12

DESIGN CONSIDERATIONS AND MANUFACTURING TECHNIQUES

12.1 AXIALLY VS. RADIALLY SPLIT

Figures II.3 and II.4 illustrated the axial and radial options that are available for many centrifugal process gas compressors. The decision as to whether an axially or radially split casing should be used depends on a number of factors that are highlighted next.

12.2 TIGHTNESS

The radially split design has circular casing joints or flanges with a perfectly even load distribution (Figs. 12.1 and 12.2). The leakage of gas at the two covers can thus be prevented most effectively. Besides metal-to-metal contact, "endless" O-rings are inserted in grooves on the two covers. By monitoring the pressure between two adjacent rings, the tightness can be controlled. For toxic, flammable, and explosive gases the barrel design is therefore always of advantage. For this reason, the latest issue of the API Standard 617 specifies the radially split casing construction for gases containing hydrogen if the hydrogen partial pressure exceeds 13.8 bar (200 psig).

12.3 MATERIAL STRESS

The cylindrical design with the smallest possible inner diameter is obviously the most suitable construction. With axially split casings the space available for bolting is further restricted at the two shaft penetrations. To achieve the tightness required, high contact pressure at the joints is required. The necessary forces in the bolts are often higher than would

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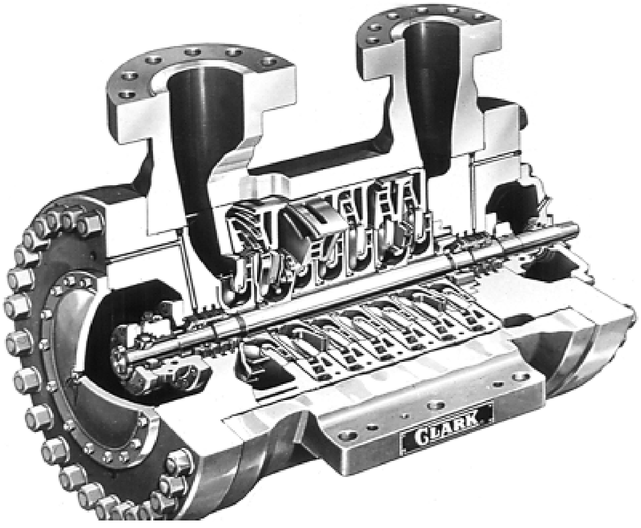


FIGURE 12.1 Radially split compressor with bolted-on heads, suitable for low-pressure service. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.2 Radially split compressor with a shear-ring head closure. *(Dresser-Rand Company, Olean, N.Y.)*

be required by the static gas forces if the casing flanges were perfectly rigid and flat. For large compressor frame sizes the radially split design can therefore be the only possible solution, even at moderate pressures.

12.4 NOZZLE LOCATION AND MAINTENANCE

For operating pressures where an axially split design would be perfectly adequate, barrel compressors are sometimes preferred since the nozzles can be arranged in any radial direction. If the necessary space in the axial direction at the nondriven shaft end is available for the horizontal pullout of the inner casing cartridge, inspections, rotor changes, or complete cartridge replacements can be accomplished quickly without removing any process piping to and from the compressor. For tandem units, on the other hand, the first or low-pressure casing should be of the axially split design up to the highest possible operating pressure. A barrel compressor coupled at both shaft ends has to be removed for overhaul or replacement of the rotor.

Seals and bearings in state-of-the-art barrel compressors can, however, always be serviced and replaced with the barrel casing remaining in place. Finally, it should be noted that only at moderate pressures can a ring of sturdy bolts successfully attach the end walls to the outer casing (Fig. 12.1). As pressure increases, however, it becomes mechanically impractical to provide a sufficient number of bolts of sufficiently large diameter to contain the pressure. This is when shear-ring enclosures (Figs. 12.2 and 12.6) are the preferred solution.

Compressor manufacturers are often able to give graphical guidelines or plots that allow a purchaser to zero in on probable casing recommendations. This is done in Fig. 12.3 (compare Fig. 12.74), where the potential client can determine which Elliott compressor frame should be selected to compress 33,000 inlet cfm (56,200 inlet m^3/h) of process gas to a gauge pressure of 450 psi (31 bar).

Plots of pressure and flow determine that the horizontally split 46 M frame, with a capacity of 22,000 to 34,000 inlet cfm (37,000 to 58,000 m^3/h), will serve. If a hydrogen-rich gas of very low molar mass is involved, a vertically split 46 MB (barrel-type) frame would be selected.

A vertical line drawn to the speed plot establishes that compressor speed can range approximately from 4600 to 6000 rpm. Since we are approaching the maximum flow capacity of this particular compressor frame, it would be best to operate in the upper half of the speed limit.

12.5 DESIGN OVERVIEW*

12.5.1 Casings

Horizontally split centrifugal compressors consist of upper and lower casing halves that are fastened together by stud bolts through mating flanges at the horizontal centerline. Where moderate pressures are encountered, this type of construction may offer advantages in maintainability. Access to the internals of the compressor is gained by a single vertical lift of the upper casing half. Figure 12.4 illustrates this design. Access to the radial bearings, thrust bearings, and seal for inspection or maintenance does not require removal of the upper casing half.

As depicted in Fig. 12.5, vertically split compressors consist of a case that is formed in the shape of a cylinder and open at either end. End closures or heads are attached at either end. Access to the internals of the compressor is gained by removing the outboard head.

Refer to Fig. 12.6 for a typical cross section of a centrifugal compressor, showing the major features that will be discussed, including the thrust and radial bearings, the seals, the rotor and impellers, and the various gas flow path configurations available.

* Sections 12.5 through 12.12 were developed and contributed by Harvey Galloway and Arthur Wemmell, Dresser-Rand Company, Olean, N.Y.

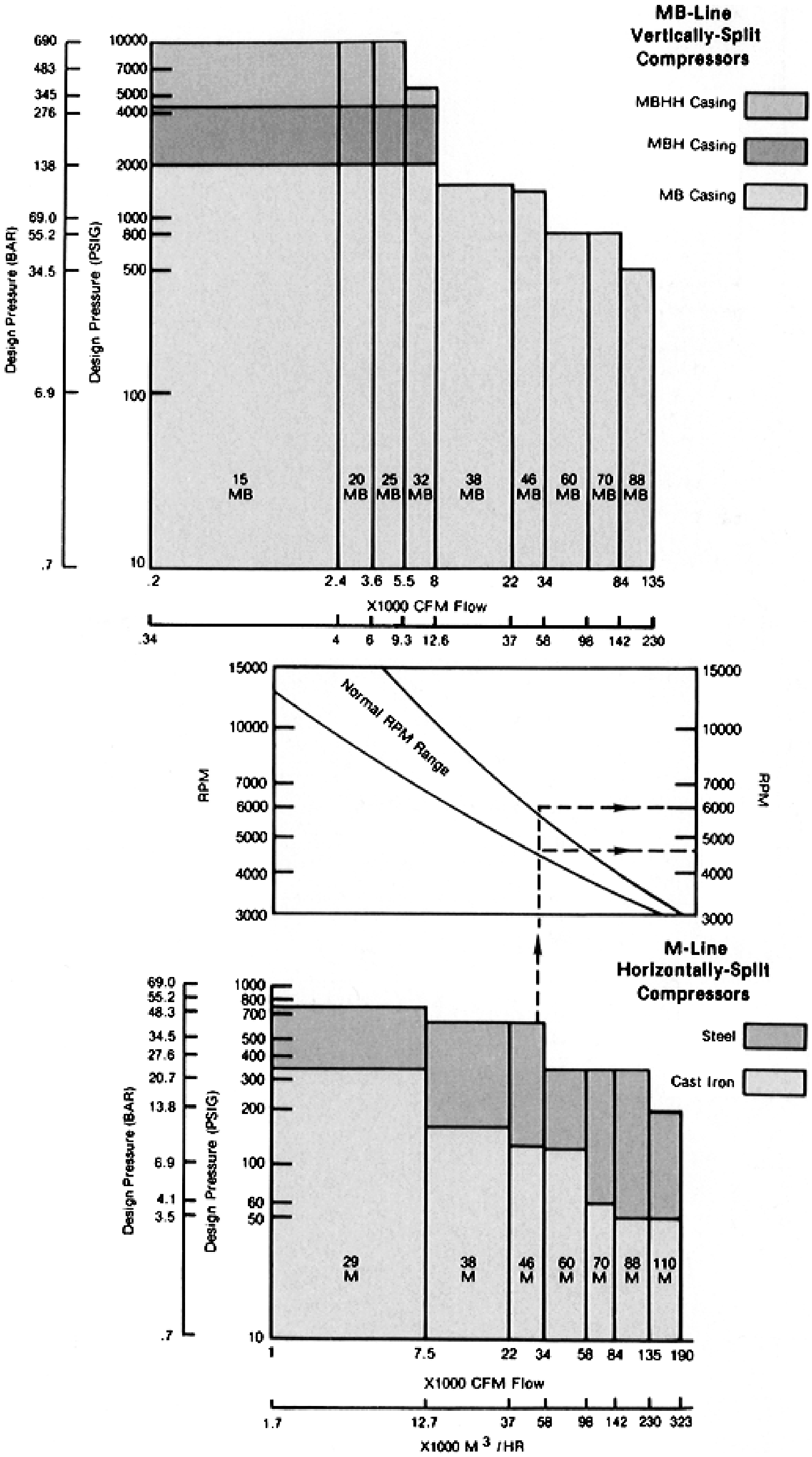


FIGURE 12.3 Typical centrifugal compressor selection chart. *(Elliott Company, Jeannette, Pa.)*

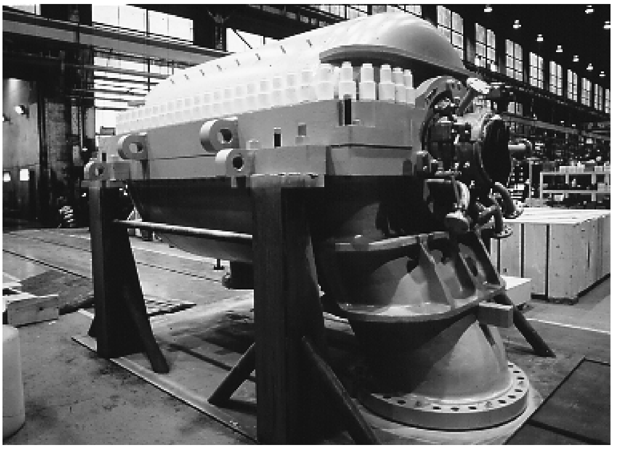


FIGURE 12.4 Horizontally split centrifugal compressor. *(Dresser-Rand Company, Olean, N.Y.)*

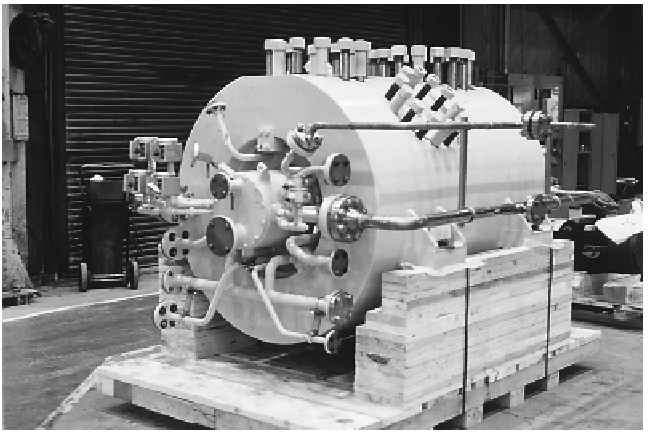


FIGURE 12.5 Vertically split centrifugal compressor. *(Dresser-Rand Company, Olean, N.Y.)*

Although our emphasis is on high-pressure multistage centrifugals, we do want to acknowledge the use of axial compressors in the petrochemical, refinery, and chemical industries. Accordingly, Fig. 12.7 depicts the combination of an axial-flow compressor at the initial gas intake with a radially bladed compressor section prior to gas discharge.

The multitude of compressor applications has resulted in a variety of case and nozzle configurations. The more common case and nozzle configurations are illustrated in Fig. 12.8.

Looking first at the flow path in Fig. 12.9, this compressor has a *straight-through flow path*, meaning that the gas enters through the main inlet of the compressor, passes through the guide vanes into the impeller, is discharged from the impeller into the diffuser through the return bend and into the next impeller, and so on until the total flow is discharged through the nozzle at the other end of the compressor.

Flexibility of nozzle orientation on a straight-through compressor allows various positions of the suction and discharge flanges. The most common orientation is both nozzles

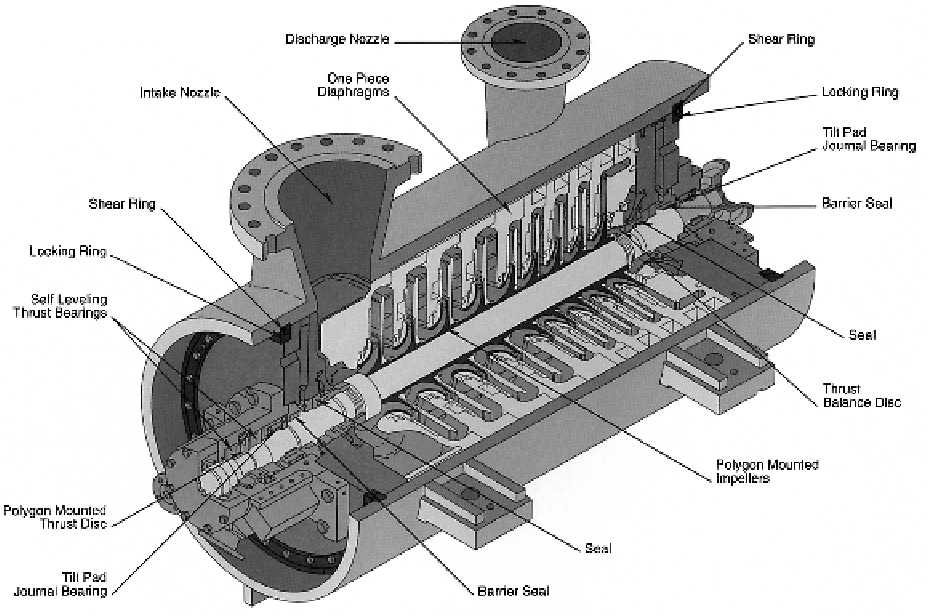


FIGURE 12.6 Major components of multistage centrifugal compressors. *(Dresser-Rand Company, Olean, N.Y.)*

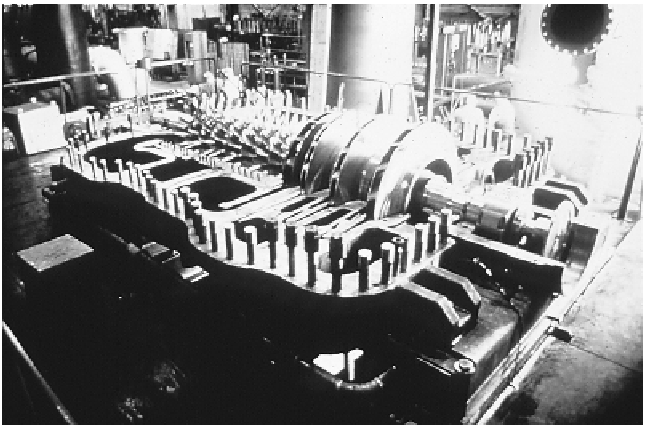
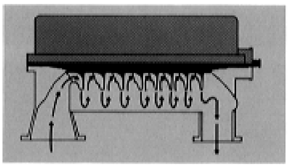


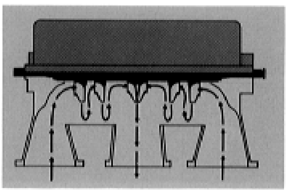
FIGURE 12.7 Combination axial/radial-flow compressor. *(Dresser-Rand Company, Olean, N.Y.)*

pointed up or down, but in some applications, such as gas recirculators and boosters, the nozzles are located on the side (Fig. 12.10).

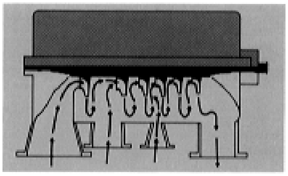
Reviewing the cross section of a *compound compressor* (Fig. 12.11), it can be seen that the flow path is the same as that of two straight-through compressors in series. That is, the total flow enters at the main inlet of the compressor and is totally discharged at the first discharge connection, is cooled or otherwise reconditioned, reenters the compressor at the second inlet connection, and is totally discharged at the final discharge nozzle.



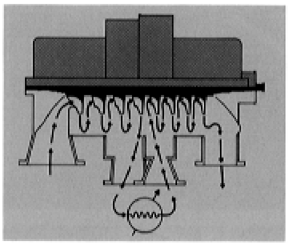
Straight-Through Flow
A convenient arrangement. May employ as many as 12 stages. Most often used for low-pressure process gas compression.



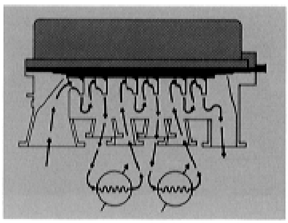
Double-Flow
Doubles the possible maximum flow capability of the compressor. Since the number of impellers handling each inlet flow is only half that of an equivalent straight-through machine, the maximum head capability is necessarily reduced.



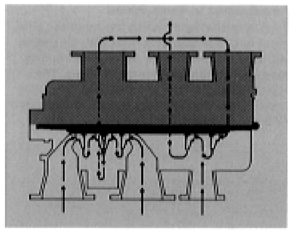
Side Streams
Side stream nozzles permit introducing or extracting gas at selected pressure levels. These flows may be process gas streams or flows from economizers in refrigeration compression. Side "loads" may be introduced through the diafram between two stages or, if flow is high, into the area provided by omitting one or two impellers.



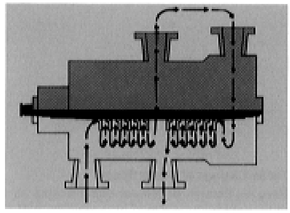
Iso-Cooling (cooling between stages)
When gas temperatures within the compressor reach 300 to 400°F (90 to 200°C), efficiency is improved and compressor power reduced by cooling the gas. Iso-Cooling nozzles permit the hot gas to be taken from the compressor to an external heat exchanger, then returned to the following stage at reduced temperature for further compression.



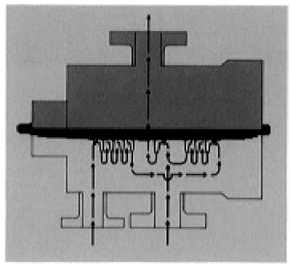
Double Iso-Cooling
High molecular mass gases such as chlorine (mass 70.91) heat much more rapidly during compression than do low molar mass gases. In such service, in order to maintain compression efficiency (and to avoid deposits within the compressor due to polymerization), it may be necessary to remove the hot gas from two points within the compressor, passing the gas stream through two intercoolers.



First Section Double-Flow
Capacity of this compressor is significantly increased by arranging double-flow compression in the first section. This is only feasible where head requirements are moderate to low, as in applications with low-pressure rise and/or high molar mass (propane or propylene refrigeration, for example).



Back-to-Back
Minimizes thrust when a high-pressure rise is to be achieved within one casing. Note how the thrust forces acting across the two "banks" of impellers tend to neutralize one another. A high-pressure, barrel-type casing.



Back-to-Back plus Recirculation Feature
This unit is typical for combined feed gas and circulator service. Last stage flow is combined with recycle flow. Mixing of flows can be either internal or external by means of an added discharge nozzle. Crossover flow is internal through passages in the inner casing.

FIGURE 12.8 Compressor casing and nozzle configuration. *(Elliott Company, Jeannette, Pa.)*

In applications with high-compression ratios, intercooling is desirable to minimize gas temperature and power requirements as well as to meet other process requirements. In many applications, compounding can reduce the number of compressor casings required. Figure 12.12 depicts a string, or train, of casings. Gas discharged from the first casing is led to suitable external heat exchangers before reentering into either the following compressor section or the next casing.

For high pressure ratios, arranging the impellers in a back-to-back configuration results in two inlet and two discharge nozzles (Fig. 12.13). This arrangement balances the axial forces on the rotor and eliminates the requirement for a balance piston, thereby reducing

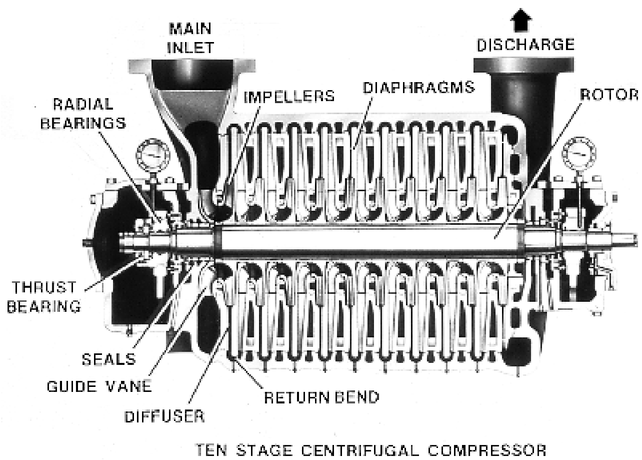


FIGURE 12.9 Straight-through compressor casing. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.10 Side-oriented nozzles on booster compressor. *(Dresser-Rand Company, Olean, N.Y.)*

shaft horsepower. Here, the gas flow enters the casing (case) at one end and leaves the case near the middle. The gas is redirected after externally cooling, if desired, in a second section at the opposite end of the case and finally leaves the casing in the center at the required discharge pressure.

Two high-pressure compressors operating in series are shown in Fig. 12.14 while undergoing a load-full pressure test. These units both utilize a back-to-back configuration with an integral crossover.

