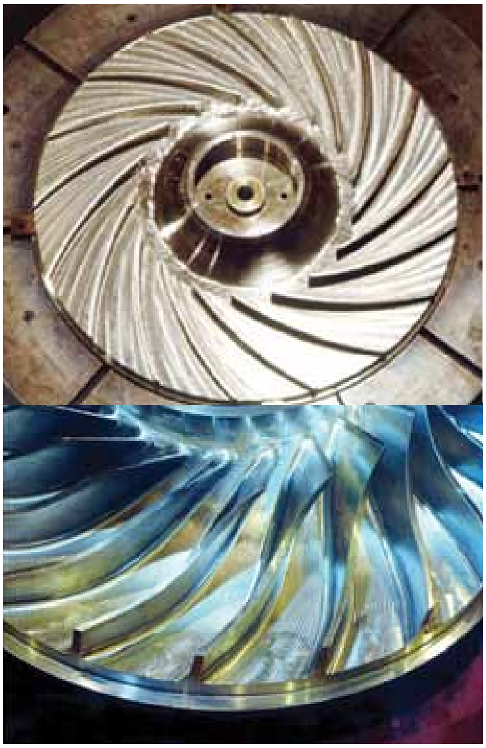
The selection of blade style depends on many factors; from an aerodynamic perspective, the most important is the impeller flow coefficient. The flow coefficient, φ, relates an impeller's volumetric flow capacity, Q, operating speed, N, and exit diameter, D_2:

$$\varphi = \frac{Q}{N \times D_2^3} \quad (1)$$

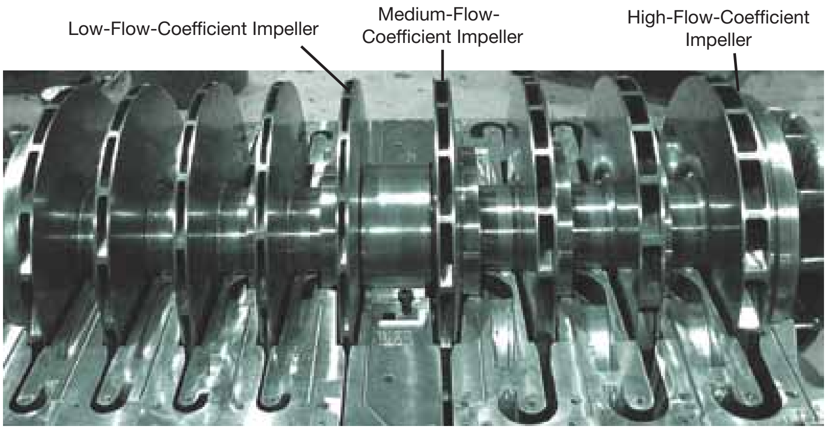
Low-flow-coefficient impellers are characterized by long, narrow passages, while high-flow-coefficient impellers have much wider passages to accommodate the higher flowrates.

The compressor rotor shown in Figure 7 contains impellers with a wide range of flow coefficients. The impeller with the highest flow coefficient is located at the right end of the rotor. The remaining impellers are progressively narrower, with increasing fluid pressure and decreasing volumetric flowrate. The impeller with the lowest flow coefficient (at the left, closest to the center of the machine) is much nar-

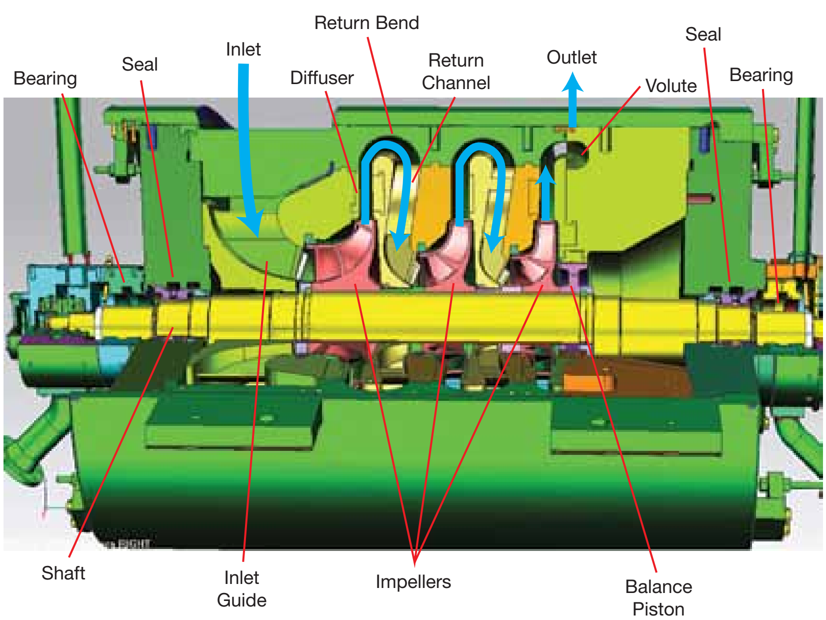
▲ **Figure 5.** Impellers can be either shrouded (top) or open (bottom).