In the cross-sectional view of Fig. 12.15, examples of both incoming and outgoing sidestreams are shown. Flow enters the main inlet and is compressed through one impeller to

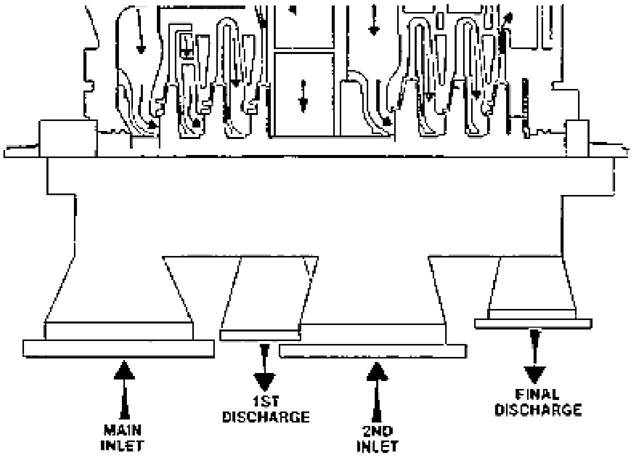


FIGURE 12.11 Cross-sectional view of a compound compressor. *(Dresser-Rand Company, Olean, N.Y.)*

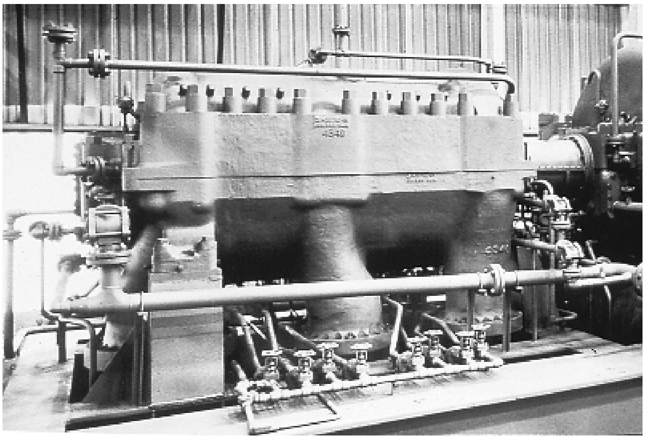


FIGURE 12.12 Compressor string, or casing train. *(Dresser-Rand Company, Olean, N.Y.)*

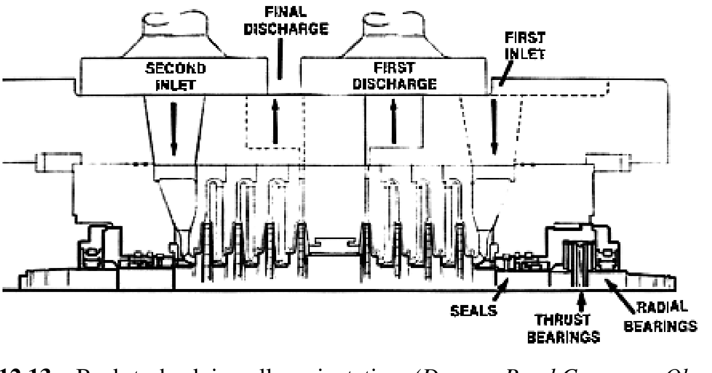


FIGURE 12.13 Back-to-back impeller orientation. *(Dresser-Rand Company, Olean, N.Y.)*

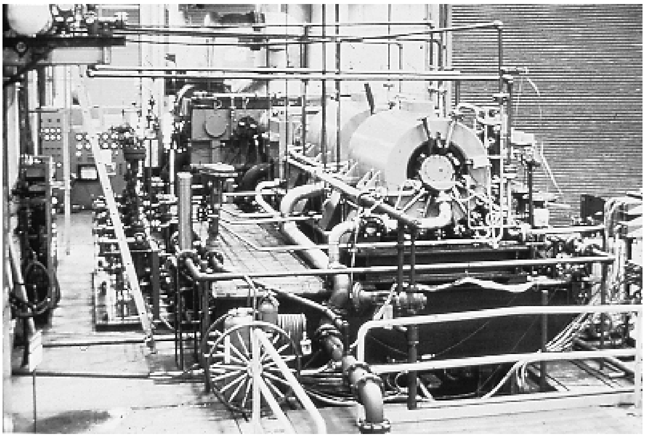


FIGURE 12.14 High-pressure compressor in a series flow arrangement. *(Dresser-Rand Company, Olean, N.Y.)*

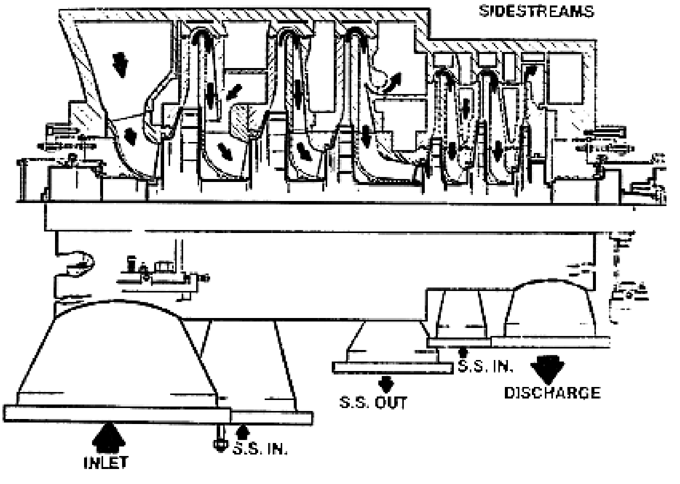


FIGURE 12.15 Cross-sectional view of a sidestream compressor. *(Dresser-Rand Company, Olean, N.Y.)*

an intermediate pressure level, at which point an incoming sidestream flow is mixed with the main inlet flow in the diaphragm area ahead of the next impeller. The total mixed flow is compressed to a higher pressure level through two impellers; a small portion of the flow leaves the compressor through an outgoing sidestream to satisfy a process requirement. The remainder of the flow is compressed through one impeller, mixed with an incoming sidestream, compressed through two stages, and exits through the final discharge.

For refrigeration cycles and other process requirements, the capability to admit or discharge gas at intermediate pressure levels is required. Compressors provide sidestreams with minimum flow disturbance and provide effective mixing of the main and sidestream gas flows (Fig. 12.16).

In the double-flow configuration of Figs. 12.17 and 12.18, the compressor is divided into two sections. It is effectively operating as two parallel compressors. An inlet nozzle is

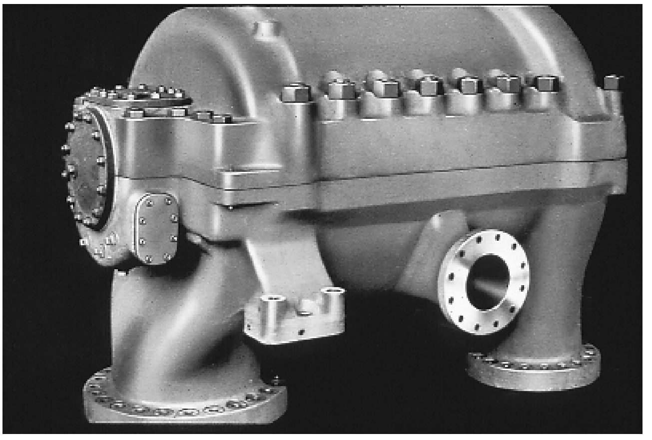


FIGURE 12.16 Sidestream compressor. *(Dresser-Rand Company, Olean, N.Y.)*

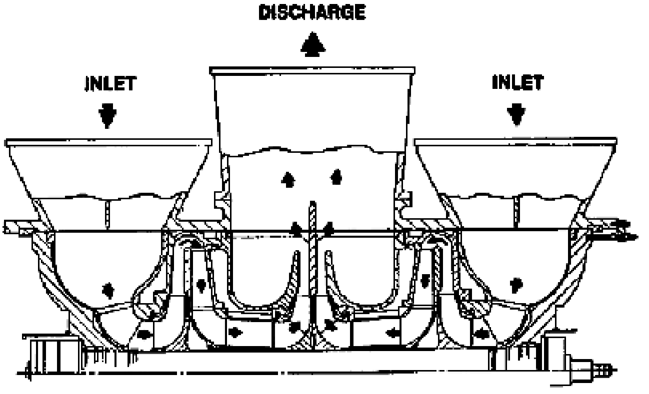


FIGURE 12.17 Cross-sectional view of a double-flow compressor. *(Dresser-Rand Company, Olean, N.Y.)*

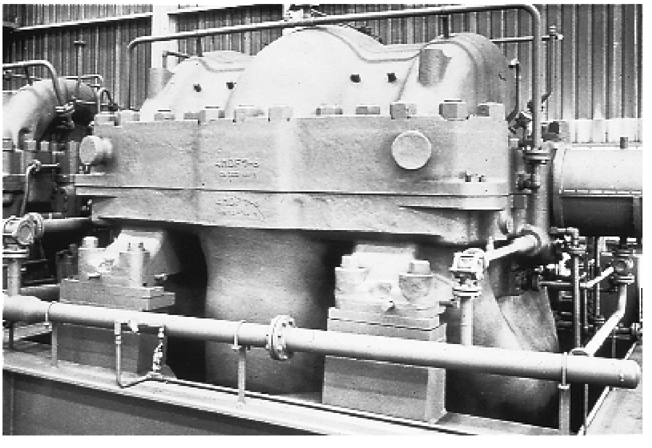


FIGURE 12.18 Double-flow compressor installation. *(Dresser-Rand Company, Olean, N.Y.)*

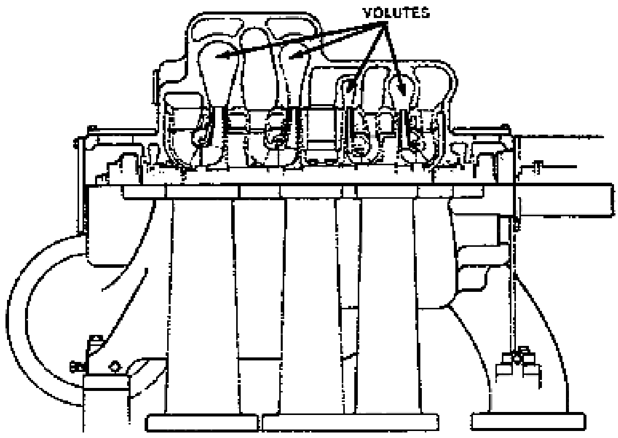


FIGURE 12.19 Cross-sectional view of an over-the-impeller volute-type diffuser. *(Dresser-Rand Company, Olean, N.Y.)*

located at either end of the compressor case. The discharge flow from each section is extracted from a common discharge nozzle at the center of the case. The impellers of one section face in the opposite direction from the impellers in the other section, achieving thrust balance over all operating conditions.

This concept effectively doubles the capacity of a given frame size and has several advantages:

- *Smaller frame.* For a given capacity, a compressor one frame smaller than the single-flow configuration can be used, thus reducing compressor costs.
- *Speed match.* In many applications, the flow from the double-flow compressor is discharged to a single-flow compressor of the same frame size. This permits operations at the same speed and allows the use of a single driver or duplication of drivers.

In yet other applications, it is desirable to cool after each stage of compression. The case provides eight nozzle connections, allowing intercooling after each impeller. Note that this particular type of design (Figs. 12.19 and 12.20) allows the use of over-the-impeller volute-type diffusers since the stage spacing required for the nozzle allows sufficient room for the volute area required. A typical application utilizing this case and nozzle arrangement is that of oxygen compression in an air separation plant.

In addition to the multistage centrifugal compressors discussed, single- and multistage overhung impeller designs are available for low-pressure-ratio applications. Several of these beam-type configurations are shown in Figs. 12.21 through 12.24. Allowable casing pressures have been extended up to 2000 psi for this type of casing. Design consideration must be given to startup when the casing is pressurized, because of high axial forces on the impeller.

Note the vertically split construction and the location of the inlet nozzle, which allows direct inlet flow into the impeller. These units are referred to as *direct inlet* or *axial inlet centrifugal compressors*. The fact that the inlet gas enters the impeller without requiring a 90° turn results in lower aerodynamic inlet losses (Fig. 12.23).

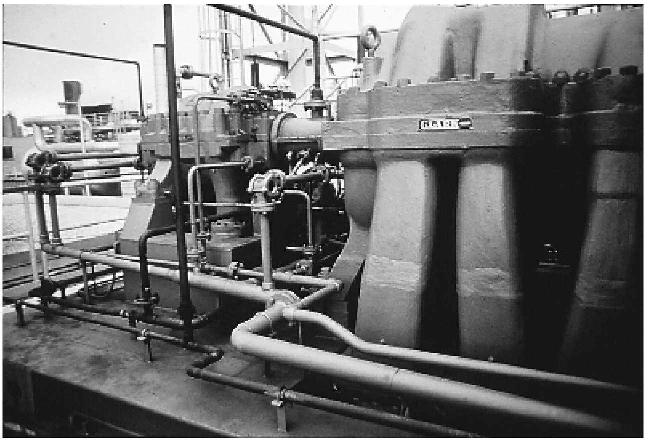


FIGURE 12.20 Compressor with intercooling after each section. *(Dresser-Rand Company, Olean, N.Y.)*

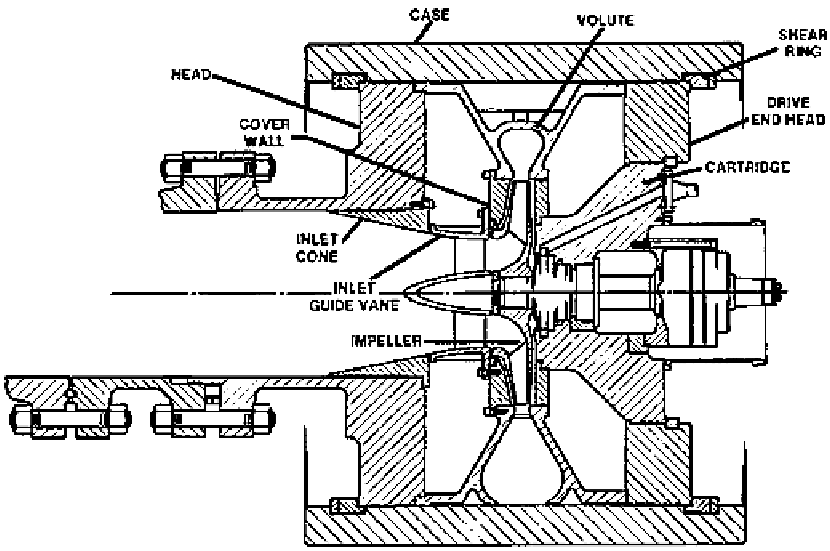


FIGURE 12.21 Cross-sectional view of a single-stage overhung impeller type of compressor. *(Dresser-Rand Company, Olean, N.Y.)*

The most common method of head retention for both vertically and horizontally split cases is by use of stud bolts. The previous use of stud bolts on vertically split casings has been superseded by the use of segmented shear rings.

On horizontally split casings (Fig. II.3) a large number of studs are secured in the lower case flange. The upper case flange has been drilled to receive the studs, and nuts are fastened on the studs when the upper case is properly positioned.

In earlier designs, vertically split casings of either between-bearing or overhung design were similarly fitted with studs that pass through the heads, and the heads are secured by stud

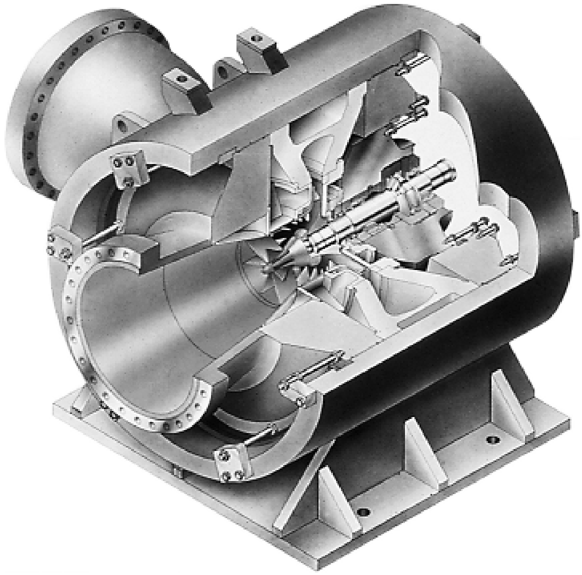


FIGURE 12.22 Single-stage overhung compressor for pipeline service. *(Dresser-Rand Company, Olean, N.Y.)*

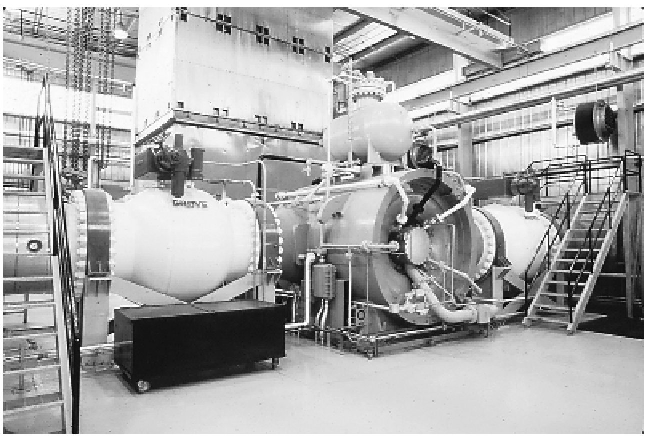


FIGURE 12.23 Single-stage overhung compressor installation for pipeline service showing side nozzles. *(Dresser-Rand Company, Olean, N.Y.)*

nuts (Fig. 12.1). In contrast, the current use of segmented shear-ring head retention designs for between-bearing as well as overhung design vertically split centrifugal compressors offers the benefits of greater strength and faster disassembly. Proper assembly is assured by elimination of precise torque requirements of bolted heads. In shear-ring designs, the head or end closures are retained by segmented shear-ring members positioned in an annular machined groove in the case. The incorporation of an O-ring assures positive sealing.

On large-capacity vertically split compressors (Fig. 12.24), the shear-ring head retainer eliminates the need to handle large stud nuts. A small secondary spacer ring assembly allows

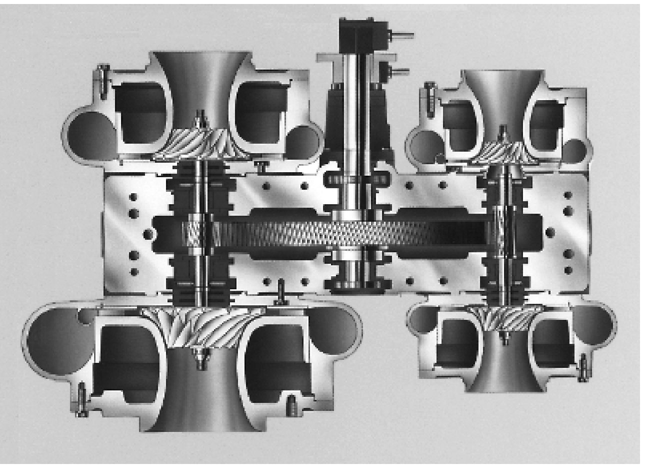


FIGURE 12.24 Multistage overhung impeller compressor. *(Mannesmann-Demag, Duisburg, Germany)*

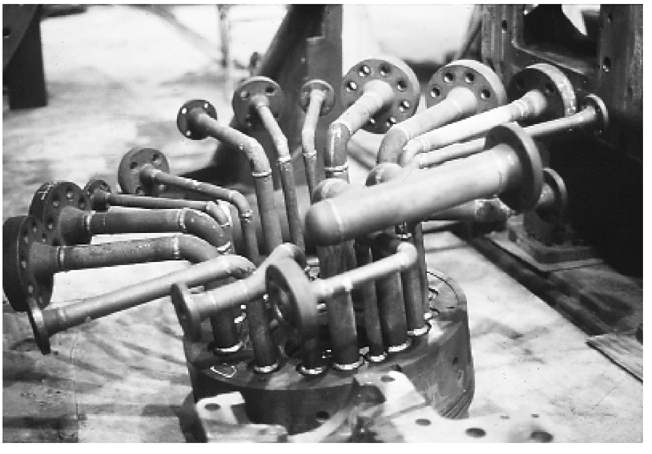


FIGURE 12.25 Piping connections entering a compressor head. *(Dresser-Rand Company, Olean, N.Y.)*

removal and installation of the primary shear ring. As illustrated in Figs. 12.2 and 12.6, the most advantageous use of shear-ring head retention is with very high pressure casings. It is difficult to properly design bolted heads for casings with pressure ratings over 5000 psi because of the physical size of the studs required.

Moving to Fig. 12.25, we note how surprisingly many external connections are required to enter the case through the heads. As shown in the photograph, virtually all available space is used in a small high-pressure head by the various external support systems such as lube and seal oil supply and drains, vents, and reference pressure lines.

As discussed later, compressor casing materials consist of cast iron or cast steel, fabricated steel, or forged steel. Horizontally split centrifugals commonly are produced from castings or fabrications, whereas vertically split casings use all three types. The cast steel or cast iron casings are more commonly used in horizontally split cases since case pressure

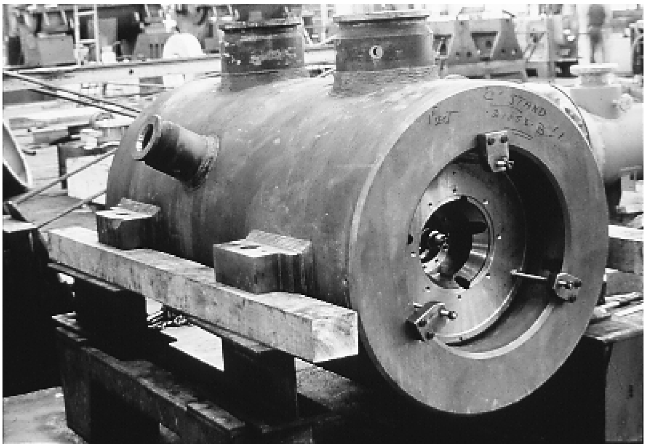


FIGURE 12.26 Reinjection compressor with a forged steel vertically split casing. *(Dresser-Rand Company, Olean, N.Y.)*

ratings are relatively low. Case patterns allow flexibility in the number of stages but result in fixed- as opposed to variable-stage spacing.

Advanced manufacturing and welding techniques are also being used in the production of compressor casings. Welded construction has supplemented the high-capacity line of cast cases and provides the option of purchasing horizontally or vertically split compressors for high-capacity low- and medium-pressure service. Manufacturing techniques used for welded casings are basically the same whether the compressor is horizontally or vertically split.

Forged steel vertically split casings (Fig. 12.26) are required for very high pressure applications. Case pressure ratings over 10,000 psi have been tested successfully for gas reinjection applications with units operating in the field well over 7000 psi.

12.5.2 Flow Path

Having completed our review of compressor casings, we now turn our attention to the stationary flow path. The stationary flow path is contained within the casing and is matched aerodynamically to the inlet and outlet passages and the impellers. Figure 12.27 depicts this stationary flow path in a vertically split compressor cross section. Note the casing components that include the heads.

Moving to a more detailed examination of a typical centrifugal compressor, we observe in Fig. 12.28 the stationary flow path and rotor, removed from the casing. This part of the compressor, referred to as the *bundle*, contains guide vanes, diaphragm, return bend, diffuser, and discharge volute. It is evident that on a vertically split compressor, the complete bundle is removed as a single piece. To remove the rotor, the bundle is usually designed with horizontal splits that allow removal of the top half. Some compressor designs employ one-piece rather than split-bundle stationary components, which are stacked onto the rotor shaft at the same time the impellers are installed.

Note also that the inlet wall is the first diaphragm and completes the inlet channel. The gas passage formed by the inlet wall directs the gas into the first impeller. The opposite side, inboard, of inlet wall forms the diffuser area from the first impeller. After the diffuser, the gas enters the return bend or crossover, which turns the gas streams 180°. The return channel guides the gas into the next impeller. The diaphragm is the stationary element between

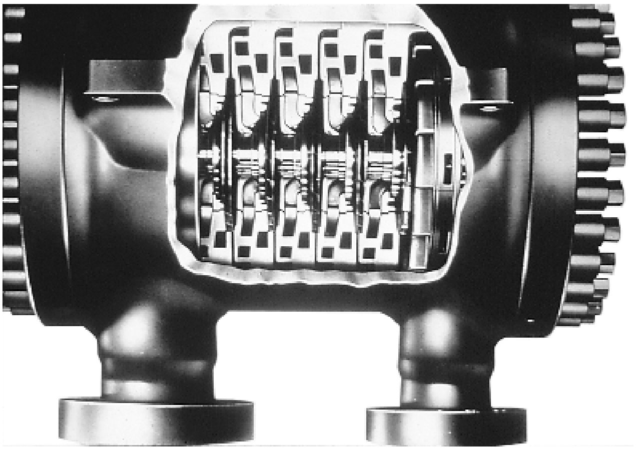


FIGURE 12.27 Stationary flow path in a vertically split compressor. *(Dresser-Rand Company, Olean, N.Y.)*

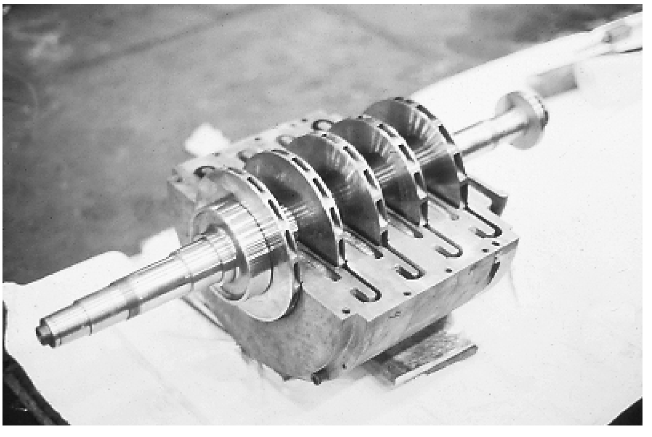


FIGURE 12.28 Stationary flow path and rotor. *(Dresser-Rand Company, Olean, N.Y.)*

two stages that forms half of the diffuser channel stage, the return bend, and half the return channel.

Figure 12.29 shows how the inlet guide vanes direct gas flow into the impeller "eye." One method of controlling the stage performance characteristics is through the use of different inlet guide vane angles. Guide vanes can direct the flow into the impeller against rotation, radially, or with impeller rotation. The guide vanes shown are of fixed predetermined angles.

If we want to change the compressor operating performance by means other than speed reduction, movable inlet guide vanes can be incorporated. They are more effective on single-stage compressors (see Fig. II.1), with diminishing effect as stages are added. It is extremely difficult from a mechanical standpoint to install and operate movable inlet guide vanes (Fig. 12.30) in any but the first stage of a centrifugal compressor.