▲ **Figure 6.** Impeller blades can be either two-dimensional (top) or three-dimensional (bottom). The 2D blades in the top image have a circular arc shape, whereas the blades in the lower image have a complex 3D shape.



▲ **Figure 7.** This compressor rotor has impellers with a wide range of flow coefficients, from low at the left to high at the right.



▲ **Figure 8.** A cross-sectional view of a three-stage centrifugal compressor with a between-bearing design.

rower than the high-flow-coefficient impeller on the right.

Low-flow-coefficient impellers typically have simpler blades, such as the 2D blades on the top of Figure 6, which are defined by circular arc sections. Higher-flow-coefficient impellers typically have complex, 3D blades (Figure 6, bottom) that cannot be defined by any simple geometric shape. More complex blade shapes are defined with sophisticated computer programs that determine the blade contours necessary to ensure optimal aerodynamics.

Due to their narrow passages and simpler blades, low-flow-coefficient impellers deliver lower efficiency than high-flow-coefficient impellers. However, applications may require the use of impellers with a low flow coefficient when the process flowrate is small relative to the operating speed or impeller diameter (see Eq. 1), or when upstream stages or compressors reduce the flowrate, and require low-flow-coefficient stages to complete the compression process.

Centrifugal compressor configurations

Flow exits an impeller in both a radial direction and a tangential direction, and is often described as swirling outward. The control, or guidance, of this swirling flow is one of the primary purposes of a centrifugal compressor's stationary components. The other purpose is to efficiently convert the dynamic pressure exiting the impeller into static pressure. The specific stationary components depend on the style of centrifugal compressor in use.

Multistage centrifugal compressors generally fall into two categories: between-bearing designs, and integrally geared designs.

Between-bearing configurations

In the between-bearing design, the impellers are mounted on a single shaft. Figure 8 shows a cross-section of a three-stage between-bearing compressor. A driver (*e.g.,* an electric motor, steam turbine, or gas turbine) rotates the shaft and all of the impellers at the same speed. Flow enters the compressor via the inlet and flows into the inlet guide, which distributes the flow circumferentially around the machine to provide a uniform velocity and pressure at the entrance of the first-stage impeller.

As described earlier, the rotating impeller adds kinetic energy or velocity pressure to the gas and the flow exits the impeller with tangential velocity (or exit swirl). The flow then swirls outward though the diffuser along a spiral path (Figure 9). As the flow moves outward in the diffuser, it encounters a larger area (due to the increasing radius) and the flow velocity decreases. This decrease in velocity results in an increase in static pressure (*i.e.,* the velocity pressure is converted to static pressure). At the exit of the diffuser, the flow passes through the return bend, which redirects the flow from spiraling radially outward to spiraling radially inward. Next, the flow passes through the return channel, which has vanes that capture the swirling flow and reorient it radially inward toward the center of the compressor. This process removes any remaining tangential velocity, preparing the flow for the next stage of compression. The flow then enters the next inlet guide and impeller. This cycle is repeated in every impeller stage until the desired discharge pressure and reduction in volumetric flow is achieved. Finally, the gas stream exiting the final diffuser is captured by a volute (or collector), which captures the flow around the circumference of the compressor and guides it into the discharge piping.

Arrangements. Between-bearing compressors come in a wide variety of configurations that fall into two categories: straight-through and back-to-back.

In the straight-through arrangement, the flow enters one end of the compressor and exits the opposite end. The compressor shown in Figure 8 has a straight-through arrangement.

In the back-to-back arrangement, the impellers face in opposite directions, as shown in Figure 7. In this design, the main inlet is at the right end of the rotor and the impellers guide the flow toward the center of the machine. After passing through four impellers (four stages of compression), the flow is piped to the secondary inlet at the left end of the compressor. The remaining five impellers complete the compression process and the flow exits at the center of the compressor. This configuration reduces the pressure on the shaft end seals.

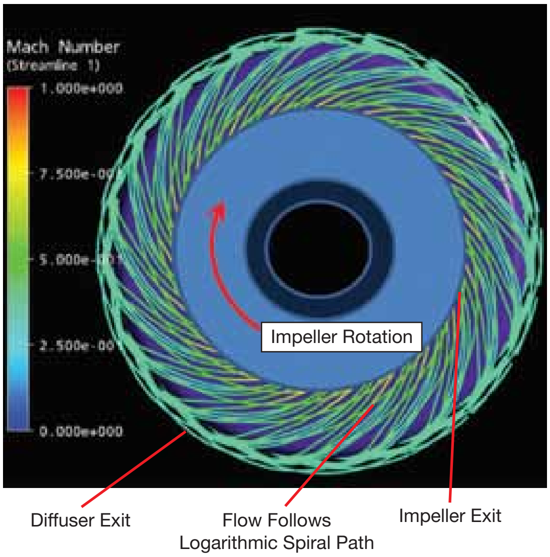
Both the straight-through and back-to-back arrangements can be configured to allow intercooling throughout the compression process. Intercooling is often necessary to keep the temperatures of the compressor material at acceptable levels to maintain their strength. Intercooling also reduces the power required to complete the necessary compression.

Compressors with a high discharge pressure tend to use a back-to-back arrangement. Each original equipment manufacturer (OEM) has different design rules that dictate when to use the back-to-back arrangement, however, it is commonplace for compressors with discharge pressures above 2,000 psi.

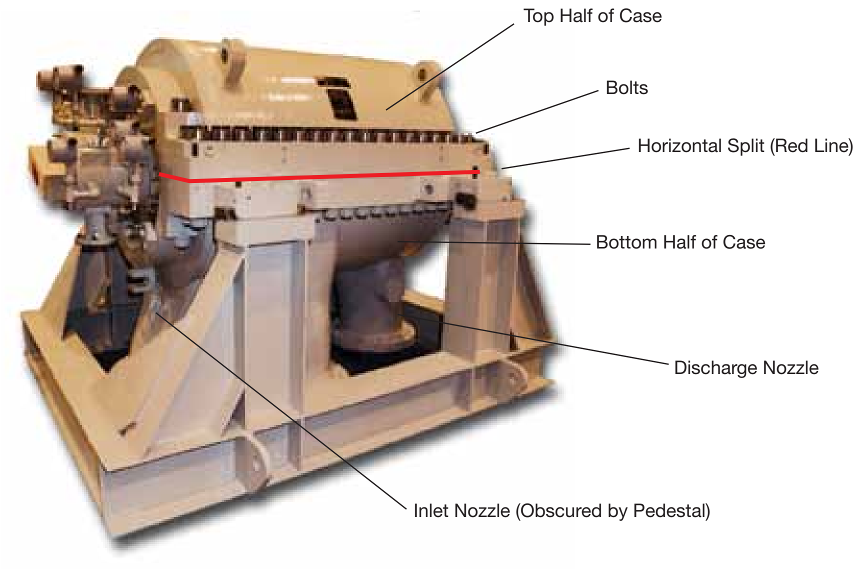
Casings. Between-bearing compressors are available with one of two basic case types: horizontally/axially split or radially split. In the horizontally/axially split compressor, the casing is comprised of two halves with the horizontal joint bolted together (Figure 10). These cases are typically limited to lower-pressure applications (*i.e.,* discharge pressures less than 1,200 psia).

Radially split compressors are typically referred to as barrel compressors (Figure 11). The increased strength provided by the cylinder of the barrel casing allows barrel compressors to operate at much higher pressures than horizontally split compressors. In fact, barrel compressors that exceed 12,000-psia discharge pressure are available. For these extremely high-pressure applications, elaborate sealing systems are needed to prevent leakages.

The high rotational speed and length of centrifugal compressor rotors make robust support or bearing systems that limit vibrations to acceptable levels necessary. Therefore, between-bearing



▲ **Figure 9.** In this computational fluid dynamics (CFD) depiction, as the flow swirls outward from the impeller, the flow velocity decreases.