Next, in Fig. 12.31, the return bend is clearly shown in the internal portion of the stationary flow path. After the gas leaves the last-stage diffuser, it is collected in a discharge

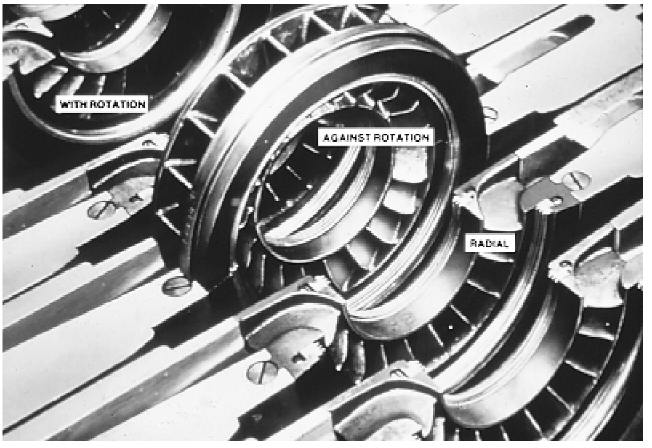


FIGURE 12.29 Inlet guide vanes, stationary (fixed) type. *(Dresser-Rand Company, Olean, N.Y.)*

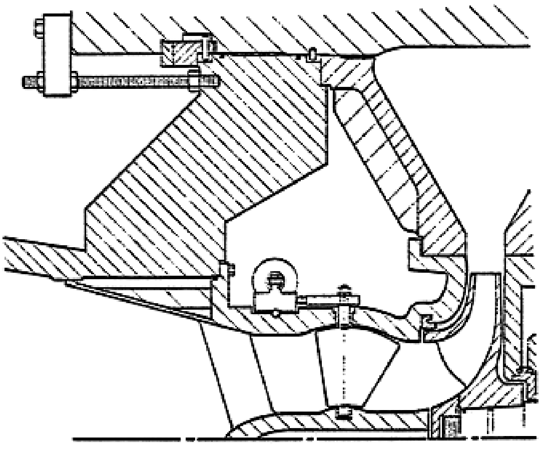


FIGURE 12.30 Movable inlet guide vanes. *(Dresser-Rand Company, Olean, N.Y.)*

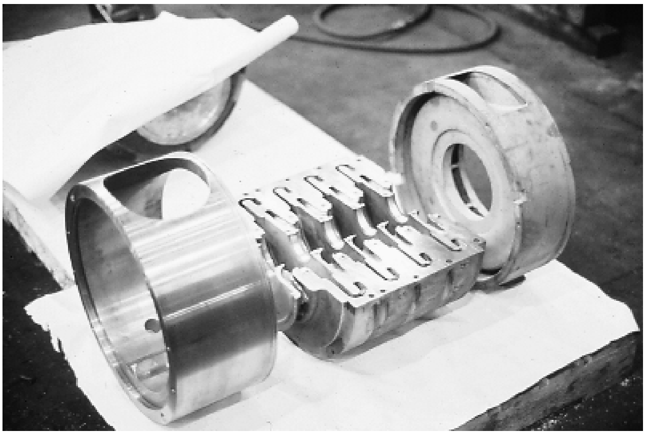


FIGURE 12.31 Internal return bend flow path. *(Dresser-Rand Company, Olean, N.Y.)*

volute (Fig. 12.32) and then directed to the discharge nozzles. The figure on the left shows a typical scroll-type over-the-impeller volute; on the right is a parallel wall diffuser followed by a more conventional spillover volute.

The scroll-type over-the-impeller volute (Fig. 12.19), as well as the spillover volute, are complex shapes from a manufacturing point of view and are usually a casting. The over-the-impeller volute does offer low-diffusion losses over a wide operating range but requires a larger case diameter and stage width than does the parallel wall diffuser-spillover volute. In horizontally split compressors the stationary flow-path components are designed so that they are removed with the top half. The rotor, resting in the lower half, is then readily accessible for inspection and/or removal. This is shown in Fig. 12.33.

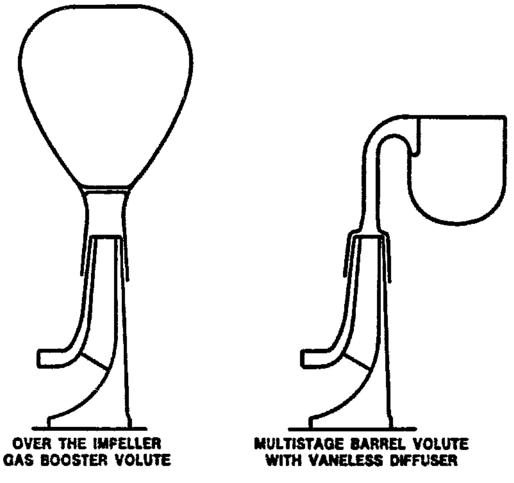


FIGURE 12.32 Discharge volutes: over-the-impeller and spillover types. *(Dresser-Rand Company, Olean, N.Y.)*

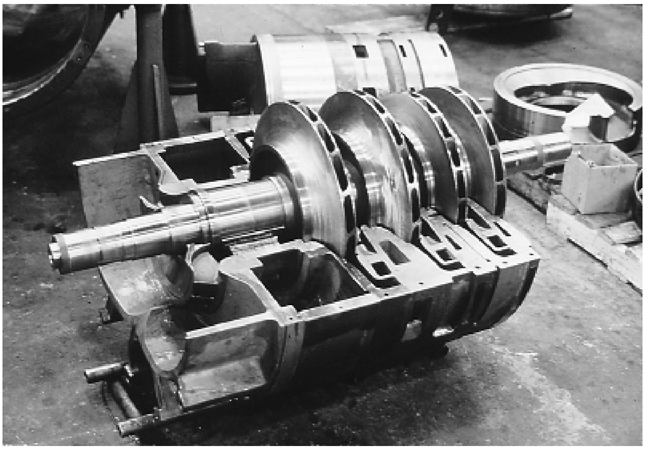


FIGURE 12.33 Compressor rotor resting in the lower half of stationary flow path components. *(Dresser-Rand Company, Olean, N.Y.)*

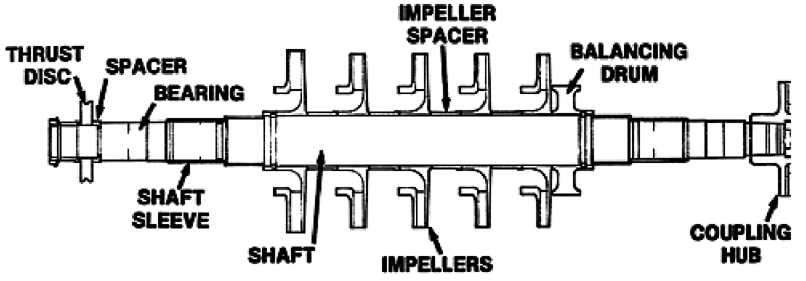


FIGURE 12.34 Compressor rotor nomenclature. *(Dresser-Rand Company, Olean, N.Y.)*

12.5.3 Rotors

Having examined the compressor case and stationary flow path, we now turn our attention to the rotating flow path, or rotor. A thorough understanding of rotor nomenclature is necessary to understand some important design considerations. Rotor nomenclature is explained in Fig. 12.34.

There is a step in the shaft at the bearing area. On the thrust bearing end, a precision ground spacer that butts against the shaft shoulder is installed. This spacer locates the thrust disk, which in turn will locate the rotor in the compressor.

The major components of a centrifugal compressor rotor are:

- Shaft
- Impellers
- Balancing drum (if required)
- Impeller spacer
- Thrust disk
- Coupling hub

In the seal area, sleeves are provided to protect the shaft. Under labyrinth seals, the sleeves are stainless steel; under oil film seals, the sleeves are often monel with a hard colmonoy or similar overlay to protect against scratches from dirt particles in the oil or gas.

Figure 12.35 shows a complete rotor being installed in a horizontally split centrifugal compressor. All rotor components have been attached to the shaft and the rotor has been balanced prior to installation.

The shaft is precision machined from an alloy steel forging. This solid rotor shaft design ensures maximum parallelism of rotor components. Impellers and balance pistons are normally forged steel, SAE 4330, with stainless steel available for corrosive gas applications. Impeller spacers are typically machined from a 400 series stainless steel.

Between each impeller is a spacer sleeve (Fig. 12.36). In addition to the function of locating the impellers on the shaft, sleeves also protect the rotor shaft in the event of contact with the labyrinths.

12.5.4 Impellers

A cross section of the impeller (Fig. 12.37) reveals the three components: blade, disk, and cover. The blade increases the velocity of the gas by rotating and causing the gas to move

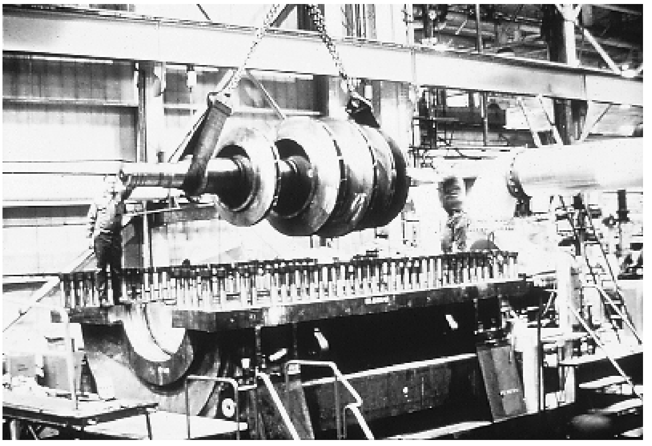


FIGURE 12.35 Compressor rotor installed in a horizontally split centrifugal compressor. *(Dresser-Rand Company, Olean, N.Y.)*

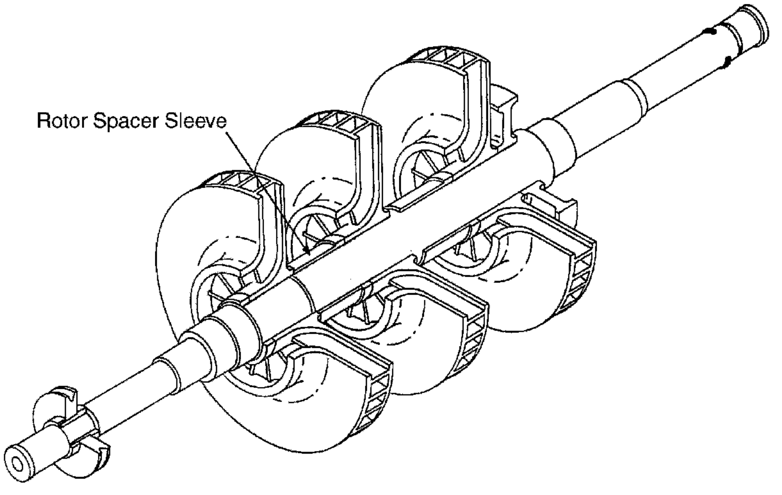


FIGURE 12.36 Spacer sleeves for a centrifugal compressor rotor. *(Dresser-Rand Company, Olean, N.Y.)*

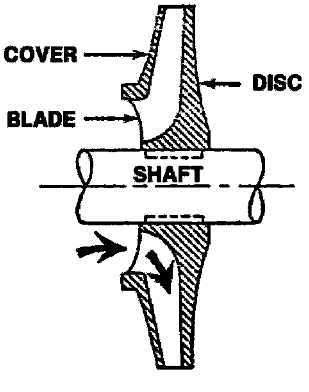


FIGURE 12.37 Cross-sectional view of an impeller. *(Dresser-Rand Company, Olean, N.Y.)*

from the inlet, the impeller eye, to the top or outside diameter. The impeller disk or hub is attached to the shaft and drives the blade. The cover is attached to the blades and confines the gas to the blade area.

To provide flexibility to meet the many process requirements, several types of impellers are used. These include closed impellers, which have a blade mounted between a disk and cover, the cover being the inlet side of the impeller; and open or semiopen impellers, which, as the names imply, consist of a disk and blade but have the cover removed (Fig. 12.38).

A number of manufacturing methods are used in the production of impellers. These include riveted construction, where the disk and covers are joined to the blades by rivets, cast construction, electrolytic machining, five-axis milling, and welding. It is noteworthy that five-axis milling is becoming increasingly important, and its availability for high-efficiency high-strength impellers should be explored first.

Over the history of the centrifugal compressor, the most universally used manufacturing method had been riveted construction. This method is rarely used today and has been replaced by welded impellers in modern installations.

Sand and die-casting techniques are available to fabricate various types of impellers. Advanced casting methods are applied in the manufacture of open impellers (Fig. 12.39).

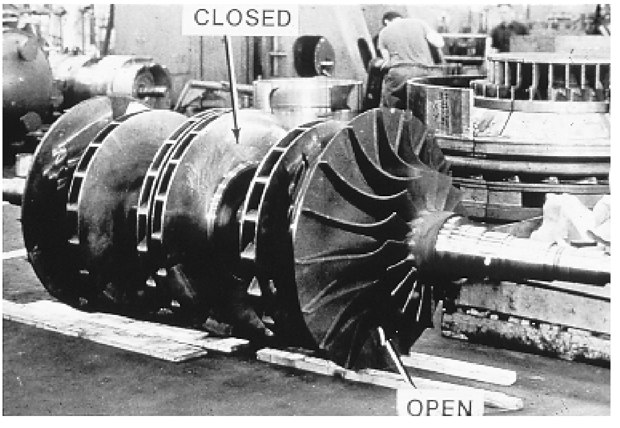


FIGURE 12.38 Impeller types found in centrifugal compressors. *(Dresser-Rand Company, Olean, N.Y.)*

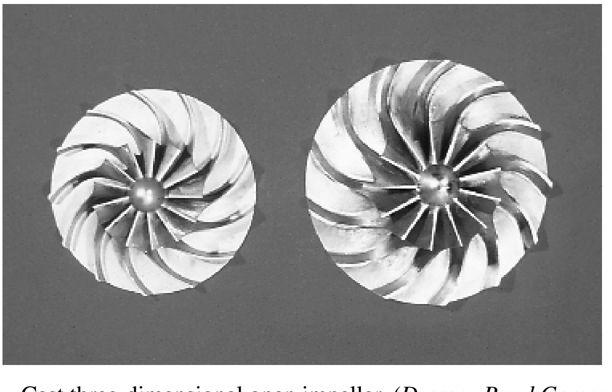


FIGURE 12.39 Cast three-dimensional open impeller. *(Dresser-Rand Company, Olean, N.Y.)*

Riveted impellers (Fig. 12.40) are fabricated by riveting the blade to both the cover and the disk. Two types of riveted construction exist. As shown, the blade sides are rolled over at a 90° angle and attached by rivets through the rolled-over portion. To decrease the susceptibility of the riveted impeller to corrosive and erosive failure, an integrally riveted impeller was developed. Integrally riveted impellers require a thicker blade since the rivets attach to the blade edge, thus eliminating the requirement to bend the blade 90° on both sides (Fig. 12.41).

Welded impellers are structurally homogeneous. Welded construction (Fig. 12.42) also allows maximum flexibility to alter aerodynamic designs. Pattern arrangements required for cast impellers and tools associated with electrochemical milling methods are not necessary to modify the aerodynamic design.

FIGURE 12.40 Riveted impeller, Z-type blading. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.41 Integrally riveted impeller. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.42 Structurally homogeneous welded impeller. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.43 Three-piece welded impeller. *(Dresser-Rand Company, Olean, N.Y.)*

In impellers of high specific speed, where the three-piece and open construction techniques are employed, three-dimensional blade shaping provides optimum aerodynamic geometry.

Varying manufacturing techniques are used to produce the different types of welded impellers, including:

- Three-piece construction
- Open impeller construction
- Two-piece construction [tip width greater than $\frac{5}{8}$ in. (16 mm)]
- Two-piece construction [tip width less than $\frac{5}{8}$ in. (16 mm)]

In three-piece construction (Fig. 12.43) the disk and the cover are machined from forgings. The die-formed blades are then tack-welded to the cover with the use of locating fixtures. The final welding is a continuous fillet weld between the blade and the cover. Subsequently,

FIGURE 12.44 Two-piece welded impeller. *(Dresser-Rand Company, Olean, N.Y.)*

the blade cover assembly is joined to the disk by a continuous fillet weld between the disk and the blade.

Impellers of open construction consist of a disk and blades (Fig. 12.39). The cover is eliminated. This type of impeller is characterized by an inducer section that directs the gas flow into the eye of the impeller. The blades are either die formed or precision cast. The welding procedure is the same as for three-piece construction, with the final weld being a continuous fillet weld between the disk and the blade.

In two-piece construction (Fig. 12.44) the blades are machined on either the disk or cover forging. The impeller is completed by a continuous fillet weld to the mating piece (disk or cover) around the entire blade interface. This type of construction is used for impellers with a relatively low tip width/diameter ratio (i.e., low specific speed).

Observing Fig. 12.44, we note that the welding techniques described previously are limited to impellers with a channel width of more than $\frac{5}{8}$ in. (16 mm). Increasing numbers of applications are now requiring the advantages of welded construction for impellers with a channel width of less than $\frac{5}{8}$ in. (16 mm). To meet this need, an advanced welding technique provides a method of welding the blades to the disk from the outside of the disk. The method produces an impeller with greater strength than that of a riveted impeller because of the continuous weld along the entire length of the blades. Any blade contour may be designed without affecting the weldability of the impeller, thus minimizing compromises in aerodynamic design. Using this technique, welded impellers can be manufactured to the smallest practical aerodynamic width.

Figure 12.45 illustrates the wide range of impeller sizes required to serve the widely varying process industry applications. The large rotor is from a compressor with flow to 180,000 cfm (5100 m^3/min) and operates to a speed of 4000 rpm. The small rotor is for a compressor that has a flow capability to 3500 cfm (100 m^3/min) and operates at speeds to 20,000 rpm. This wide range of welded impeller sizes and types requires a variation of manufacturing techniques.

Balancing drums (Fig. 12.46) are employed to modify or adjust the axial thrust developed by compressor rotors. These drums are typically required when all impellers are facing in the same direction. A balancing drum is mounted behind the last stage impeller, as shown in Fig. 12.47.

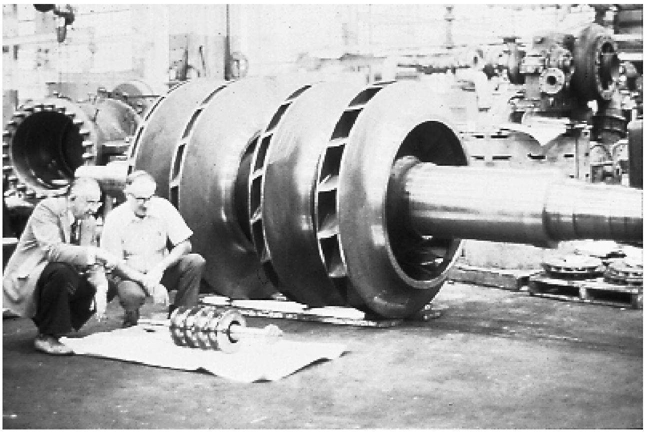


FIGURE 12.45 Rotor sizes used in centrifugal compressors. *(Dresser-Rand Company, Olean, N.Y.)*

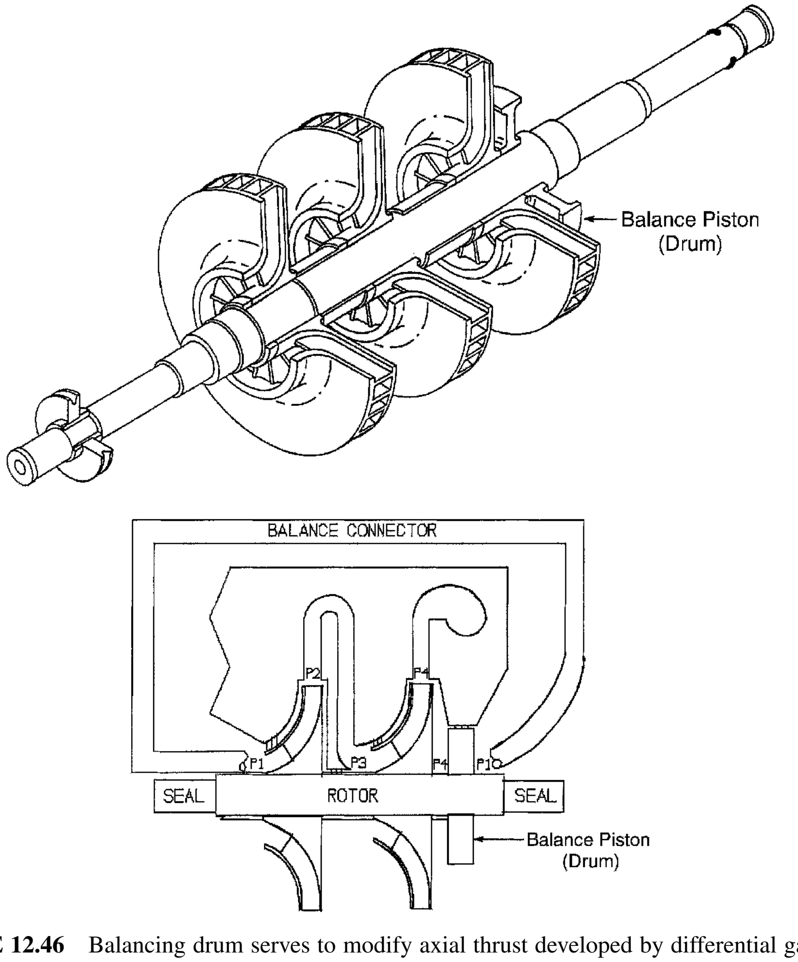


FIGURE 12.46 Balancing drum serves to modify axial thrust developed by differential gas pressures in compressors. *(Dresser-Rand Company, Olean, N.Y.)*

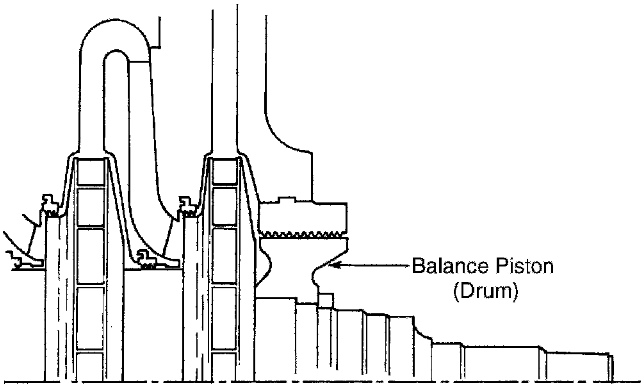


FIGURE 12.47 Balancing drum mounted behind a last-stage impeller. *(Dresser-Rand Company, Olean, N.Y.)*

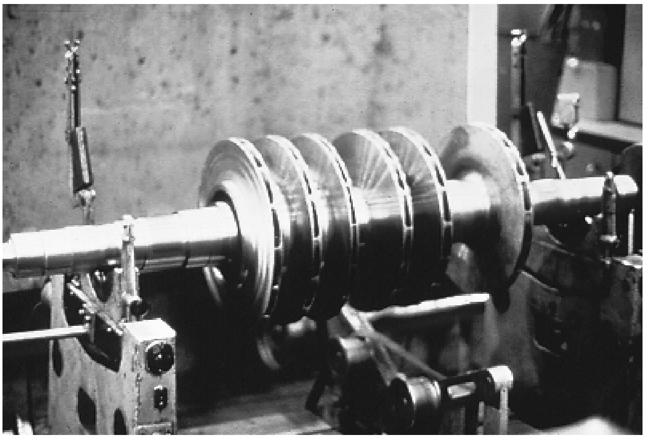


FIGURE 12.48 Centrifugal compressor rotor being balanced. *(Dresser-Rand Company, Olean, N.Y.)*

Impellers are mounted on the shaft with a shrink fit with or without keyways, depending on the frame size. Prior to rotor assembly, impellers are dynamically balanced and oversped. Impellers are mounted in pairs beginning at the center of the shaft; successive pairs of impellers are added, one from each end, until the rotor is complete. The rotor is dynamically balanced after the addition of each set of impellers. At each balancing operation, balance correction is done only on the newly added components. Figure 12.48 illustrates a balance operation.

As the flow requirements of a centrifugal compressor application increase, the use of radial-flow impellers may be restricted because of low efficiency. The development and use of mixed-flow impellers (Fig. 12.49) results in acceptable efficiencies since the gas is allowed to flow through the impeller channels at angles less than 90°. As the name implies, a mixed-flow impeller is neither radial flow or axial flow but somewhere between the two extremes.

FIGURE 12.49 Mixed-flow impeller. *(Dresser-Rand Company, Olean, N.Y.)*

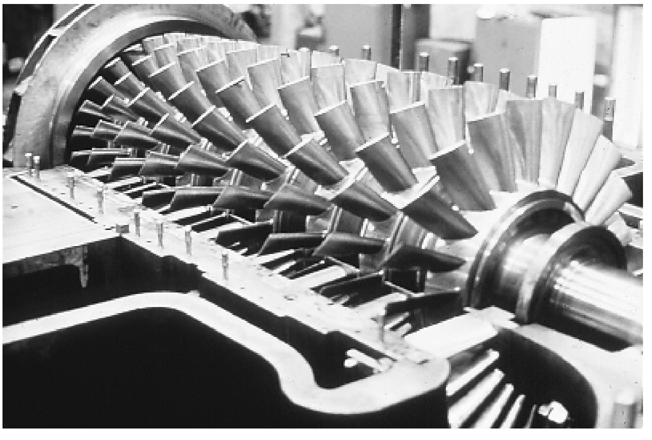


FIGURE 12.50 Axial-bladed compressor rotor. *(Dresser-Rand Company, Olean, N.Y.)*

12.5.5 Axial Blading

On very high flow applications at low suction pressure, such as atmospheric air, the use of axial-bladed compressors (Fig. 12.50) is attractive. Because the gas flows axially through the rotating flow path, turning losses are minimized and thus high efficiency is attainable. Limitations exist in using axial-bladed compressors in high-density gas streams, due to resulting high rotor thrust loads. Further consideration must be given to the fact that axial compressor performance generally results in less stability and reduced overload characteristics compared to those of typical radial impellers with backward bent blading.

12.5.6 Seals

The compressor industry has developed a complete range of seals for all types of applications. Four basic types of seals are offered: (1) labyrinth, (2) contact, (3) oil film, and (4) gas seals.

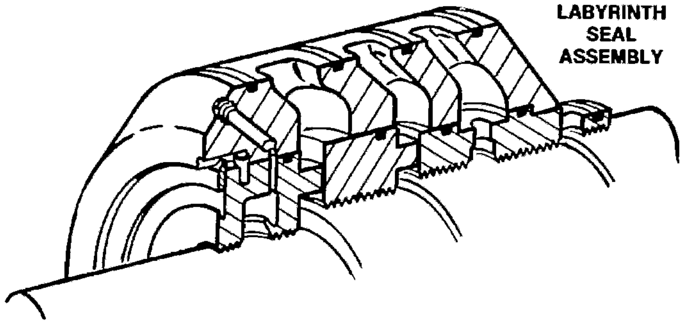


FIGURE 12.51 Labyrinth seals. *(Dresser-Rand Company, Olean, N.Y.)*

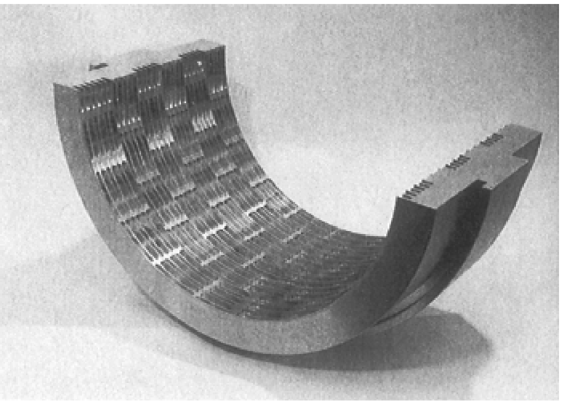


FIGURE 12.52 Special anti-instability labyrinth, or honeycomb seal. *(Nuovo Pignone, Florence, Italy)*

Labyrinth seals are suitable for use as main casing seals for compressors operating at moderate pressures. These seals are available with ports for injecting inert gas and/or educting process gas as required, depending on the process. This type of seal (Fig. 12.51) has been used for over 25 years in air and oxygen compressors. It is almost identical to the labyrinth seals typically found in steam turbines.

Modern labyrinth seals are often made of a honeycomblike material (Figs. 12.52 and 12.53). Honeycomb seals provide an order-of-magnitude more direct damping, lower whirl frequency ratios, and reduced leakage compared with conventional labyrinth seals. The whirl effects are causing aerodynamic instabilities or *rotating stall*. Counter-measures are sometimes based more on experimentation than on solid computer-generated analytical predictions. Nevertheless, both honeycomb seals and shunt holes (Fig. 12.54) will reduce both gas whirl (or swirl) risk and intensity. This abatement action is sometimes called *reduced cross-coupling*, although purists will assign minor differences to the respective definitions of the two terms.

Published nondimensional data on honeycomb seals are given in Figs. 12.55 and 12.56; these are of primary interest to designers of high-pressure compressors since swirl effects and the attendant aerodynamic instabilities are usually associated with high differential pressures.

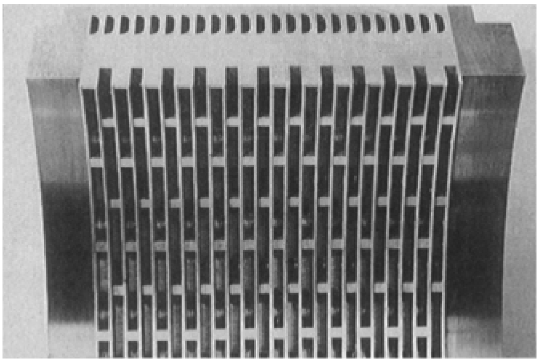


FIGURE 12.53 Honeycomb seal segment for a high-pressure centrifugal compressor. *(Nuovo Pignone, Florence, Italy)*

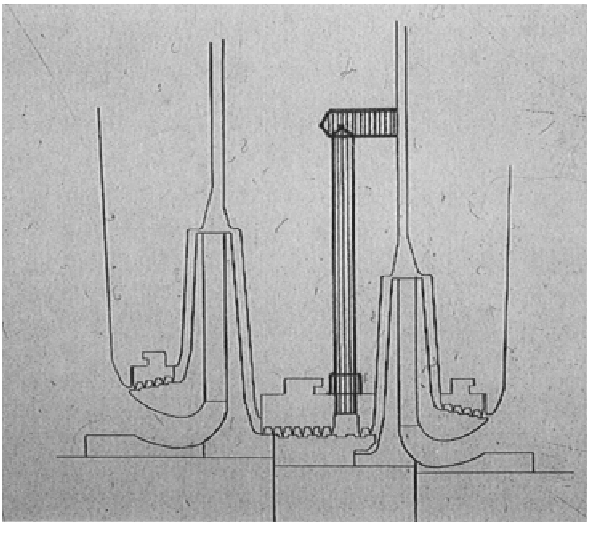


FIGURE 12.54 Shunt holes connecting the vaneless diffuser of the last impeller with the first grooves of the labyrinth. This minimizes inlet gas swirl and thus cross-coupling action. *(Nuovo Pignone, Florence, Italy)*

The mechanical contact seal (Fig. 12.57) was developed in the early 1950s. The major benefit of this type of seal is its ability to maintain a positive seal when the compressor and oil systems are shut down. Mechanical contact seals are suitable for intermediate pressures of 450 psi (32 kg/cm^2) and are particularly popular in refrigeration applications.

The primary components of this seal are:

- A spring-loaded stationary seal ring
- A floating carbon ring
- A rotating seal ring
- A spring-loaded shutdown piston
- An oil pressure breakdown ring
- A labyrinth with provision for buffer gas injection

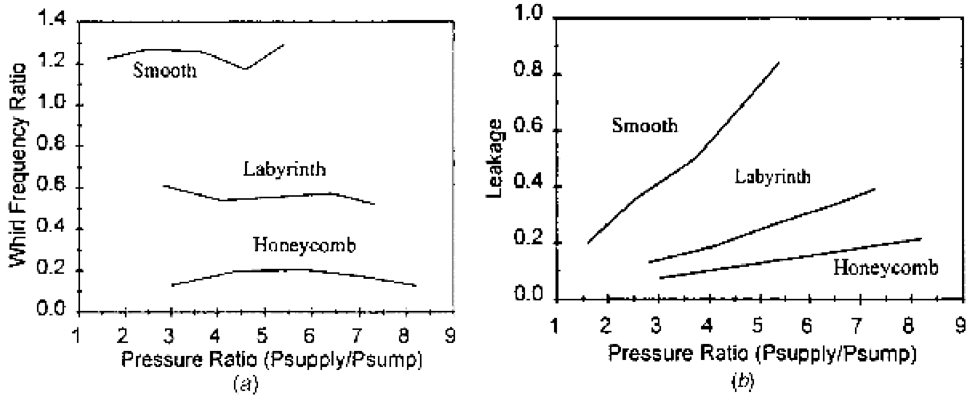


FIGURE 12.55 Rotor stability and leakage of honeycomb seals: (*a*) honeycomb seals improve rotor stability; (*b*) honeycomb seals leak less. *(Rotordynamics-Seal Research, North Highlands, Calif.)*

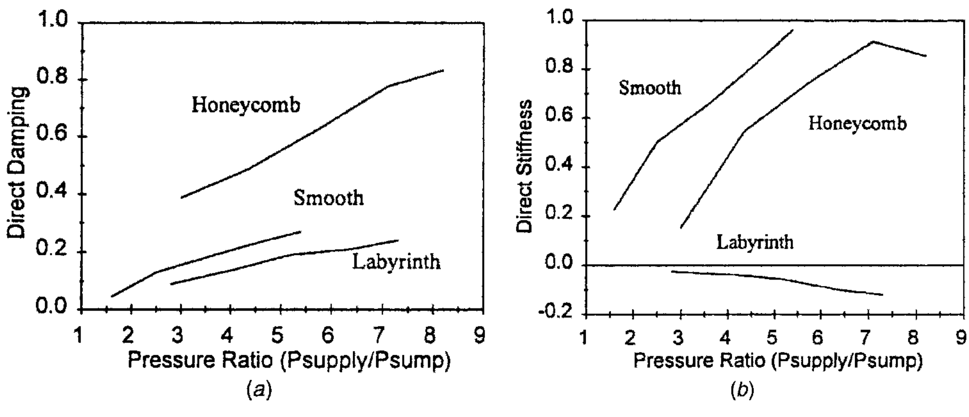


FIGURE 12.56 Damping and stiffness of honeycomb seals: (*a*) honeycomb seals produce more damping; (*b*) honeycomb seals have positive stiffness. *(Rotordynamics-Seal Research, North Highlands, Calif.)*

The contact seal design is unique in that it provides a separate sealing surface in the shutdown condition. In the event of a failure of the carbon ring, a standard contact seal would be inoperative. The design will still maintain a positive seal at shutdown conditions; consequently, it provides a fail-safe feature.

When the compressor is stopped under pressurization, the shutdown piston is held closed by this gas pressure. Sealing is accomplished by means of an elastomeric ring sealed against the rotating shaft ring. For operation, oil is introduced at a pressure of 25 psi (1 to 2 kg/cm^2) above the gas pressure. This oil pressure overcomes the gas pressure and spring tension on the shutdown piston, causing the piston to open and admit oil flow to the carbon ring seal. As the compressor is started, the carbon ring floats between the rotating ring and the stationary ring. The carbon ring seeks its own rotational speed, approximately one-half of shaft speed. The seal is loaded by springs, forcing the stationary ring against the carbon ring.

A small amount of oil flows across the carbon ring. This oil is prevented from contacting the gas stream by means of a labyrinth seal with an oil slinger on the shaft and is drained to

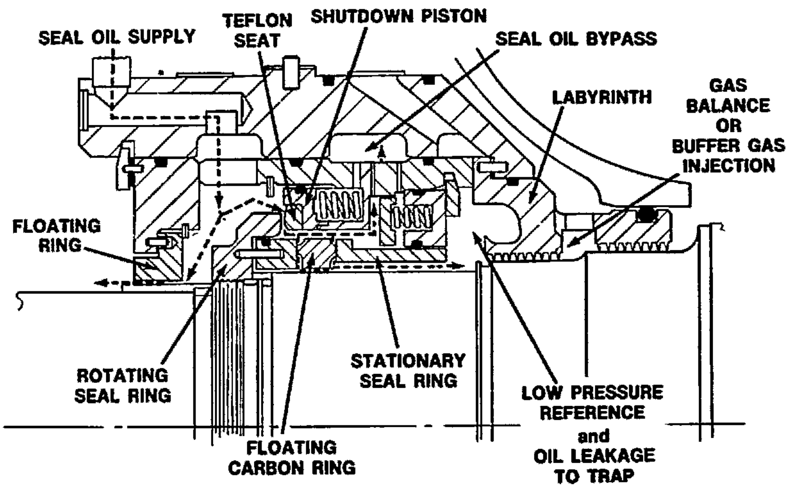


FIGURE 12.57 Mechanical contact seals. *(Dresser-Rand Company, Olean, N.Y.)*

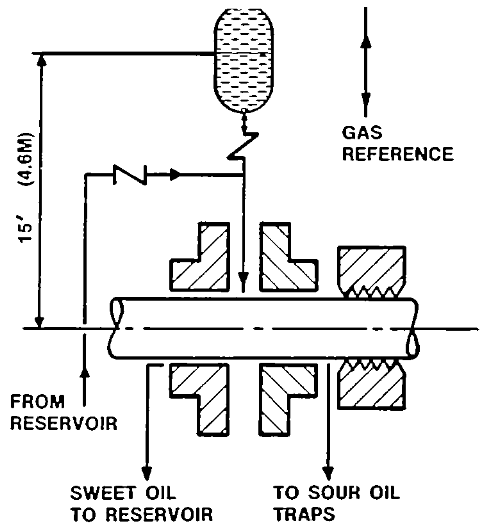


FIGURE 12.58 Oil film seal assembly. *(Dresser-Rand Company, Olean, N.Y.)*

a trap assembly. The majority of oil flows across the entire seal assembly, thus cooling the seal. It is then returned to the reservoir. A small amount of oil flows across a floating ring on the outboard end into the bearing chamber. The floating ring provides an orifice that maintains oil pressure in the seal area.

It was the development of the oil film seal (Fig. 12.58) that made possible the application of centrifugal compressors in high-pressure applications for hazardous gases. This seal design was introduced in gas transmission service in 1948. The major benefits of the oil film seal are:

- *Simplicity.* The oil film seal is simple in concept and does not involve rotating or contacting parts. This provides minimum service and ease of maintenance.

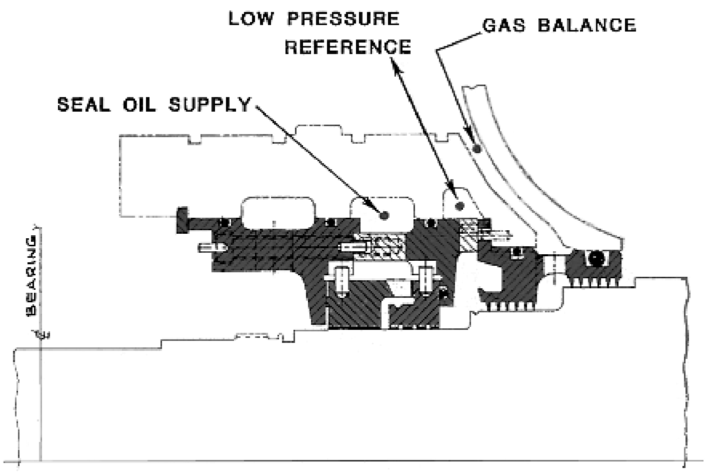


FIGURE 12.59 Oil film seal details. *(Dresser-Rand Company, Olean, N.Y.)*

- *High pressure.* The oil film seal has higher pressure capability than that of other types of seals, and with continued development, its capability may be virtually unlimited.
- *Fail-safe.* In the event of damage to seal rings, oil consumption will increase, but a seal will be maintained as long as sufficient oil is supplied, allowing continued operation of the compressor.

As further illustrated in Fig. 12.59, the oil film seal consists of two or more babbitted rings that do not rotate but are free to float radially to follow shaft movement. Oil from an overhead tank (see Fig. 12.58) is introduced between the rings at a pressure slightly above the reference gas pressure applied to the top of the tank.

The oil flows across these rings to the internal and external drains. Oil flow is quite small across the inner ring because of the low [5 psi (0.35 kg/cm^2)] differential pressure, which is controlled by the height of the tank above the compressor centerline. Oil flowing across the internal ring opposes the outward flow of gas, thus effecting a positive seal. This small amount of oil is insufficient for cooling the inner ring. The major flow of oil passes through the outer ring or rings and takes the full pressure drop between the oil supply pressure and the atmosphere drain to the reservoir. This oil flow passes by the back of the inner seal ring and provides cooling of this ring.

An optional labyrinth with provision for buffer gas injection can be provided, when required, to keep the seal oil drain separate from the lube oil drain. As described previously, the basic oil film seal consists of a high-pressure or inner ring and a low-pressure or outer ring. As seal pressures began to exceed 1000 psi (70 kg/cm^2), it became necessary to modify the seal to add another ring, allowing the pressure breakdown from seal supply pressure to atmosphere to be accomplished over two rings.

To properly seal a centrifugal compressor, careful attention must be given to gas pressure within the casing. A series of connecting ports allows the designer to seal against a known and predetermined internal pressure. The most significant element is to provide a pressure-flow bleed connection between the gas balance port and the seal reference area. While the recompression of this balance piston chamber gas requires additional shaft horsepower, it

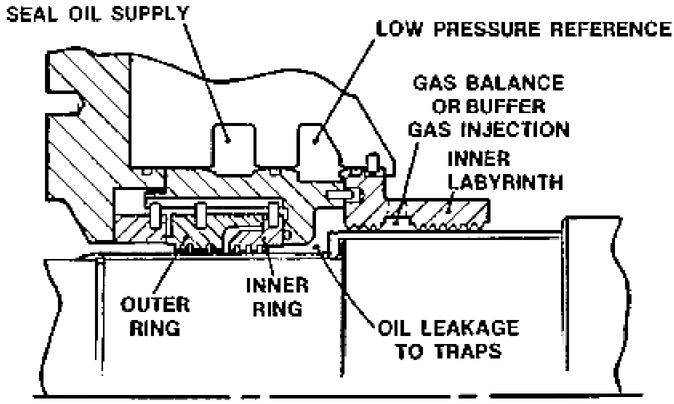


FIGURE 12.60 Balance piston chamber and seal reference area relationship. *(Dresser-Rand Company, Olean, N.Y.)*

results in sealing against an internal gas pressure slightly above suction pressure and not the high-discharge pressure. It also balances and thus equalizes the gas pressure on the seals at both the suction and discharge ends of the compressor, thereby simplifying the seal oil system controls. Figure 12.60 illustrates the essentials.

Trapped bushing seals (Fig. 12.61) are often incorporated in compressors made by the A-C Compressor Corporation. Under static conditions the trapped bushing seal operates like any other bushing-type seal. A sealant (normally, the oil from the lubrication system) is supplied at about 10 gal/min to the seal at a positive pressure above that of the process gas. The sealant flows in three ways:

1. A relatively small portion of the sealant buffers the process gas at 1 to 2 gal/hr in the clearance area between the inner portion of the stepped dual bushing (4) and the impeller (2). At this point the sealant enters the inner drain and is ultimately bled down to atmospheric pressure by way of a trap.
2. A larger portion of the sealant takes a pressure drop to the atmospheric outer drain between the outer portion of the stepped dual bushing (4) and the impeller (2). This flow rate is a function of the process gas pressure level.
3. The excess sealant passes through the stator (3) cooling passages and out of the seal, where a 5-ft head of sealant is maintained above the process gas pressure by a level control. This level controller (LC) modulates a level control valve to provide a pressure drop for the sealant's return to the reservoir.

While operating normally, the trapped section of the trapped bushing seal performs like a pump during dynamic operation. All of the sealant flow paths are exactly the same as during static operation except that the sealant buffering the process gas is whirled by the trapped portion of the seal, which consists of two principal parts:

1. The step within the stepped dual bushing (4) works in combination with the outer shoulder on the rotating impeller (2) as a pump, to ensure that the entire clearance area between the inner portion of the stepped dual bushing (4) and the impeller (2) is

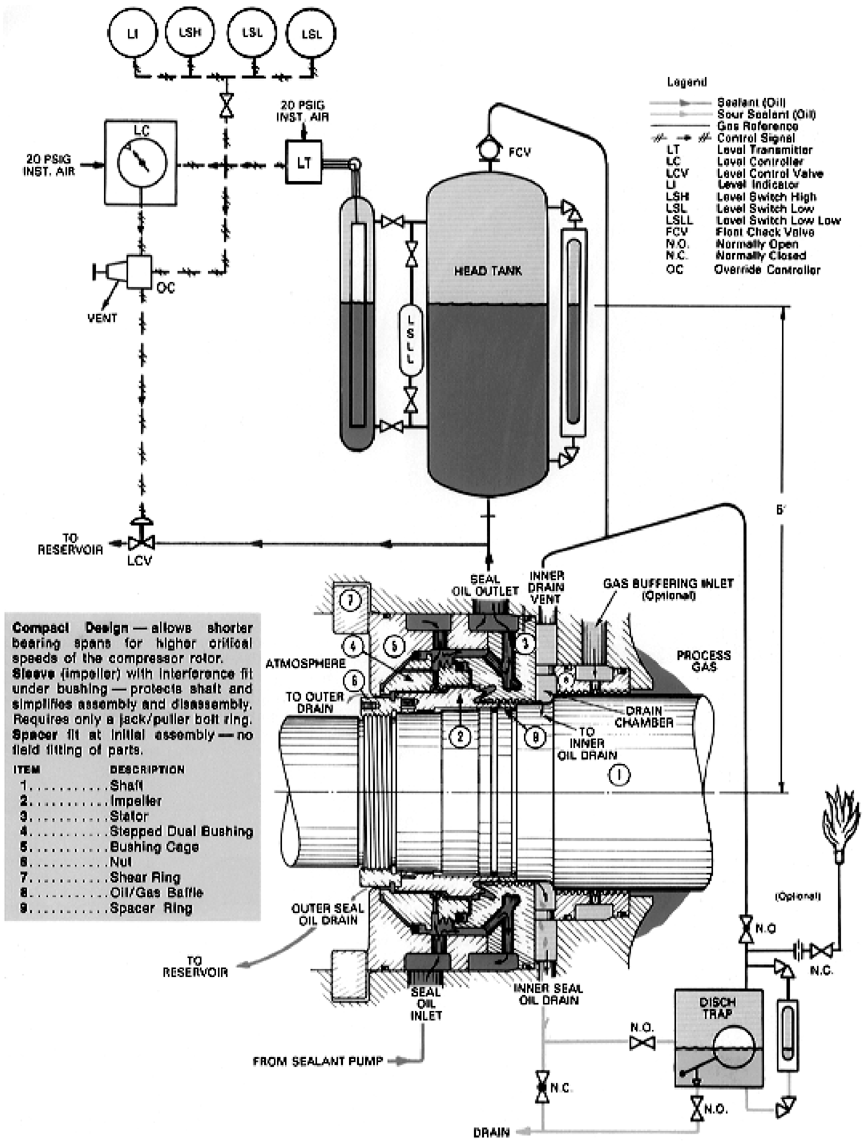


FIGURE 12.61 Trapped bushing seal system. *(A-C Compressor Corporation, Appleton, Wis.)*

a positive pressure level with respect to the process gas. Pressure patterns in this clearance area are similar to those in a lightly loaded journal bearing.

2. The portion of the stator (3) that meshes with the inner portion of the rotating impeller (2) acts as a dead-end pump without a suction source. The pumping action just balances the head of the pumping action of the outer shoulder of the impeller (2) plus the 5-ft head maintained in the head tank; hence, there is no pressure drop through the clearance between the stepped inner portion of the dual bushing (4) and the impeller (2). This pressure drop being zero, the inner sealant flow is considerably less than an untrapped bushing seal.