▲ **Figure 10.** A horizontally/axially split compressor is held together with bolts. The horizontal joint is highlighted in red.

configurations typically require two radial bearings and one thrust bearing to support the shaft and to compensate for changes in axial thrust as the compressor operates at different flow conditions. (References 1 and 2 provide further details on these and other commonly used bearings.)

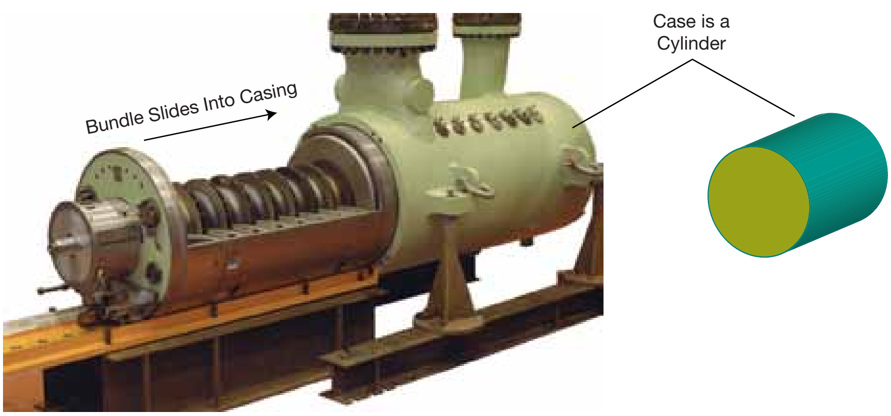
Another critical component, especially in high-pressure compressors in hydrocarbon or toxic gas applications, is the shaft end seal, which keeps the process gases from leaking to the atmosphere. Gas seals have gained wide acceptance in the turbomachinery community and are the seal of choice for most applications. Significant advances have been made in recent years on these seals. (References 1 and 3 provide more information on the design and advancement of gas seals.)

Integrally geared configurations

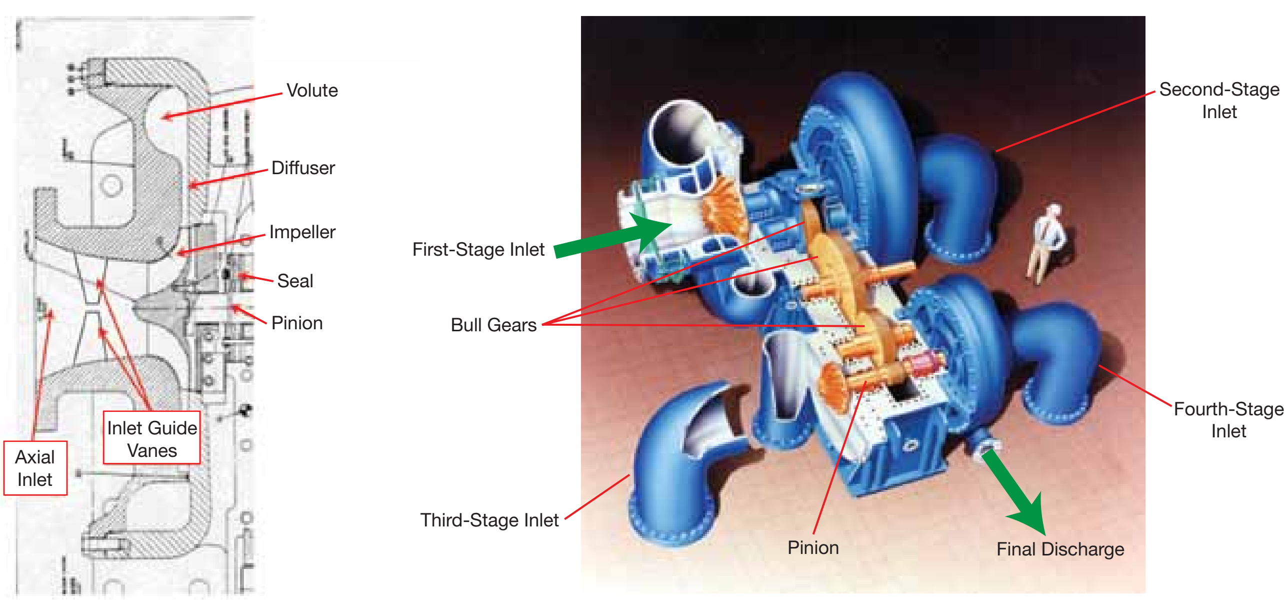
In an integrally geared compressor, the impellers are mounted at the ends of multiple pinions that can rotate at different speeds depending on the gear ratio between the individual pinions and the bull gear(s). The number of impellers and number of pinions vary depending on the application. Typical integrally geared compressors have two to eight (or more) pinions with two impellers mounted at the opposite ends of each pinion. This configuration is simpler aerodynamically, but more complicated rotordynamically and mechanically, than the between-bearing configuration.