The fifth major type or configuration of compressor seal is the *gas seal*. Developed in the late 1970s, this relatively new sealing device is of sufficient importance to be given special coverage in Chapter 13.

12.6 BEARING CONFIGURATIONS

12.6.1 Radial Bearings

Radial bearings, sometimes referred to as *journal bearings*, support the compressor rotor. Technology in bearing design has increased significantly over the years to meet the increasing demands on the equipment. The original centrifugal compressors were furnished with plain sleeve bearings. The typical fully concentric straight-sleeve bearing was discovered to be inadequate as rotor speed increased dramatically in the late 1940s. In the 1950s, a pressure dam bearing design was developed to increase resistance to half-frequency whirl. The bottom half of a bearing liner is the same as in the straight-sleeve bearing, but the top half is relieved (Fig. 12.62). Thus, an area of high pressure is generated where the relief slot terminates. This increases the bearing loading. Continuing development has led to improvement in the stability of lightweight rotors operating at high speeds. This work culminated in the multishoe tilting pad radial bearing (Figs. 12.63 and 12.64). This bearing design has been so successful that it is now considered standard throughout the industry. In tilting pad radial bearings, the pad surface in contact with the bearing housing is radiused, allowing it

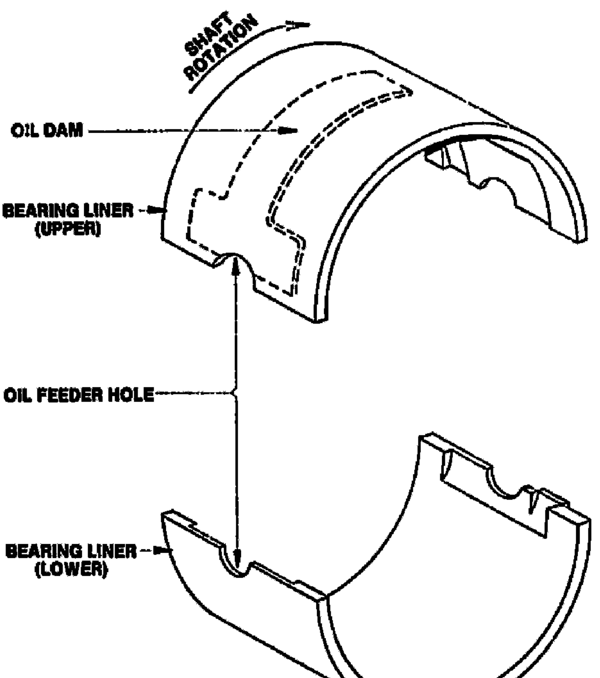


FIGURE 12.62 Pressure dam bearing. *(Dresser-Rand Company, Olean, N.Y.)*

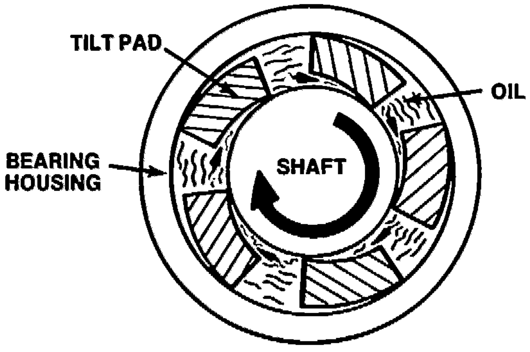


FIGURE 12.63 Tilt pad bearing. *(Dresser-Rand Company, Olean, N.Y.)*

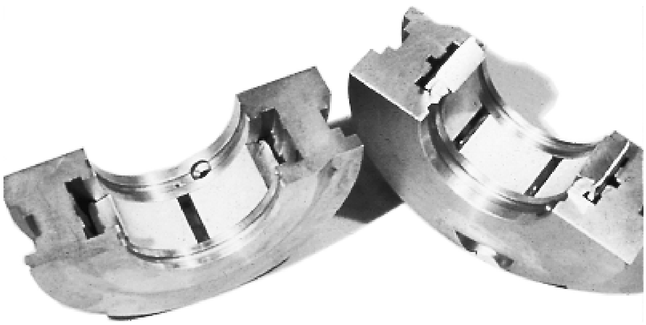


FIGURE 12.64 Radial tilt pad bearing assembly. *(Dresser-Rand Company, Olean, N.Y.)*

to pivot against the bearing housing. As the shaft rotates, a hydrodynamic film is formed between the journal and each pad. The oil enters the clearance between each pad and the shaft, tending to force the pad leading edge away from the shaft. Since the pad can pivot or tilt in its housing, the clearance at the trailing edge of the pad is reduced and the clearance at the leading edge of the pad is increased. This results in a wedge-shaped clearance between the pad and the shaft that produces hydrodynamic pressure in the bearing. By making design adjustments in the shape of the pads and bearing clearance, bearing stiffness and damping characteristics can be controlled.

The tilting pad bearing shown in Fig. 12.64 is a five-shoe bearing that has one pad located on the vertical centerline in the lower half of the bearing housing. This ensures that the shaft is supported properly when it is at rest. Other tilting pad bearings may have three or four shoes, and not all rotors are designed for load-on-pad (i.e., load-between-pad orientation may be necessary to obtain desired rotor behavior at high speeds).

12.6.2 Thrust Bearings

One of the most critical components of a centrifugal compressor is the thrust bearing (Fig. 12.65). Axial thrust is generated in a centrifugal compressor by the pressure rise through the impellers. The major portion of the thrust load is compensated by either a balancing drum or by placing the impellers in a back-to-back arrangement, whereby the thrust generated by

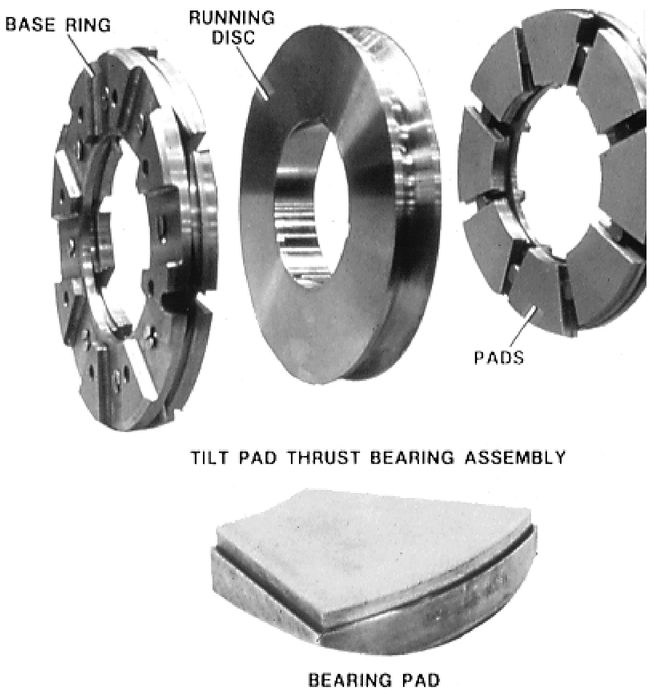


FIGURE 12.65 Thrust bearing assembly. *(Dresser-Rand Company, Olean, N.Y.)*

one set of impellers opposes the thrust generated by the other set of impellers. In either case, the relatively small residual load is carried by the thrust bearing. The thrust bearing must also be suitably designed to withstand additional load and thrust reversals that may occur during normal operating conditions.

The pressure environment surrounding each impeller creates an unbalanced axial force on the impeller and thus the rotor. The total axial unbalance on the rotor is calculated and is first compensated by the installation of a balance piston as part of the compressor design. The balance piston is generally sized to compensate for approximately $100 + 10\%$ of the total unbalance force generated by the impellers at the design operating parameters. The remaining or resulting unbalance force or thrust on the rotor is absorbed by the casing through the thrust bearing.

Figure 12.65 depicts the actual bearing parts, including the base ring, running disk, and bearing pads. This bearing design has been extremely successful as evidenced in successful operation in thousands of centrifugal compressors in the process industry. It has exhibited excellent load-carrying capabilities and even proven ability to withstand reverse rotation without damage to the bearings.

As an option, self-equalizing bearings (Fig. 12.66) can be provided on large frame compressors. This option can be applied when there is concern for potential thrust disk misalignment with the shaft. The self-equalizing bearing provides a uniform distribution of load on the thrust shoes over a wider range of thrust disk misalignment than can be accommodated by standard bearings. The disadvantage of this optional arrangement is that a greater shaft overhang is required which can become a limiting factor in critical speed analysis or rotor dynamics studies.

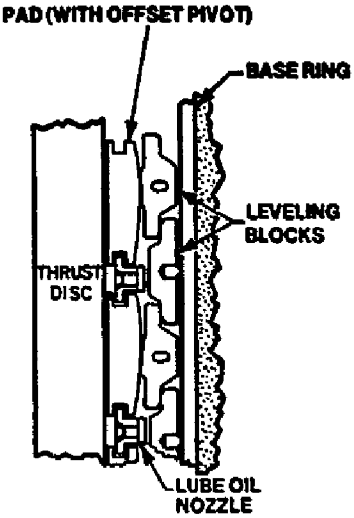


FIGURE 12.66 Self-equalizing thrust bearing. *(Dresser-Rand Company, Olean, N.Y.)*

12.6.3 Flexure Pivot Tilt Pad Bearings

Up to now, our overview dealt exclusively with rocking pivot tilt pad bearings. The insert in Fig. 12.67 allows us to compare these more conventional tilt pad bearings to a relative newcomer, the Flexure Pivot tilt pad bearing. The one-piece construction of this particular tilt pad bearing eliminates the multipiece construction typical of conventional tilt pad bearings. This also reduces the manufacturing tolerances significantly, as can be visualized from Fig. 12.68, which depicts a split-design Flexure Pivot tilt pad bearing. Thrust-type Flexure Pivot bearings have been supplied with a number of compressors, and Fig. 12.69 shows this simple, yet effective design.

A recent development is the hydraulically fitted thrust runner disk (Fig. 12.70). On centrifugal compressors, the thrust disk must be removable from the shaft to enable the inner seals to be removed from the shaft during maintenance. To prevent fretting of the shaft material under the thrust disk, the common method of attachment of the thrust disk is by shrink fitting. Shrink fitting by heating the disk is unacceptable because of the obvious hazards of applying open flame heat to the disk at the plant site.

The hydraulically fitted thrust disk shown here allows removal of the disk without heating the rotor. This design features a thin sleeve that is mounted to the shaft with a loose fit. The outside surface of the sleeve is cone shaped or tapered to allow the thrust disk to be mounted onto the sleeve by hydraulic pressure.

To remove the hydraulically fitted thrust disk, one proceeds as shown in Fig. 12.71. Oil at high pressure from pump 1 is supplied through holes in the center of the shaft and sleeve to expand the thrust disk over the sleeve. Pusher pump 2 pushes the disk on the sleeve. Releasing the hydraulic pressure on pump 1 allows the thrust disk and sleeve to shrink to the shaft. The high 1.5 mils/in. (1.5 μm/mm) of shaft diameter shrink fit ensures a tight fit on the disk and sleeve to the shaft.

The coupling hubs of most modern centrifugal compressors are mounted hydraulically. The resulting interference fit allows the transmission of torque from the driver to the compressor without the use of keys or keyways. The hydraulic-fit method of coupling attachment involves a tapered shaft end, a matching taper in the coupling bore, and a method of

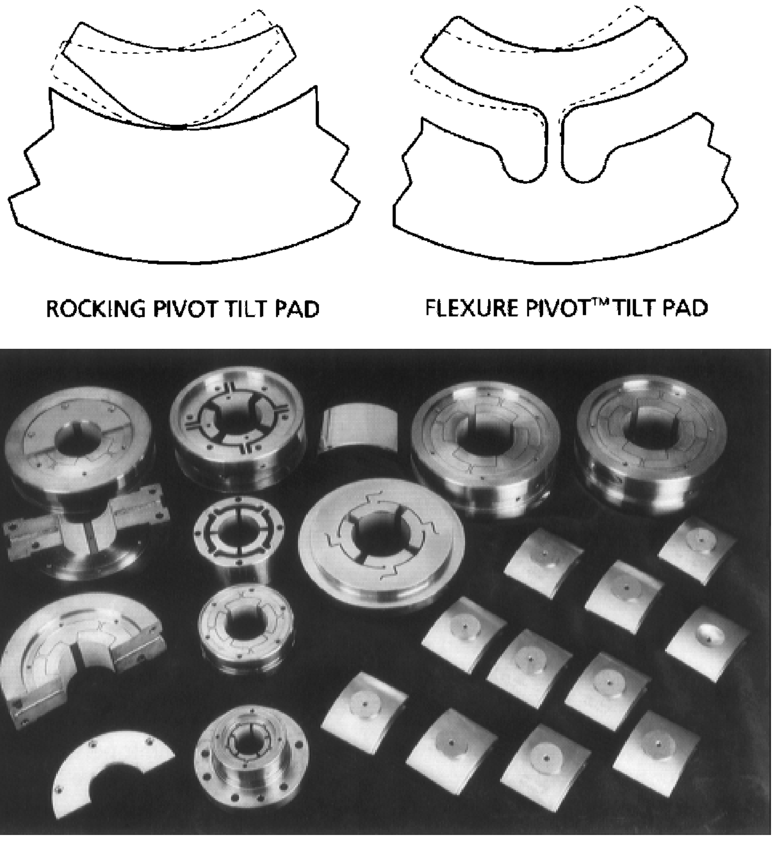


FIGURE 12.67 Radial Flexure Pivot tilt pad bearings. *(Bearings Plus, Inc., Houston, TX.)*

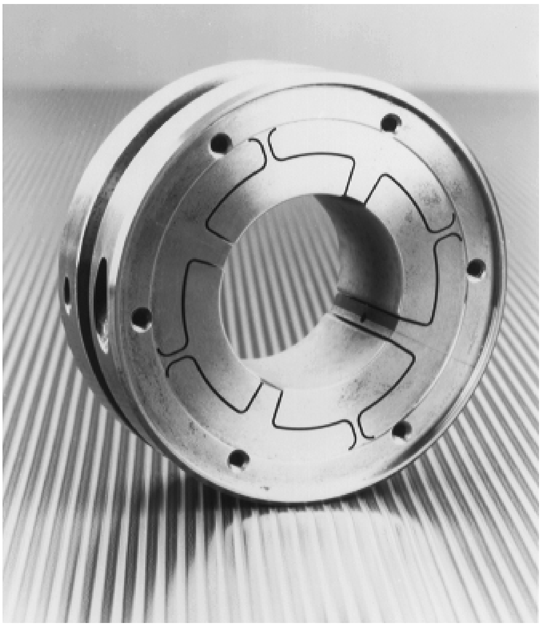


FIGURE 12.68 Radial Flexure Pivot tilt pad bearing, split design. *(Bearings Plus, Inc., Houston, TX.)*

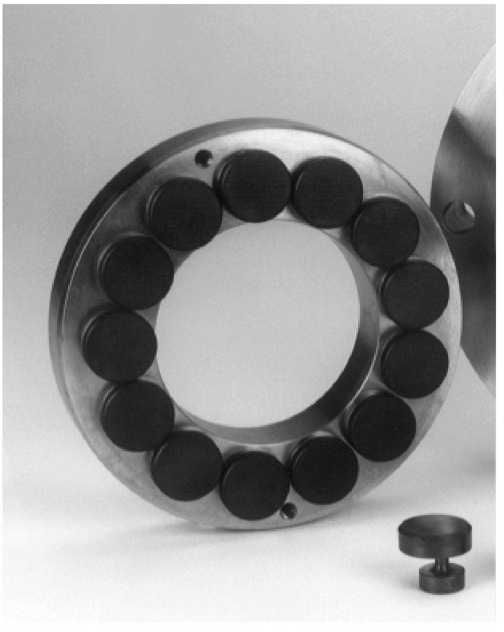


FIGURE 12.69 Thrust-type Flexure Pivot bearing for a small compressor. *(Bearings Plus, Inc., Houston, TX.)*

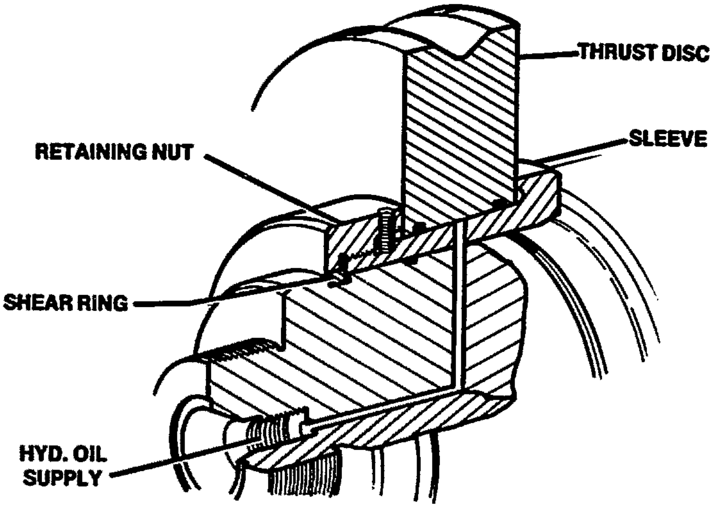


FIGURE 12.70 Hydraulically fitted thrust disk. *(Dresser-Rand Company, Olean, N.Y.)*

introducing high-pressure oil between the shaft and the coupling hub. A high-pressure oil fitting is provided in the end of the shaft. The coupling hub is installed on the end of the shaft, and high-pressure oil is introduced to expand the hub, which is then moved axially along a shaft taper to a measured distance in relation to the shaft end. Once the coupling is in place, oil pressure is released, and the coupling assumes a tight shrink fit on the shaft end (Fig. 12.72).

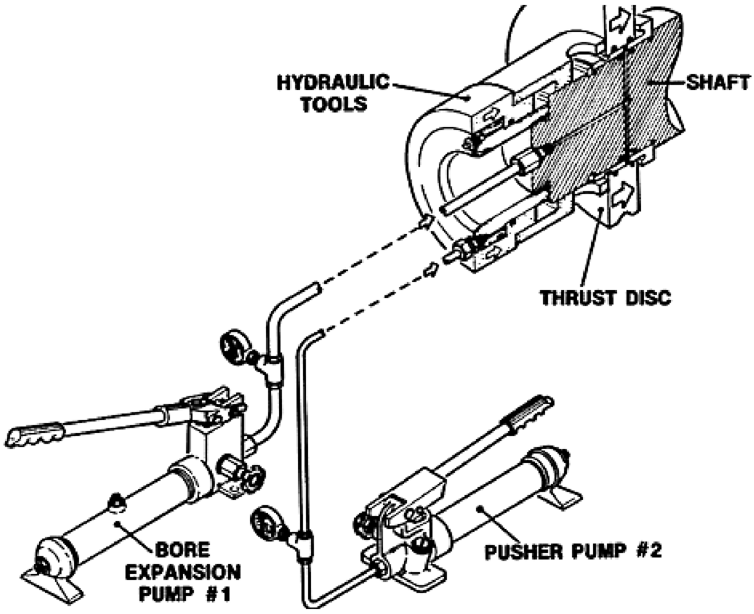


FIGURE 12.71 Hydraulic fit-up procedures. *(Dresser-Rand Company, Olean, N.Y.)*

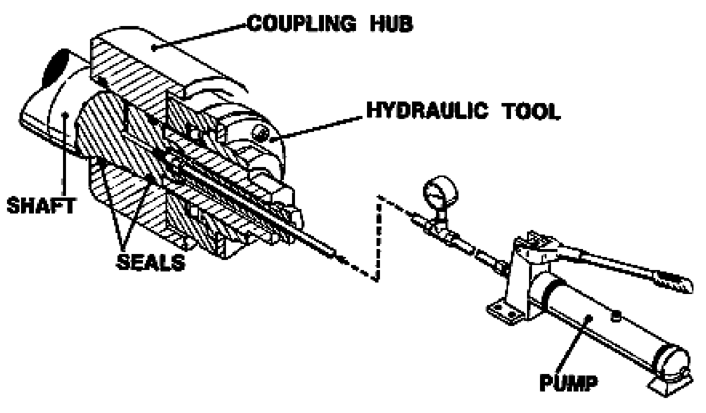


FIGURE 12.72 Hydraulically fitted coupling hub. *(Dresser-Rand Company, Olean, N.Y.)*

An alternative method of hydraulically mounting coupling hubs involves the injection of high-pressure hydraulic fluid through a threaded port machined into the coupling hub, 90° to the coupling or shaft axis. A hydraulic ram then pushes the expanded hub axially up the tapered shaft.

Centrifugal compressors require one or more major support systems to supply oil to lubricate the bearings, contact seals, and monitoring or control system. A typical lubricating system furnishes clean cool oil to the journal and thrust bearings. The lube system can be separate or combined with the seal oil system. This important subject is dealt with later in the book. Also, to properly monitor, control, and protect a complex and expensive piece of machinery such as a high-speed centrifugal compressor, a variety of systems are available.

A well-designed control system, including surge control, is a key element in the reliable operation of the centrifugal compressor. Again, this is discussed separately.

12.7 CASING DESIGN CRITERIA

The requirements for particular capacity and pressure capabilities of a centrifugal compressor design are determined by a manufacturer based on an analysis of the markets being served. Once a market need is determined, conceptual design for the equipment can begin. The casing receives much initial attention, as it represents the major pressure-containing structure.

The maximum-capacity requirement determines the areas required both inside the compressor casing and in the attached nozzles. The pressure requirement influences choices as to whether the casing (often called the *case*) can be horizontally or vertically split; whether it can be cast, formed from plate steel, or must be forged; what approximate thicknesses are required; and what form the nozzle connections must take. ASME formulas, from Section VIII, Division 1 of the ASME Code, can be used to rough out a minimum case thickness to start. API Standard 617 requires that all cases be designed such that hoop-stress values comply with ASME Code maximum allowable stress values.

As an example, a case for a large-capacity propylene refrigeration compressor was made from steel plate, ASTM A516, grade 65 (minimum tensile strength of 65,000 psi). The ASME Code permits a maximum allowable stress of one-fourth of the minimum tensile strength: thus, 16,250 psi. The casing inside diameter was sized at 112 in. for the large-volume flow. For a case rating of 350 psig, with a requirement for 100% radiograph inspection of the welds, the minimum case wall thickness is

$$
\begin{aligned}
t = \frac{PR}{SE - 0.6P} = \frac{(350)(56)}{(16,250)(1.0) - (0.6)(350)} &= 1.222 \text{ in.} \\
\text{corrosion allowance} \quad & \underline{+0.125 \text{ in.}} \\
\text{minimum allowable wall thickness} \quad & 1.347 \text{ in.}
\end{aligned}
$$

where t = thickness, in.
P = pressure, psig
R = radius, in.
S = allowable stress, psi
E = joint efficiency

It should be pointed out that the actual case wall thickness for the compressor above was 3.75 in., almost three times the minimum required. Compressor case thicknesses are usually quite conservative compared to code thickness requirements, since the primary design criterion is the mechanical rigidity of the case. Casing hoop-stress levels, based on a simplified formula of stress = $P \times R/t$, at maximum rated pressure, typically fall in the following ranges:

3,000–5,000 psi	horizontally split, cast iron
5,000–7,000 psi	horizontally split, cast steel
5,000–7,000 psi	horizontally split, fabricated steel plate
6,000–9,000 psi	vertically split, fabricated steel plate
10,000–16,000 psi	vertically split, forged steel

Account must be taken of the location and support of end closure heads and internal pieces, as required. Deflections at these positions must be held within predetermined limits. Consider, for example, an O-ring on the outside diameter of a vertically split case bundle. O-rings are commonly used between the bundle and case to prevent recycle of gas in a compressor section. The case growth under pressure must be considered at the O-ring land location to ensure that the O-ring is not extruded in operation. O-rings in the outside diameter of the heads at the case ends present the same problem. Neither would it be desirable to have a horizontally split case deflect to such a degree as to allow opening of stationary internal splits, again causing excessive recycle flow. Deformation at the case rail fits must be held to reasonable levels to control split gaps and interstage labyrinth seal clearances.