Flow enters the first stage of an integrally geared compressor (Figure 12) via an axial inlet or straight run of pipe. Depending on the design, the flow might pass through an inlet guide before entering the impeller. The impeller adds kinetic energy to the gas stream. Flow exiting the impeller enters a diffuser, which converts a portion of the velocity pressure to static pressure. At this point, the flow enters a discharge volute (collector). The flow from the volute is then piped to the axial inlet of the next stage, which eliminates the need for the return bend and the return channel used in the between-bearing design.

Integrally geared compressors have several advantages over between-bearing configurations. The



▲ **Figure 11.** In a radially split compressor, the case is a cylinder and a compressor bundle slides into the casing.



▲ **Figure 12.** An integrally geared compressor has multiple pinions driven by bull gears. An impeller is mounted on each end of each pinion.

flow in the inlet section of the between-bearing design must be distributed around the circumference of the compressor, which requires extra turning and creates pressure losses. The axial inlet of an integrally geared compressor, on the other hand, requires a straight run of pipe and therefore has lower aerodynamic losses than the inlet of the between-bearing design. Additionally, because the impellers can be mounted to different pinions in an integrally geared compressor, it is possible to further tune the performance of a stage by varying the impeller's speed or choosing an impeller with a different diameter altogether. The elimination of the return bend and return channel in an integrally geared compressor reduces the losses in each stage, although the volute or collector losses are only slightly lower.

However, the aspects of the integrally geared design that give it an aerodynamic advantage are also the root of its rotordynamic and mechanical disadvantages. The integrally geared compressor contains a large number of bearings and seals — a minimum of two bearings and two seals for each pinion and additional bearings for the bull gears. Vibration related to gear meshing in integrally geared units is sometimes a problem. Because the impellers are located outside the bearing supports, they can wobble and vibrate if not mounted or balanced properly.

Between-bearing and integrally geared compressors both have advantages and disadvantages, and the choice between the two styles often depends on the particular application.

Specifying compressor performance requirements

An article scheduled for the July 2013 issue of *CEP* will address compressor performance testing and will include a detailed discussion of compressor performance parameters. This article focuses only on those parameters that end users must specify to an OEM when requesting a quote for a new compressor.

At a minimum, the user must specify the flow range that the compressor must handle. This can be specified as a range of mass flowrates (typically lb/min or kg/min) or volumetric flowrates (ft^3/min or m^3/min). Next, the user must specify the composition of the gas to be compressed as well as the range of pressures and temperatures of the gas as it enters the compressor for each operating condition. The OEM will use a real-gas equation (as agreed upon with the user) to determine the gas properties for the mixture. Finally, the user must specify the pressure ratio that the compressor must achieve. Alternatively, the user might specify the discharge pressure that must be achieved and then work with the OEM to determine an acceptable range of inlet pressures.

In many situations, the end user will also specify a driver. This driver needs to be able to deliver a certain amount of horsepower and operate over a specific speed range. Therefore, the OEM must size the compressor to operate within the driver's speed range while not exceeding the horsepower capability of the driver. Alternatively, the user might ask the OEM to select the driver as well as the compressor and then make the purchasing decision based on the overall compression system cost.

End users often must choose among compressors with impellers that have a range of head coefficients and pressure ratios. Head ($Head_p$) is the measure of the amount of energy required to elevate a fixed amount of gas from one pressure level to a higher pressure level:

$$Head_p = \frac{k}{k-1} zRT_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \qquad (2)$$

where k is the ratio of specific heats (C_P/C_V); C_P is the specific heat at constant pressure; C_V is the specific heat at constant volume; z is the compressibility of the gas; R is the gas constant (1,545/mole-weight) in (ft-lb$_f$)/(lb-mol)(°R); T_1 is the inlet temperature in °R; P_1 is the inlet pressure in psia; and P_2 is the discharge pressure in psia.

The head coefficient, μ_p, relates the head increase to the operating speed, N, and impeller exit diameter, D_2. The head coefficient can be determined by:

$$\mu_p = \frac{Head_p g_c}{U_2^2} \qquad (3)$$

where g_c is the gravitational constant; U_2 is the peripheral velocity of the impeller trailing edge ($N\pi D_2/720$) in ft/s; D_2 is the impeller blade exit diameter in in.; and N is the rotational speed in rotations per min.

In general, impellers that generate a high head or high pressure ratio have a narrower flow range than impellers that generate a lower head or lower pressure ratio. High-head-coefficient impellers also provide lower rise-to-surge than lower-head-coefficient designs. Rise-to-surge is a measure of how much the pressure increases between the design flowrate and the flowrate at which surge will occur. Compressor surge is a complete breakdown in compression that occurs when a compressor is run either at a much lower flowrate than intended or at a much higher discharge pressure than intended. End users often specify a minimum rise-to-surge value when purchasing compressors.

Figure 13 is a simplified compressor performance map showing the head coefficient curves for high- and low-head-coefficient impellers. The black arrows immediately below each line indicate the relative values of rise-to-surge of the two impeller styles. The low-head-coefficient impeller has a steeper rise-to-surge slope than the high-head-coefficient impeller. Therefore, for a given change in flowrate, there will be a greater change in operating pressure for the low-head-coefficient design than for the high-head-coefficient design.

Surge is a violent phenomenon, and it can cause extensive damage to compressor components. Therefore, sophisticated control systems are put in place to keep the compressor from operating in surge. Many surge control systems monitor the compressor's discharge pressure to determine where the compressor is operating. Because the low-head-coefficient impeller has a steeper rise-to-surge slope, the system can more precisely determine where the compressor is operating and do a more effective job of keeping the compressor out of surge.

Knowing the mass flowrate, the head that the compressor must generate, and the compressor efficiency, you can determine the horsepower needed to drive a compressor. The efficiency, η, relates the actual work done on the gas (*i.e.,* compressing the gas) to the total work input into the compression system (*i.e.,* from the driver, overcoming bearing losses, etc.):

$$\eta = \frac{Work\ Out}{Work\ In} = \frac{k-1}{k}\left(\frac{\ln Pr}{\ln Tr}\right) \tag{4}$$

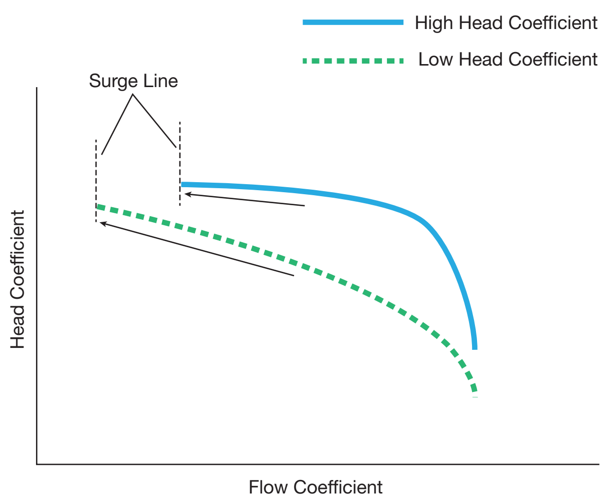
where k is the ratio of specific heats; Pr is the pressure ratio; and Tr is the temperature ratio. The right side of Eq. 4 is the formula for polytropic efficiency, the definition most frequently used by compressor OEMs.

The horsepower requirement is related to the mass flowrate, head, and efficiency as follows:

$$Horsepower \approx \frac{Mass\ Flow \times Head}{Efficiency} \tag{5}$$

Because horsepower is inversely proportional to efficiency, the compressor with the highest efficiency will require the least horsepower. Horsepower is directly proportional to both mass flowrate and head, so the higher the flowrate and/or head, the higher the horsepower requirement of the driver will be.

When selecting a centrifugal compressor, the OEM's application engineers will meet with the end user's process engineers to review the process requirements, develop various compressor arrangements, and then agree on the most effective configuration for the given application. The OEM will provide the end user with a formal proposal that includes the expected performance maps for the compressor or compressors needed to satisfy the requirements. ◆



▲ **Figure 13.** Compressors with a high head coefficient have a smaller rise-to-surge than compressors with low head coefficients. The black arrows under the curves indicate the level of rise.

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