Horizontally split cases create an additional concern over the pressure capability of the split joint. Split flanges must be rigid enough to maintain their shape and proximity at pressures of 1½ the case rating, or hydrostatic pressure. This usually requires very thick flange sections, held together with numerous large highly stressed studs. Casing contours of horizontally split cast cases often contain bulges and indentations at nozzle and volute sections and around studs. A balance must be achieved between the aerodynamic requirements on the inside and the mechanical and maintenance requirements on the outside. This gives rise to difficult shapes and resulting stress concentrations that must be recognized and accommodated.

Typical materials used in the construction of horizontally split cases are listed in API 617 and include:

	Material	Application Range (°F)
Cast iron	ASTM A278	−50 to 450
Ductile iron	ASTM A395	−20 to 500
Cast steel	ASTM A216	−20 to 750
	ASTM A352	−175 to 650
Fabricated steel	ASTM A516	−50 to 650
	ASTM A203	−160 to 650

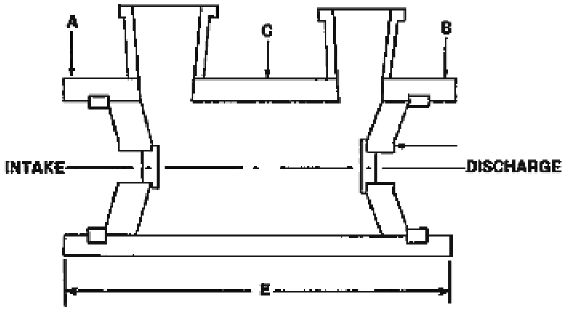
Lower-temperature services often use high-nickel-content materials such as ASTM A296.

Typical bolting materials for split flanges include ASTM A307 grade B for cast iron casings and ASTM A193 grade B7 for cast and fabricated steel casings. Nuts are typically supplied per ASTM A194 grade 2H. For low temperatures, ASTM A320 bolting is usually supplied.

The most difficult area of design for a vertically split case is usually the overhang outboard of the head shear ring. The pressure inside the case tends to swell the case in the middle, causing the overhung ends to deflect inward. At the same time, the pressure on the head enclosure results in a force on the shear ring that tries to deflect the overhung end outward. The stress concentration effect of the angular shear-ring groove aggravates the condition. These considerations usually make necessary a finite element analysis of the area before final design details are established.

The casing heads are also typically subjected to a finite element analysis; it is obviously important to control deflections. Efforts to apply ASME formulas intended for simplified flat head design can lead to erroneous results because these formulas are just not applicable to typical compressor head shapes.

A manufacturer has to have confidence that casing and head deflections can be accurately predicted; the manufacturer must be certain that the compressor design will perform



LOCATION	ACTUAL	PREDICTED
A RADIAL	.007	.005
B RADIAL	.007	.005
C RADIAL	.007	.007
D AXIAL	.031	.028
E AXIAL	.008	.012

FIGURE 12.73 Dial indicators measure key areas of a compressor on the manufacturing floor. *(Dresser-Rand Company, Olean, N.Y.)*

as intended. Excessive or unexpected movements can distort the case, cause tilting of a head that can lead to excessive runout between the thrust bearing and thrust disk, and cause gas leaks both internal to the machine (recycle) and/or through a case seal. This concern leads to strain gauge and dial indicator measurements of various cases and heads during pressure testing. These data are compared with predictions in a continuing effort to update and improve the analytical tools. As an example, Fig. 12.73 shows the location of dial indicators on key areas of a medium-sized propane compressor and compares predicted (based on finite element analysis) and actual deformations, thus serving to verify the analysis.

Typical materials for vertically split casings include:

	Material	Application Range (°F)
Welded	ASTM A516	−50 to 650
	ASTM A203	−160 to 650
Forged	ASTM A266	−20 to 650

Again, lower temperatures are possible with high-nickel-content materials. Pressure containing heads are designed from:

	Material	Application Range (°F)
Cast steel	ASTM A216	−20 to 750
	ASTM A352	−175 to 650
Plate	ASTM A516	−50 to 650
Forged	ASTM A266	−20 to 650
	ASTM A350	−150 to 650

Shear and retainer rings are usually from similar material.

A generalized pressure and flow chart is shown in Fig. 12.74. Comparison with Fig. 12.3 will demonstrate that specific capacity and pressure steps vary from manufacturer to manufacturer. The pressure capability of horizontally split designs is limited to approximately 1000 psi because of the practical problems of sealing a split at higher pressure levels with associated increased distortion.

High-pressure compressors may be defined as those operating over 1000 psi, and these fall in the vertically split category. Equipment of this type is used in applications such as ammonia and methanol synthesis gas, CO_2 for urea, as well as natural gas storage, gas lift, and reinjection. Figure 12.75 shows how these ratings have grown over the years as the market changed. Reinjection now accounts for the highest-pressure installations in the world. To

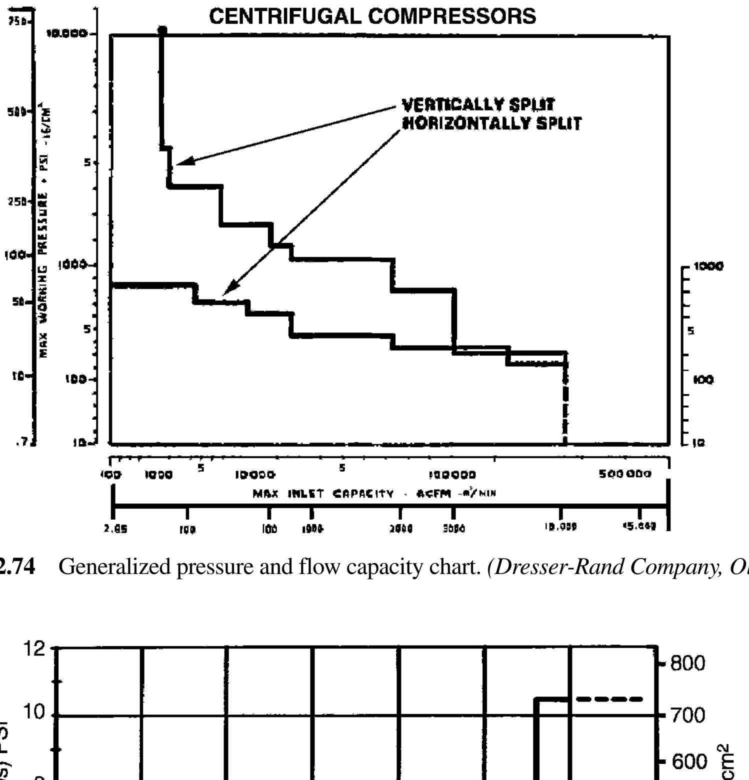


FIGURE 12.74 Generalized pressure and flow capacity chart. *(Dresser-Rand Company, Olean, N.Y.)*

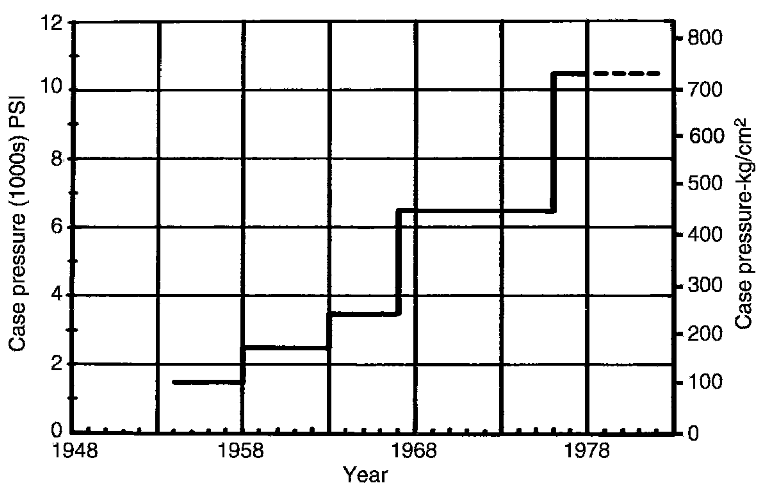


FIGURE 12.75 Pressure growth chart. *(Dresser-Rand Company, Olean, N.Y.)*

date, centrifugal compressors have been proven at 10,500 psi, and even this can be extended when the design is needed.

Figure 12.13 illustrates a typical high-pressure compressor. This particular one has eight impeller stages in a back-to-back arrangement. This design provides essentially a balanced thrust force condition that under most circumstances eliminates the need for a balance piston. The casing and heads are steel forgings, and the heads are retained by shear rings. Process pipe connections at inlet and discharge connections are made by machining flat areas on the casing and attaching flanges by means of stud bolts mounted in the case.

As discussed in Section 12.5, the choice between a straight-through design and a back-to-back arrangement for high-pressure applications is one of design philosophy. The back-to-back design is subject to the adverse effect of section mismatching. High-thrust loads can occur during operation at excessively high flow, or *stonewall*, because of loss of the second-section pressure ratio. But this is an off-design operating condition problem that can be prevented by controls or proper operation. Compressor thrust bearing failures caused by balance piston seal deterioration in high-pressure, straight-through machines point to a design or operating condition problem; these are more difficult to monitor and prevent.

The advantages of a back-to-back arrangement in a high-pressure case include:

- The aforementioned elimination of potential thrust bearing failure due to failure of the large-diameter balance piston labyrinth. (Balance piston labyrinths in straight-through designs are required to withstand differential pressures as high as 5000 psi.)
- Reduction of recirculation losses in the compressor since:
  a. The pressure led to the seals and balanced to the compressor suction is an intermediate pressure, as opposed to full discharge pressure on straight-through flow designs.
  b. Seals balanced to suction can have much smaller diameters (and therefore a much smaller flow clearance area) than those of balance piston labyrinths on straight-through flow designs.
  c. If the low-pressure section feeds contaminated gas into a scrubber or other purifier and the *cleaned* gas subsequently enters the higher-pressure section, internal gas leakage will go from the clean section to the contaminated section: The contaminated gas cannot leak into the clean gas.

High-capacity compressors may be defined as those operating at over 35,000 cfm inlet capacity. Two major petrochemical plants that use these large-capacity machines are ethylene plants and liquefied natural gas production facilities.

Figure 12.76 shows a typical large refrigeration compressor. These compressors handle gases such as propylene and propane. Fairly stringent requirements influence the selection of equipment for this service. Cases are usually designed with materials suitable for −40 to −50°F.

The ethylene compressor is still relatively small, even for the largest ethylene plants. It is usually handled in a medium-sized case. The gas is sweet and clean and the pressures are moderate. A key design consideration is the selection of materials for very low temperature, approximately −150°F.

Traditionally, for pressure ranges that offer a choice, the market has preferred the horizontally split machine from the viewpoint of maintenance. Most often, with steam turbine or electric motor drivers, the compressor and driver will be mounted on a mezzanine. Nozzles are directed downward, and the upper half of the case can be removed relatively easily, exposing all internals. The horizontally split case is still the preferred choice for

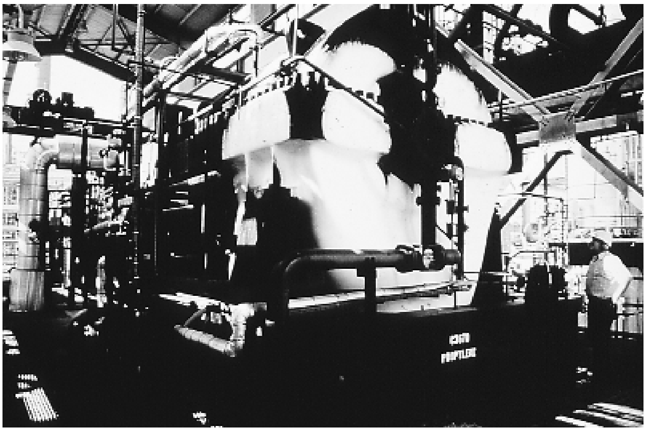


FIGURE 12.76 Large refrigeration compressor. *(Dresser-Rand Company, Olean, N.Y.)*

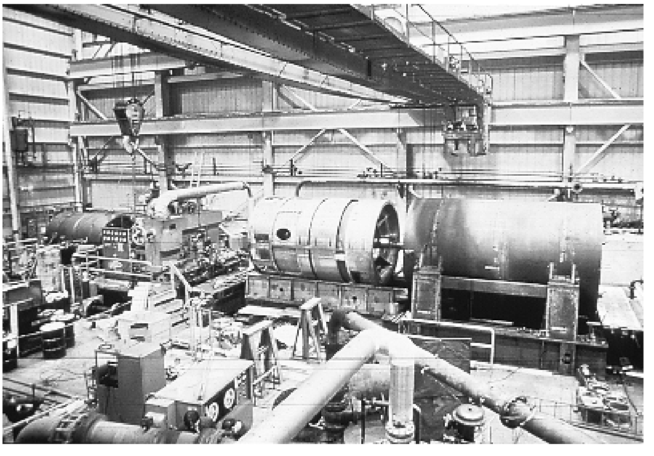


FIGURE 12.77 Assembly and maintenance procedure for a vertically (radially) split compressor casing. *(Dresser-Rand Company, Olean, N.Y.)*

through-drive applications. Here, room does not normally exist at either end of the case to extract a bundle. In recent years, there has been a trend toward vertically split compressors (Fig. II.4) for high-capacity applications. This is especially true with gas turbine drivers, where the compressors and gas turbines are located at grade level. The gas connections for the compressors are then located at the top of the case. If horizontally split units were selected, it would be necessary to dismantle the gas piping before removing the upper half. This is not necessary with a vertically split case because the internals are withdrawn axially from the end of the case (Fig. 12.77).

When lifting capabilities are limited and crane capacity does not allow removal of the upper casing half, vertically split compressors are also preferred. After the internal assembly is removed from the casing, the upper half of the bundle assembly can be dismantled by removing throughbolts and removing one section at a time (Fig. 12.78), thus dramatically reducing crane requirements. Alternatively, the bundle top half can be removed in a single lift (Fig. 12.79).

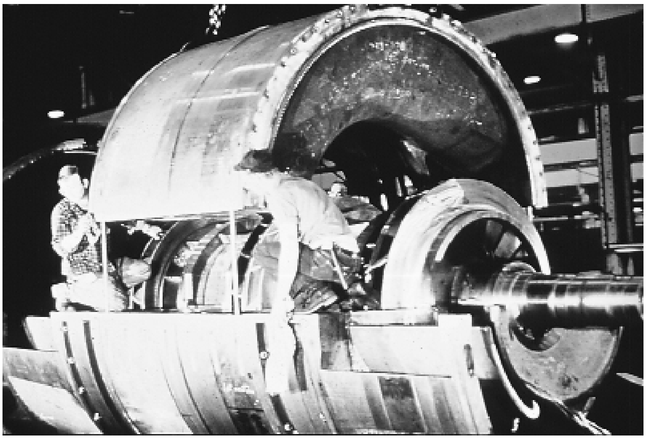


FIGURE 12.78 Vertically split compressor being dismantled. *(Dresser-Rand Company, Olean, N.Y.)*

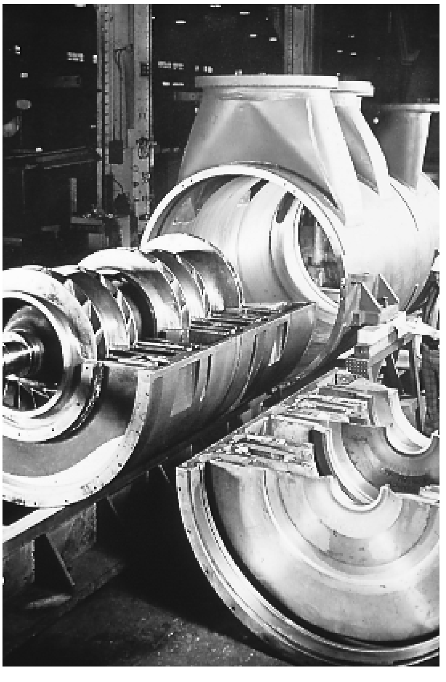


FIGURE 12.79 Removal of the bundle top half of a vertically split compressor. *(Dresser-Rand Company, Olean, N.Y.)*

The typical vertically split casing design incorporates many features to ease the installation and removal of the internal assembly. The case internal diameter is machined (Fig. 12.80) in a series of steps, the largest diameter being at the maintenance end. It is designed so that when the bundle is installed, it is only in the last inch or two of travel that the bundle rises onto its locating fits and O-ring lands. Ramps are machined into the case for guiding these pieces. For very corrosive services, stainless steel lands are welded into the case inside diameter at all locations of contact between bundle and case. The intent here is to reduce corrosion and to facilitate disassembly.

For medium-sized and large compressors, adjustable rollers (Fig. 12.81) are designed into the bundle outside diameter at several positions along its length to permit easier travel along the case and support cradle. For down-connected cases, with large holes in the bottom of the case, ribs are provided across the nozzle openings in the line of roller travel. This design feature prevents the bundle rollers from losing contact.

Cast compressor casings are usually provided with integral feet, two at the suction end and either one centered at the discharge end or two located at the sides (Fig. 12.81). The feet must

FIGURE 12.80 Compressor casing bore. *(Dresser-Rand Company, Olean, N.Y.)*

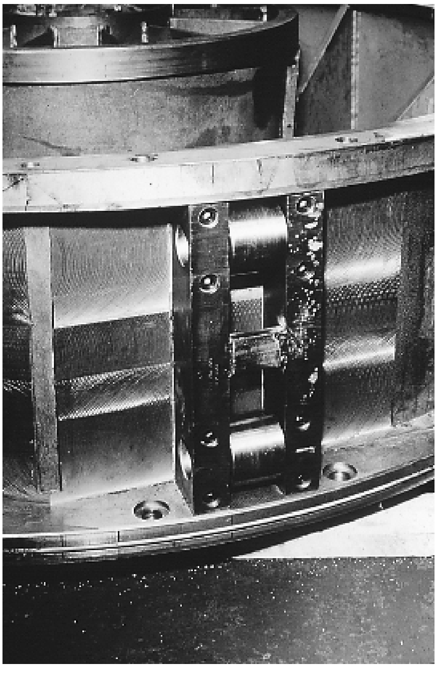


FIGURE 12.81 Adjustable rollers facilitate bundle installation. *(Dresser-Rand Company, Olean, N.Y.)*

hold the case firmly, yet permit thermal expansion. The case is typically doweled at the suction end feet, which, incidentally, is the end of the case where the thrust bearing is usually located. This allows case and rotor to grow thermally and expand in the same direction. If the discharge end is supported solely in the middle, the support may take the form of a *wobble plate* (see Fig. 12.97). These plates are kept thin to permit deflection without excessive stress. As supported weights increase, additional thin plates are added, parallel to the first. When the weight capacity of this design is exceeded, it is customary to provide two discharge end supports, one at each side of the case, called *sliding feet.* These feet typically rest on a PTFE (Teflon) pad with clearance provided around the bolting to permit travel in the axial direction.

The dowels and bolting anchoring the case are designed to withstand all reasonable piping forces and moments and torque requirements. Case and nozzle thicknesses are rarely affected by piping loads. On U.S.-built compressors, these sections are very conservatively stressed, as shown previously.

12.8 CASING MANUFACTURING TECHNIQUES

The typical line of cast cases being offered by world-scale manufacturers includes cast iron and cast steel designs to satisfy market needs. The choice of materials for a given application is a function of several variables, including pressure and gas characteristics, as pointed out in API 617. Cast iron and cast steel each require their own supply of patterns, because shrinkage rates vary significantly between the materials. The decision to offer a complete line of machines can prove expensive for a manufacturer because of the quantity and complexity of patterns that must be built and maintained. Case variations are numerous, such as cast iron versus cast steel, standard-flow models versus high-volume reduction stepped cases, straight-through and double-flow configurations, nozzle connections up or down, various sizes of sidestream connections up or down connected at various axial locations in the case, and different-sized compound connections with different orientations. All of these special items require different patterns or pattern sections.

The majority of cast cases are split horizontally as shown in Fig. 12.82. Bearing chambers may be cast integral with the case; the case would provide integral bearing supports.

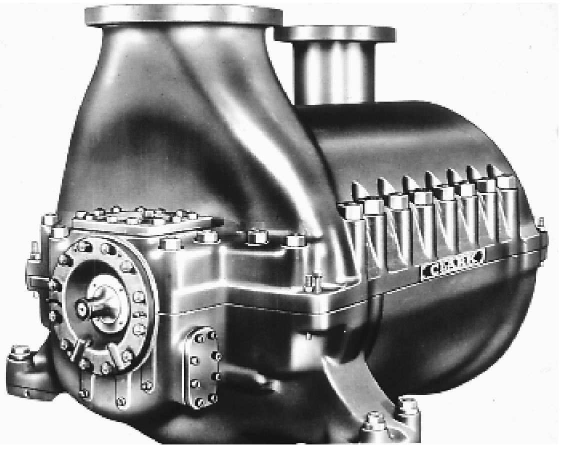


FIGURE 12.82 Centrifugal compressor with a cast casing. *(Dresser-Rand Company, Olean, N.Y.)*

This design ensures permanent built-in alignment, provided that the case is machined in one piece. Other arrangements involve bolted-on bearings.

Once the case is released from the foundry, the typical manufacturing sequence requires rough machining of the split surfaces to within a fraction of an inch of finish. Next, the sequence progresses to layout and rough machining of the inside contours and fits, welding on the case drains (cast steel only), complete stress relief, finish machining of the splits, and finish boring of the internal diameters. Return bends are generally machined integral with the case. Split-line bolting may be somewhat indented between the return bends to minimize the distance off the centerline.

Casing integrity is verified by hydrostatically testing to 1.5 times the casing design pressure. Casing splits are a metal-to-metal fit-up and generally sealed with the aid of room-temperature vulcanizing joint compound. Split-line bolting, stressed to prescribed values, pulls the casing halves together.

Fabricated case construction, an example of which is shown in Fig. 12.83, is used on the larger-capacity medium- and low-pressure vertically split and horizontally split compressors. Advanced manufacturing and welding techniques, which can vary somewhat from manufacturer to manufacturer, are used in the production of these cases. Steel plate and forgings are used.

The nozzles may be cast or may be formed from plate. The fabricated design is such that the transition from a rectangular shape (for the connection to the case) to a cylindrical shape is done by straight-line brake bending (Fig. 12.84). Very large nozzles are made from four separate pieces which are welded together (Fig. 12.85). The concept of straight-line seams is used to simplify the joining, allowing automatic submerged arc welding (Fig. 12.86). Forged steel flanges are welded to these nozzle bodies (Fig. 12.87) to complete the assembly. Again, this is done by the automatic submerged arc welding method.

The case cylinders (Fig. 12.88) are rolled from steel plate and welded along the longitudinal axis. The nozzles are then positioned and welded in place (Fig. 12.89). For long or stepped cases, two cylinders, one for the inlet end and one for the discharge end, are joined with a girth weld. To make this weld, the case is rotated on a special machine, and the submerged arc welding fixture is held stationary (Fig. 12.90).

FIGURE 12.83 Centrifugal compressor with a fabricated casing. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.84 Straight-line brake bending produces a cylindrical casing contour. *(Dresser-Rand Company, Olean, N.Y.)*

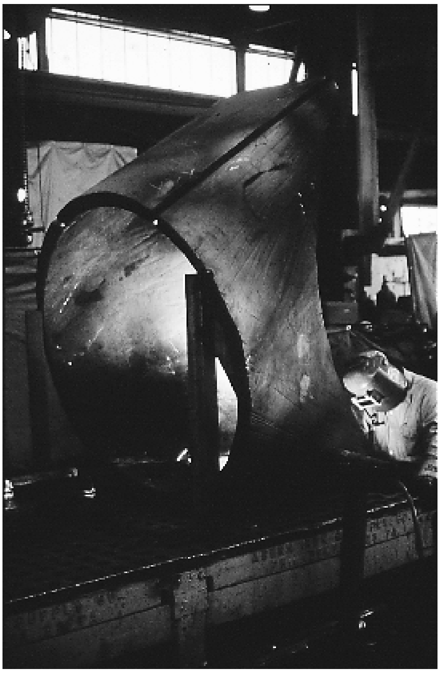


FIGURE 12.85 Nozzle welding in progress. *(Dresser-Rand Company, Olean, N.Y.)*

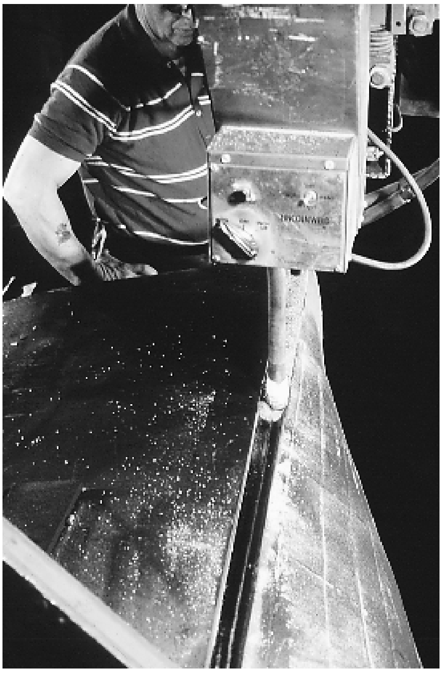


FIGURE 12.86 Automatic submerged arc welding. *(Dresser-Rand Company, Olean, N.Y.)*

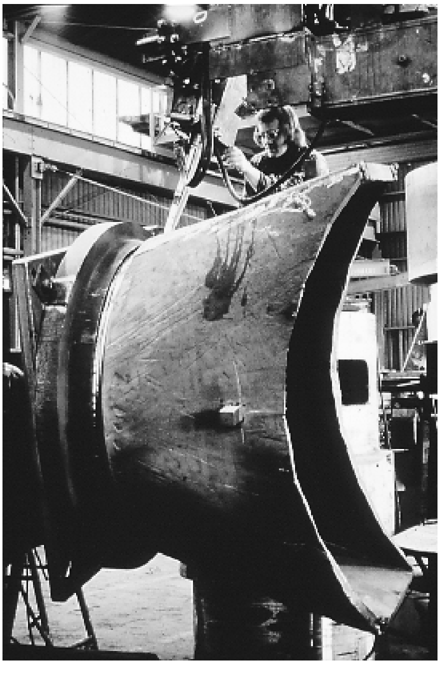


FIGURE 12.87 Forged steel flanges being welded to nozzle bodies. *(Dresser-Rand Company, Olean, N.Y.)*

FIGURE 12.88 Casing cylinder. *(Dresser-Rand Company, Olean, N.Y.)*

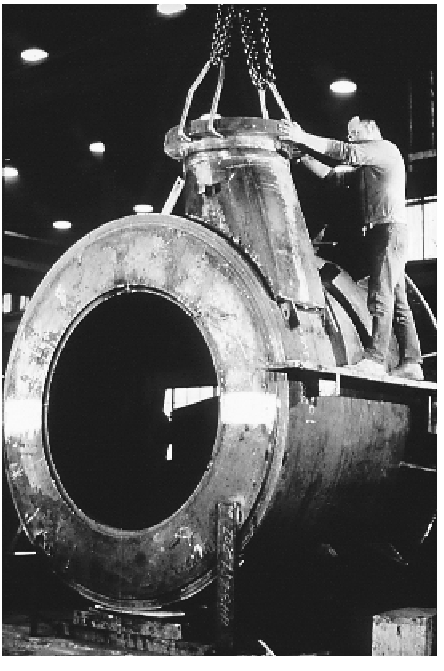


FIGURE 12.89 Nozzle positioning and welding. *(Dresser-Rand Company, Olean, N.Y.)*

The procedure thus far is the same for either a horizontally or vertically split case. If the case is split horizontally, the next step is to split the case longitudinally into two halves (Fig. 12.91). A split flange is burned out of thick plate (Fig. 12.92) and welded to each half. Return bend sections may be formed from steel plate and attached to the case by continuous structural welding to create a rib effect that provides additional stiffness to the case (Fig. 12.93).

The casings, assembled as described, are then rough-machined, heat-treated for stress relief (Fig. 12.94), and finish-machined (Fig. 12.95). Figure 12.96 shows a completed horizontally split fabricated compressor with the top half removed. Note the variation in impeller stage spacing. The large spaces allow for large-capacity sidestream entries.

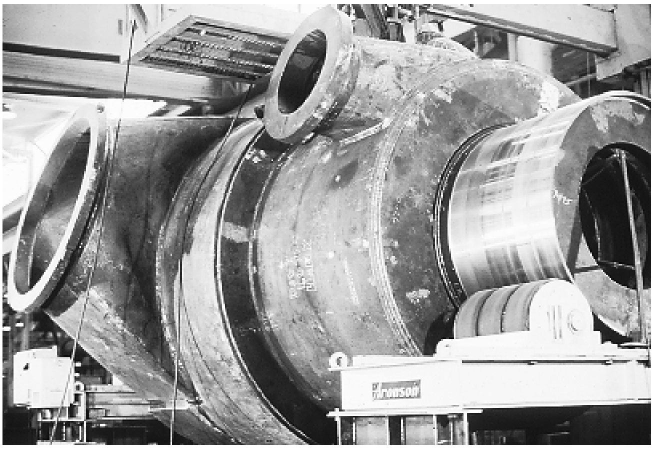


FIGURE 12.90 Case rotation fixturing and girth welding. *(Dresser-Rand Company, Olean, N.Y.)*

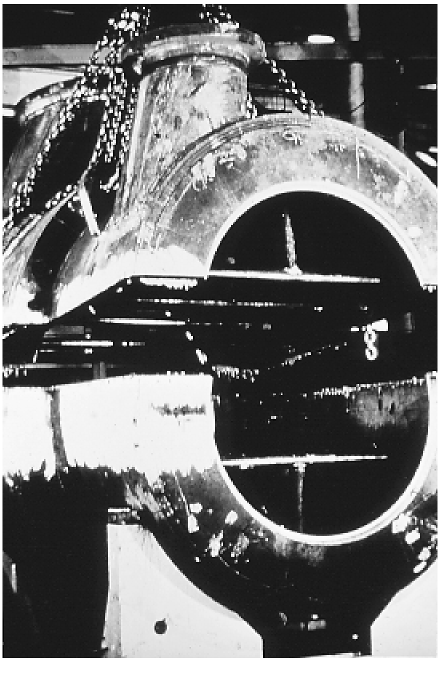


FIGURE 12.91 Longitudinal splitting of a compressor casing. *(Dresser-Rand Company, Olean, N.Y.)*

Figure 12.97 shows a completed compressor.

During manufacturing, the fabricated steel cases undergo quality assurance inspections such as:

1. All pressure welds are subject to 100% magnetic particle inspection of:
   a. Plate edges prior to welding
   b. Weld root pass
   c. Finish welds before and after stress relief

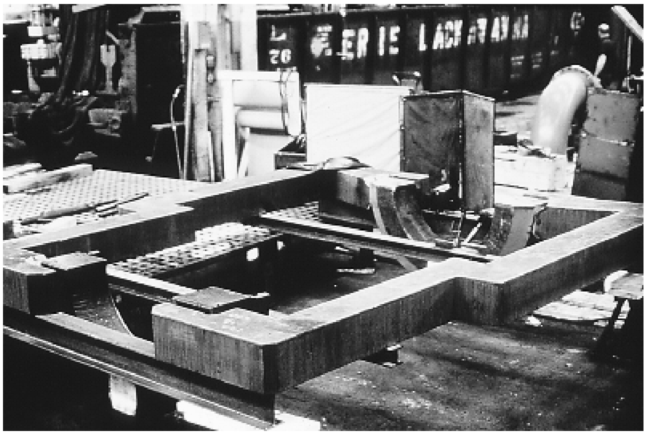


FIGURE 12.92 Flange production from a thick plate. *(Dresser-Rand Company, Olean, N.Y.)*

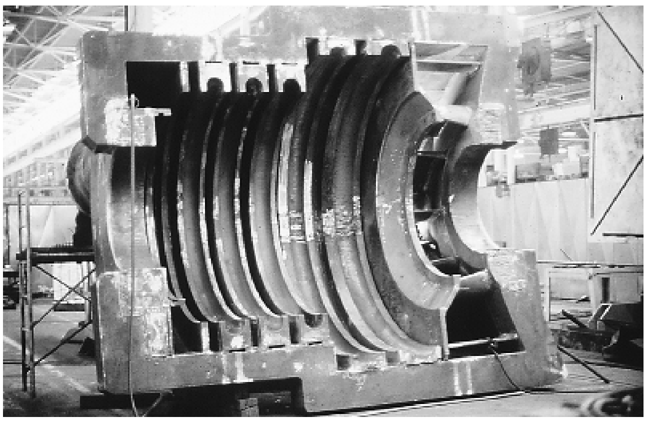


FIGURE 12.93 Attaching a return bend section to a compressor casing. *(Dresser-Rand Company, Olean, N.Y.)*

2. Structural welds are subject to 100% magnetic particle inspection of:
   a. Internal parts before stress relief
   b. External parts before and after stress relief
3. For design temperatures below −20°F (−29°C), impact tests of base material and weld metal are taken as part of the weld procedure qualification.
4. Hydrotest is done to 1.5 times the maximum working pressure.

Small high-pressure vertically split cases are often manufactured from forgings. This becomes a requirement when thicknesses become too great to roll from plate. Some of these cases are 10 to 12 in. thick to contain the high pressures. Nozzles are sometimes cast separately and welded to the case, but for very high pressure applications the process pipe connections are made by machining flat areas on the casing and installing studs for flanges.

Typical manufacturing techniques follow those established for the other casing types (i.e., rough machine, stress relief, and finish machine). Inspection procedures closely follow those for welded steel cases.

FIGURE 12.94 Heat treatment of a welded casing. *(Dresser-Rand Company, Olean, N.Y.)*

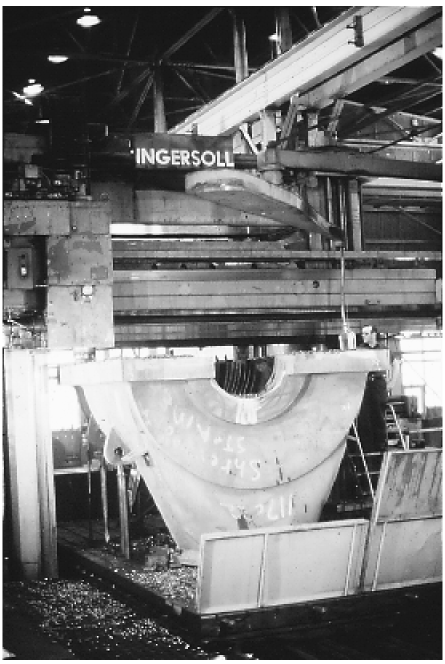


FIGURE 12.95 Finish machining in progress. *(Dresser-Rand Company, Olean, N.Y.)*

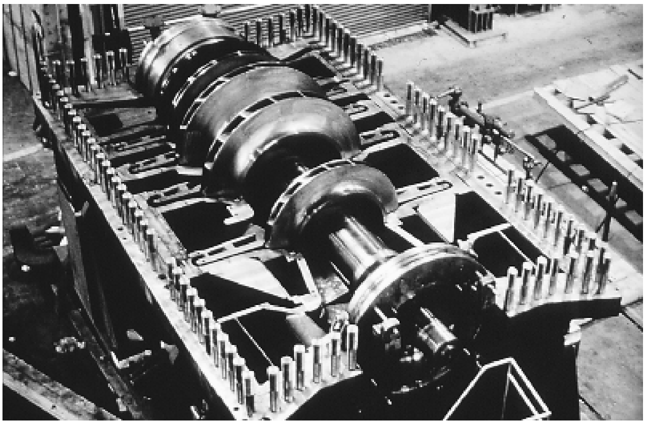


FIGURE 12.96 Horizontally split fabricated compressor, lower casing half. *(Dresser-Rand Company, Olean, N.Y.)*

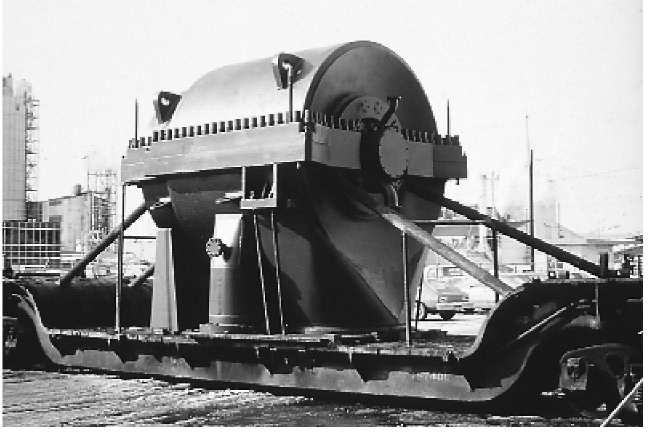


FIGURE 12.97 Completed fabricated compressor. *(Dresser-Rand Company, Olean, N.Y.)*

12.9 STAGE DESIGN CONSIDERATIONS

The most important element in a centrifugal compressor is the impeller. It is by way of the impeller that the work introduced at the compressor shaft end is transferred to the gas. The treatment the gas receives both before and after the impeller is also important. Typically, a guide vane directs the flow to the impeller, and a diffuser return bend and return channel (diaphragm) are downstream of the impeller (Fig. 12.98). Collectively, these parts make up a compressor *stage.* Some of the design considerations involved with these components are discussed in this section.

Individual impeller designs are related and evaluated by a parameter called *specific speed*, which classifies impellers based on similarity of design, such as angles and proportions. Although this was explained earlier in the book, a bit of amplification may be of interest to the reader.

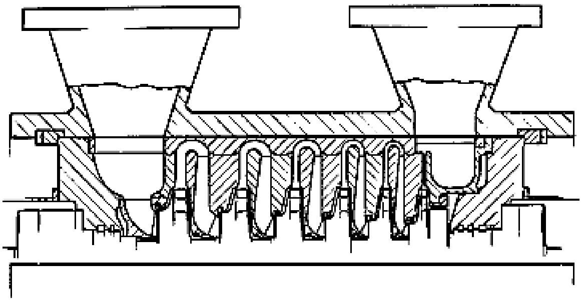


FIGURE 12.98 Stationary gas passages making up six stages. *(Dresser-Rand Company, Olean, N.Y.)*

Specific Speed One aspect of similarity is called *kinematic similarity,* where velocity ratios between two respective points within two separate designs are the same. For example, if a particular location velocity V as compared to the tip velocity U is constant between two different impellers and this is satisfied throughout the flow field, kinematic similarity is satisfied. Substituting for the velocity V, the capacity Q divided by the diameter D squared, and substituting for the tip speed U, the speed N times the diameter D, results in Q/ND^3, referred to as the *flow coefficient.* This is also referred to as a *capacity coefficient.* It represents the velocity similarities between two designs.

$$\frac{V}{U} = \frac{Q/D^2}{ND} = \frac{Q}{ND^3} = \text{constant} \Rightarrow \text{flow coefficient} \tag{12.1}$$

The second aspect of similarity is called *dynamic similarity.* Designs are considered to be similar if the forces and the pressure fields are proportional to each other. This similarity is expressed by the pressure coefficient, which is defined as head H divided by tip speed U squared.

$$\frac{H}{U^2} = \text{constant} \Rightarrow \text{pressure coefficient} \tag{12.2}$$

These two elements are combined to form specific speed. As can be seen, only operating parameters remain: speed, volume, and head.

$$\begin{aligned} N_S &= \int\left(\frac{Q}{ND^3}; \frac{H}{U^2}\right) \\ &= \int\left(\frac{Q}{ND^3}; \frac{H}{N^2D^2}\right) \\ &= \frac{\left(Q/ND^3\right)^{1/2}}{\left(H/N^2D^2\right)^{3/4}} \\ &= \frac{N(Q)^{1/2}}{H^{3/4}} \end{aligned} \tag{12.3}$$

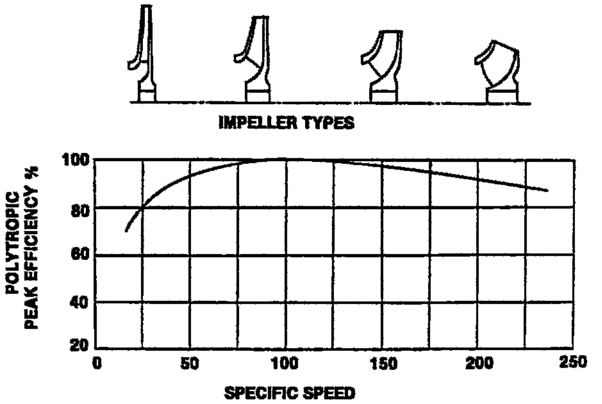


FIGURE 12.99 Efficiency and specific speed correlation based on shop tests. *(Dresser-Rand Company, Olean, N.Y.)*

where N_S = specific speed
Q = actual flow, actual ft^3/s
N = speed, rpm
H = head, ft-lb/lb
D = diameter, in.
U = tip speed, ft/min

The specific speed can be used in two ways. First, in application of equipment, it points out where the impeller is operating relative to previous experience. The second important use is the correlation between stage efficiency and specific speed. Such a correlation exists, as shown in Fig. 12.99, based on test data. Also shown is the approximate shape of the impellers in their respective specific speed area.

For any particular compressor frame size, there must exist a range of available impellers to satisfy the flow and head requirements. They may typically cover a range of specific speeds from approximately 20 to over 200. Designs on the order of 200 to 300 have been appearing in recent years because of a trend in gas transmission services of decreased head requirements and increased capacity levels. As stated before, impeller geometry changes with specific speed. One can see the pattern of change in straight-through compressors of numerous stages.

It is worth looking at some general considerations, such as radial versus backward bent blading, blade entrance angles, and impeller blade shapes. Radial-bladed impellers produce the most head but exhibit a reduced range of operation with reduced stability compared to that achievable through backward leaning of the blades. It is for this reason that most centrifugal impellers have backward-leaning blades, generally at angles of approximately 40 to 50°. This inclination has been found to produce a good balance between head and operating range.

The design flow for a particular impeller is determined by the blade entrance angle at the eye; this is illustrated in Fig. 12.100. At the inlet, in the stationary coordinate system, the velocity magnitude and direction are represented by C_{M1}. In the relative system, rotating with the blade at its velocity, the flow approaches in the direction and magnitude of W_1. The design point of a stage represents optimum efficiency; it occurs when the vector W_1 is in

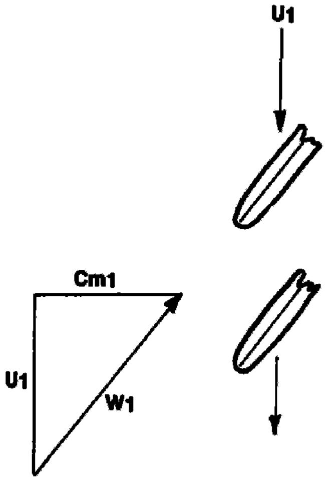


FIGURE 12.100 Impeller inlet diagram. *(Dresser-Rand Company, Olean, N.Y.)*

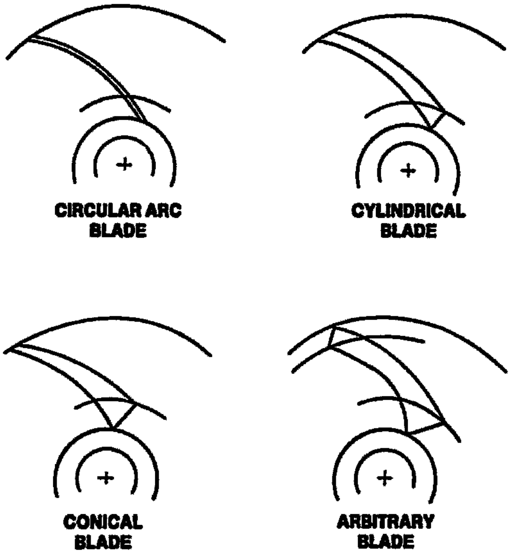


FIGURE 12.101 Impeller blade shape choices. *(Dresser-Rand Company, Olean, N.Y.)*

line with the blade angle. Since velocities across the impeller inlet eye can vary because of the turning of the flow, the blade angle must also vary across the eye to accommodate this requirement.

As the throughflow velocity C_{M1} increases or decreases, the efficiency drops, primarily because of the positive or negative angle of incidence at the leading edge. Selections are usually made within 5% of the design flow.

The level and shape of the impeller pressure coefficient curve is a function of velocity triangles at the discharge of the impeller. The more backward the blade is bent, the lower the pressure coefficient and the higher the rise to surge.

Several types of impeller blading can be chosen for a particular design requirement, and some are shown in Fig. 12.101. At top left is a circular arc blade that is oriented in the axial direction. At top right is a three-dimensional blade shape generated on the surface of an inclined cylinder. At bottom left, we depict a three-dimensional blade shape generated on a conical shape. Finally, at bottom right is a completely arbitrary blade shape. The complexity and manufacturing difficulty of these shapes increases in the order mentioned. The choice is made on the basis of satisfying the aerodynamic requirements. If more than one type of blade is satisfactory, the least complex design is selected.

Blade thicknesses are determined by a stress criterion, because the blades must hold the cover to the disk for the typical closed design at the frame speeds desired. Moreover, the blades have to be structurally sound.

The largest flow impellers for a particular frame size are of the open type, with no attached cover, as shown in Fig. 12.39. Radial blading and elimination of a cover on this particular design provide for a maximum amount of flow and head in one stage, yet keep stress levels sufficiently low, even in large diameters, to allow acceptance of maximum material yield point criteria as specified for H_2S service. Note that API 617 stipulates the use of 90,000 psi maximum yield strength, RC-22 hardness (235 BHN) steels. For ease of manufacturing, the minimum yield strength of carbon steels generally applied in H_2S service is 80,000 psi.

Open impeller blades can be backward curved if desired, with some sacrifice to head and special consideration to blade stresses. The typical open impeller has an almost axial inlet section, referred to as an inducer section. This impeller is typically used in the first stage and requires more axial space than the smaller designs. The primary advantages of this design are higher flow and pressure ratios, with some sacrifice to stability and efficiency. The impeller is designed to run with generous clearances (approximately $\frac{1}{8}$ in. for a 40-in.-diameter impeller) between its periphery and the stationary shroud. It is industry practice to provide one or two stages of backward-curved impellers behind this radial flow impeller to produce a rise to the surge performance characteristic in a given section.

A variation of the open impeller design is to equip it with a cover. Some gain is made in efficiency, but the attached cover increases the stresses and lowers the maximum speed capability significantly.

The typical welded-closed impeller used in the vast majority of applications is one of three designs: three-D welded, three-piece welded, or two-piece (milled-welded) construction. The three-D welded (three-dimensional) has a blade shape that is a portion of a rolled conical or cylindrical surface, or may be a combination of the two. An example is shown in Fig. 12.102. These blades are positioned on the disk or cover at a predetermined inclination and location. The blades then form a three-dimensional contour with respect to the cover or disk.

The choice of cone or cylinder size, location, and inclination to satisfy the aerodynamically required angles at the leading and trailing edges of the blade is difficult. In the past, it was not unusual for a design drafter to spend several weeks trying different combinations before arriving at a satisfactory geometry. Use of the computer has reduced this time to minutes, which allows the designing of more than one blade for review and detailed analysis.

The basic advantage of three-dimensional impellers is better performance, with higher efficiency. However, material, tooling, and welding requirements are higher than with other closed impellers. They also require more axial space in the machine.

The three-piece design (Fig. 12.103) is the next step down in complexity. This type of impeller also has three basic components: blades, cover, and disk, welded together. This construction allows more freedom for aerodynamic design than does the milled-welded impeller design that follows. The blades, however, do not form a true three-dimensional contour. Instead, they are rolled into a circular shape. The impeller requires less axial spacing than the three-D type discussed earlier and is less complex to manufacture.

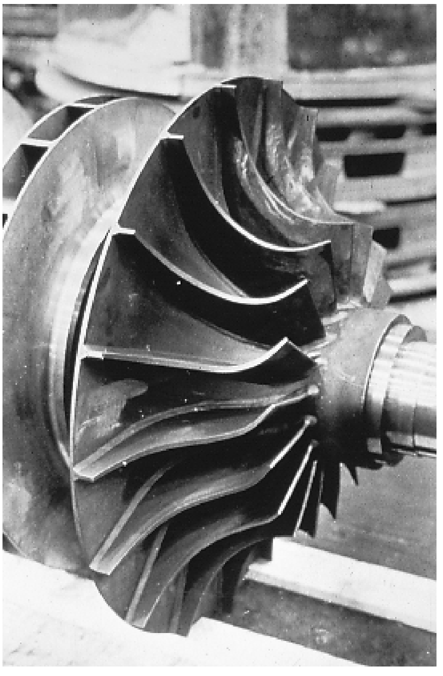


FIGURE 12.102 Three-dimensional, welded impeller. *(Dresser-Rand Company, Olean, N.Y.)*

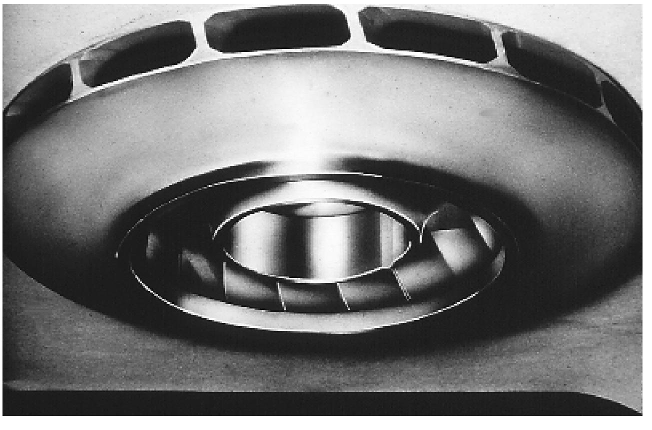


FIGURE 12.103 Three-piece impeller design. *(Dresser-Rand Company, Olean, N.Y.)*

The two-piece impeller (Fig. 12.104) has the blades milled onto the disk or cover and therefore requires welding on one side only. This type of impeller also has good efficiency and requires a minimum of stage spacing. The two-piece impeller therefore combines the features of good performance and ease of manufacturing. Figure 12.105 gives an idea of the range of impeller sizes available. The large rotor is from a large refrigeration machine, and the small rotor is typical of high-pressure services such as gas injection.

Once the flow channel through the impeller is set, the blade, disk, and cover contours and thicknesses must be developed consistent with the anticipated speed. Welded impellers are sufficiently complex to mandate computer-based stress and deflection analyses. A manufacturer's stress programs are typically backed up by extensive testing of prototype and/or

FIGURE 12.104 Two-piece impeller with integrally milled blades. *(Dresser-Rand Company, Olean, N.Y.)*

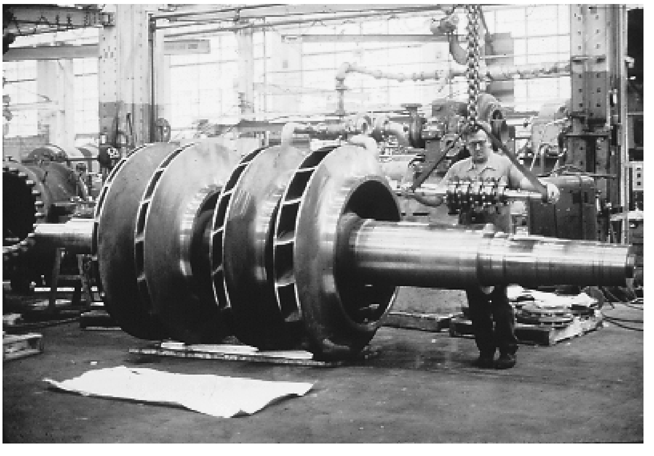


FIGURE 12.105 Range of impeller sizes typically available. *(Dresser-Rand Company, Olean, N.Y.)*

production impellers using strain gauges, proximity probe measurements of elastic deformations, stress coat and photo stress techniques for studying stress patterns, and overspeed-to-destruction tests.

From both the experimental and analytical investigations, the minimum strength requirements of welded impellers are determined, taking into account appropriate factors of safety. All impellers are heat treated per specifications developed through an engineering department, with acceptable physical property ranges targeted to cover yield and tensile strength, hardness, minimum elongation, and minimum reduction in area.

Typical strength levels used in impellers are as follows:

Material	Yield Strength Range (lb/in.^{2})
Alloy steel	80–145,000
403–410 stainless	70–135,000
17-4 PH stainless	125–170,000

We can now look at some considerations involved with the design of the other pieces that contribute to the stage, such as the guide vane, diffuser, return bend, and return channel, as depicted earlier in a six-stage compressor (Fig. 12.98). The efficiency of a given stage is a function of the friction and diffusion losses through the stage components. These loss mechanisms can be used to explain the shape of the efficiency versus specific speed curve (Fig. 12.99). Peak efficiency decreases as the specific speed is reduced or increased from an optimum range. This comes about as a result of the combined friction and diffusion losses reaching a minimum value.

Friction losses in a straight pipe are proportional to the velocity squared. The larger the pipe area, the lower the losses. Diffusion losses for areas that are changing are proportional to the velocity ratio $V_{\text{entrance}}/V_{\text{exit}}$ to the fourth power. Furthermore, losses are incurred by any bends and are proportional to the degree of turning and the tightness of the turn. The lesson in all this is that unnecessary diffusion and bending should be avoided in high-performance compressors.

Characteristically, friction losses increase at the lower specific speeds. This is due to the increased wetted surface and smaller hydraulic channel diameters. Diffusion, on the other hand, increases at the higher specific speeds, reflecting the effects of larger gas capacities being turned in tight bends. When the two losses are added, an area of minimum loss and maximum efficiency results, shown in Fig. 12.99.

The internal geometry of a compressor is more complicated than that of a bent or diffusing pipe. To calculate the velocities within a machine, a manufacturer may use advanced numerical solutions, whereby the geometry would be defined and the velocity fields that must satisfy radial equilibrium and continuity at every spatial point in the flow field would be calculated iteratively. Typically, for a compressor stage analysis, the velocity field from upstream of the impeller to upstream of the next stage impeller is analyzed.

To optimize the efficiency, the geometry of each component is reviewed in terms of velocity and velocity gradients. Unnecessary accelerations are eliminated and diffusion losses are minimized. The return bend radii are generous, the return channel contours are specially shaped, and the areas through the channel and guide vane are closely specified.

The velocity distributions for a final-stage geometry are shown in Fig. 12.106, from the trailing edge of one impeller to the leading edge of the subsequent impeller. The velocity

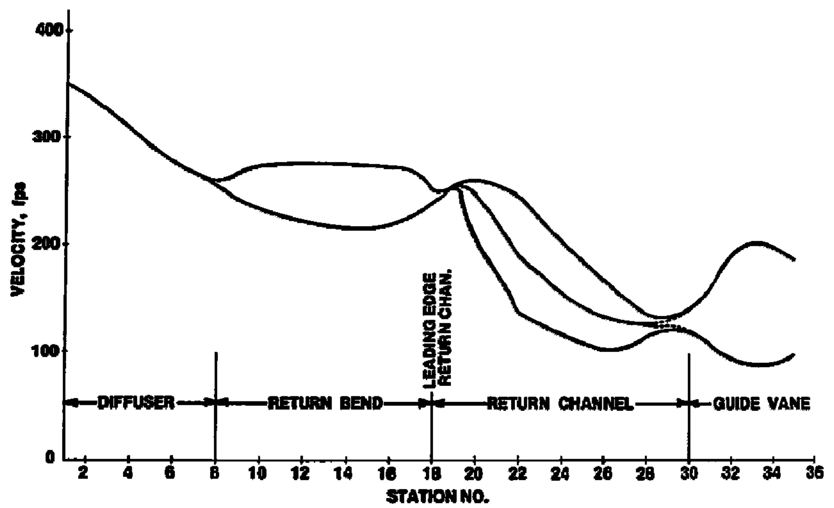


FIGURE 12.106 Velocity distributions for final-stage geometry. *(Dresser-Rand Company, Olean, N.Y.)*

decreases from the first impeller trailing edge through the vaneless diffuser in a uniform manner.

This is followed by the return bend, where the maximum velocities on the inner wall and the minimum velocities at the outer wall are proportional to the severity of the bend and the local surface curvatures. The diffusion losses on both surfaces have been minimized.

Following the return bend, the return channel velocities are shown. The mean velocity through the channel is represented by the middle line; the upper and lower velocities are those along the vane surfaces.

The return channel vanes are optimized. The vane shape, thickness distribution, and number of vanes are analyzed in conjunction with the channel axial height to arrive at the final coordinates.

Following the return channel is the guide vane, which turns the flow from radially inward flow to axial flow. This turning results in acceleration along the inner contour. Unnecessary accelerations and decelerations are carefully scrutinized and minimized.

Next is the impeller, which must be designed consistent with the approach velocities leaving the guide vane and, in turn, impart the energy increase to the fluid. The process repeats itself for as many stages as there are in the unit.

In summary, the velocity decreases in the diffuser downstream of the impeller, it is then turned from radially out to radially in, the tangential velocity or swirl is removed in the diaphragm or return channel, and finally, the flow is turned from the radial to the axial direction and enters the next impeller.

Inlet guide vanes, shown in Fig. 12.29, also provide one method of controlling stage performance, because they may be used to direct the flow into the impeller at different angles: against impeller rotation, radially, or with impeller rotation. The influence of various guide vane angles on a given impeller head characteristic is shown in Fig. 12.107.

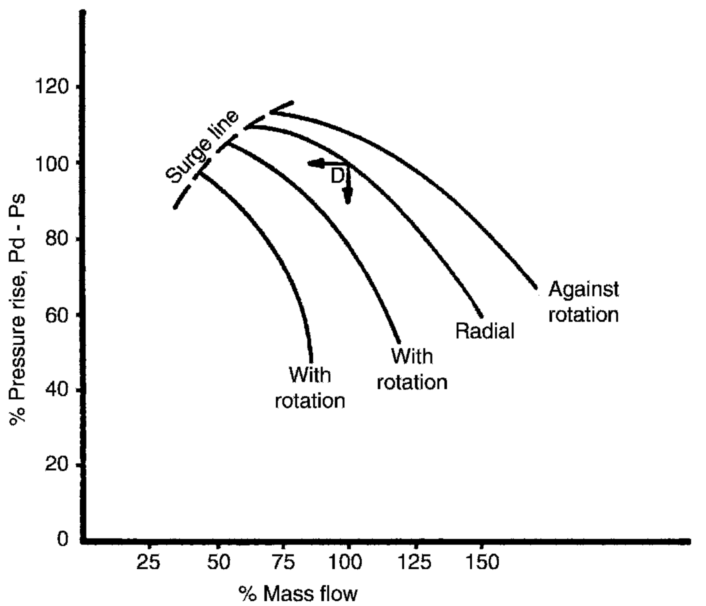


FIGURE 12.107 Guide vane angles vs. impeller head characteristics. *(Dresser-Rand Company, Olean, N.Y.)*

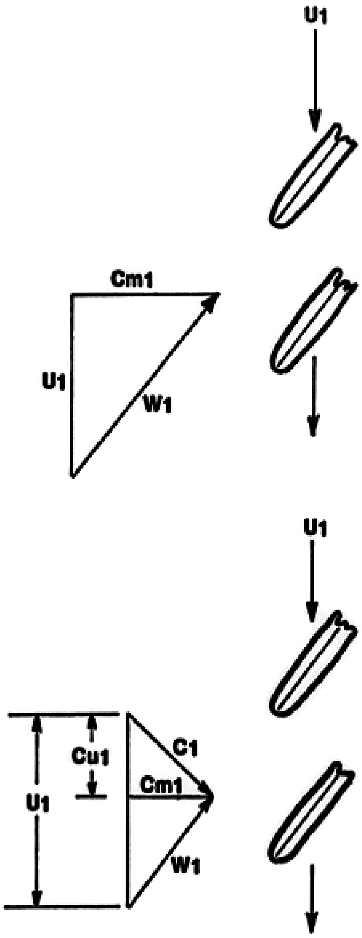


FIGURE 12.108 Impeller inlet diagram. *(Dresser-Rand Company, Olean, N.Y.)*

How this effect is caused is shown in the impeller inlet diagram (Fig. 12.108). When inlet tangential velocity or swirl exists because of turning guide vanes, the inlet triangle is modified. For a particular value of inlet tangential velocity, C_{U1}, there will be a through-flow velocity, C_{M1}, which will result in a good approach angle to the impeller blading. This is where the peak efficiency will occur.

12.10 IMPELLER MANUFACTURING TECHNIQUES

As mentioned in Section 12.5, the riveted impeller has been around for many years. There are two basic types of construction, one being two-piece with milled vanes; the other is three-piece with separate disk, cover, and blades.

Large-capacity impellers require three-piece construction (Fig. 12.103). The cover and disk are fabricated from forgings, usually of either alloy steel or a 400-series stainless. The blades are usually die-formed from stainless steel and are attached to the disk and cover by short stainless steel rivets extending through the disk or cover material and the flange provided along the blade by the forming.

FIGURE 12.109 Impeller undergoing semiautomatic welding. *(Dresser-Rand Company, Olean, N.Y.)*

Smaller-capacity impellers are of two-piece construction (Fig. 12.104), with vanes milled on either cover or disk piece. Disk and cover halves are jointed by long rivets that pass completely through the cover, the blade, and the disk. Flush riveting is achieved through the use of countersunk rivet holes. The outer surfaces are ground smooth.

For both types of construction, the pieces are typically purchased in the final heat-treated condition, with metallurgical and mechanical properties certified. They are completely machined and inspected, including a magnetic particle check, before joining. After assembly, the impeller typically undergoes balance, overspeed test, and additional final inspection, including magnetic particle and dimensional examination.

Cast impellers have also been used for some time. This technique is limited to designs that have high usage, thus justifying the typically high pattern costs. For this reason, cast impellers are not common in centrifugal process compressors, which are almost always custom designed for the application. The technique has been applied to some small impellers. However, there is obviously a practical limit to the narrowness of the flow channel.

Typically, the impellers can be made of any castable steel, including 400 series and 17-4 PH stainless steels. After the obligatory cleanup, they are heat-treated per the required specification. Bores and other surfaces are then machined as specified. Gas passage areas are essentially left untouched, aside from some possible hand-grinding or vibratory finish, if desired. Castings can be inspected or radiographed ultrasonically to verify soundness. The typical inspection process involves magnetic particle and dimensional checks before and after the balance and overspeed runs.

Welded impellers still account for the majority of impellers being designed and constructed. They are generally more rugged and will be more able to resist corrosion and erosion than riveted types. However, five-axis milling is making its mark and is often sought out for reasons of consistent quality and blade strength. Moreover, milled open impellers can be marginally more efficient than their various counterparts or competing configurations.

All materials for welded impellers, including the disks, blades, and covers, are supplied in the annealed condition, with appropriate material certifications. Pieces are preliminary-machined in preparation for welding. *Location welding*, or *tack welding*, is done by the inert gas-shielded metal-arc process (MIG). The completing welds are done by the same method used in a semiautomatic welding process (Fig. 12.109). The gas-shielded tungsten-arc (TIG) method is used at times for completing welds around blade ends.

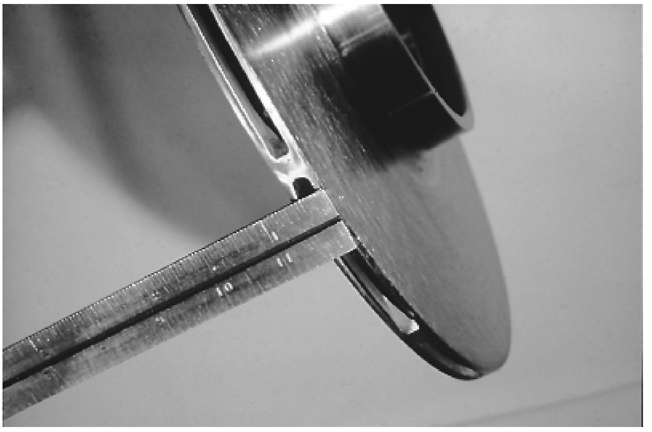


FIGURE 12.110 Narrow-width impeller. *(Dresser-Rand Company, Olean, N.Y.)*

Welding tables provide for proper preheat during the welding process. These tables are provided with heating elements in the table and cover to maintain material temperatures at approximately 600°F. The tables are mechanized to permit rotation of the workpiece under a stationary open flame, or torch.

Final mechanical properties are obtained by heat-treating (normalizing or quenching, with subsequent tempering) after welding. Typically, impellers then undergo final machining, dynamic balancing, and overspeed testing to at least 115% of maximum continuous speed, as specified in API 617.

The general manufacturing techniques used to produce the different types of welded impellers vary somewhat and will be looked at next. For a three-piece-construction impeller, the disk and cover are machined from forgings. The die-formed blades are then tack-welded to the cover with the use of locating fixtures. The final welding is a continuous fillet weld between the blade and cover. Subsequently, the blade cover assembly is joined to the disk by a continuous fillet weld between disk and blade.

An open impeller design consists of a disk and blades. The cover is eliminated. As discussed previously, this type of impeller is characterized by an inducer section that directs the gas flow into the eye of the impeller. The blades are either die formed or precision cast. The welding procedure is the same as for three-piece construction, with the final weld being a continuous fillet weld between the disk and blade. For two-piece construction, the blades are machined on either the disk or cover forging. The impeller is completed by a continuous fillet weld to the mating piece (disk or cover) around the entire blade interface.

The welding techniques described are limited to impellers with a flow channel width of more than $\frac{5}{8}$ in., to allow insertion of a torch. Increasing numbers of applications require the advantages of welded construction for impellers with channel widths of less than $\frac{5}{8}$ in. (Fig. 12.110). One method of doing this has been to weld through from the back side (Fig. 12.111). This type of impeller is manufactured from a disk forging and a cover forging, thus being of two-piece construction. Impeller blades, integral with the cover, are formed by removing metal from the inner face of the cover. A matching slot, corresponding to the blade contour, is machined in the disk at each blade location. After machining, the disk is located precisely over the cover with the blades aligned to the slots in the disk. The slot is then filled with a continuous multipass TIG weld (Fig. 12.112). An internal fillet is thus formed at the blade-to-disk junction. Using this technique, welded impellers with the

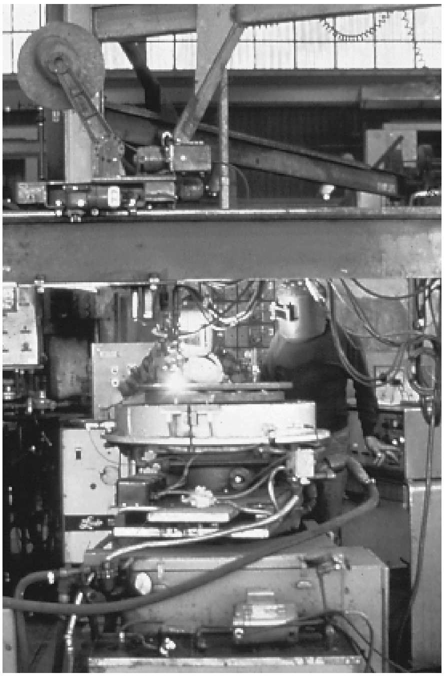


FIGURE 12.111 Slot welding a two-piece impeller. *(Dresser-Rand Company, Olean, N.Y.)*

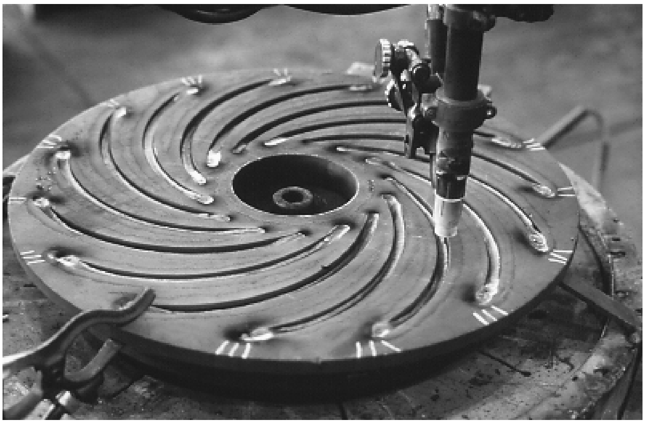


FIGURE 12.112 Slots in a disk being filled. *(Dresser-Rand Company, Olean, N.Y.)*

smallest practical channel width can be manufactured. Any blade contour may be designed without affecting the weldability of the impeller, thus minimizing compromises in aerodynamic design. The impellers (Fig. 12.113) are available in the same materials and range of properties as the fillet-welded impellers.

To ensure quality, the following general inspection procedures are typically carried out on welded impellers:

- Material certification reports are reviewed for compliance with specifications.
- Test bars for each forging are serialized and follow the impeller through all heat-treatment cycles.

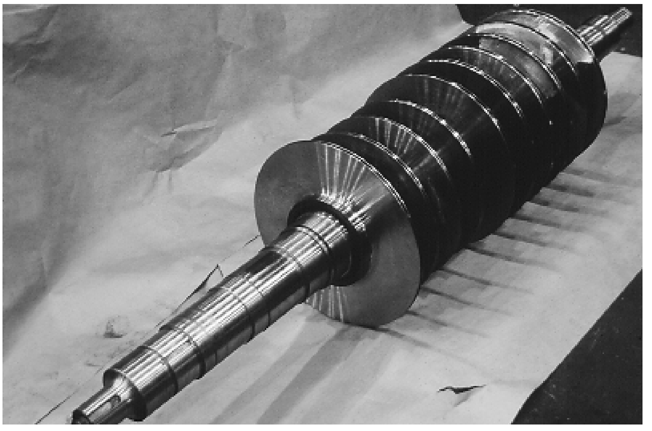


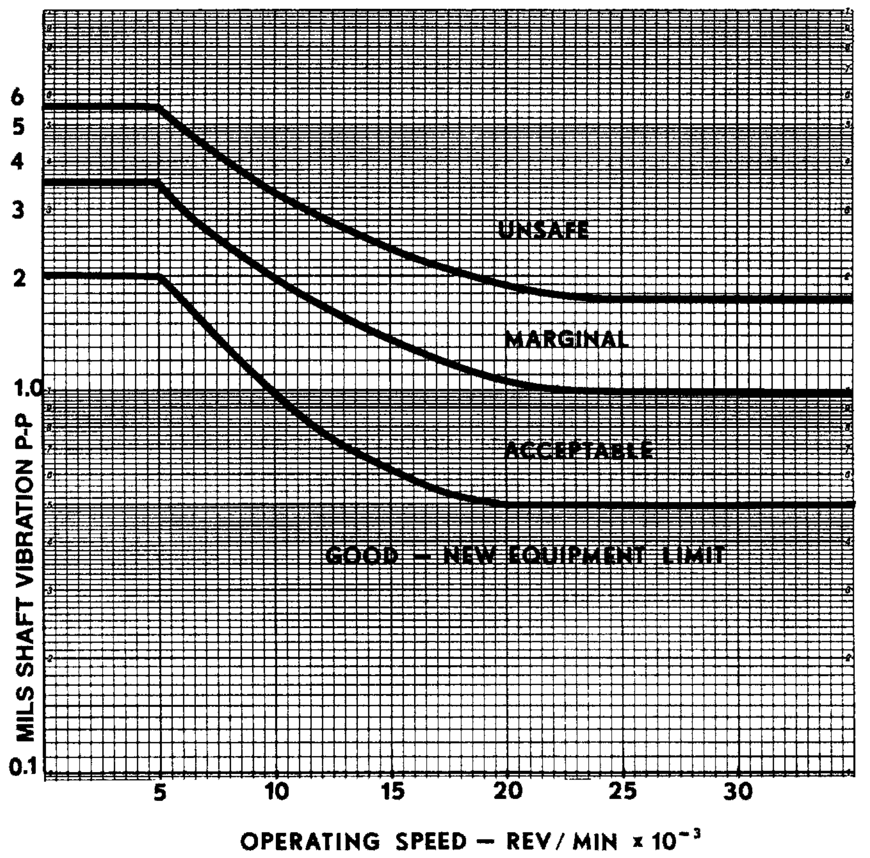
FIGURE 12.113 Finished impeller after slot welding. *(Dresser-Rand Company, Olean, N.Y.)*

- Magnetic particle inspection is conducted after each welding cycle, both before and after heat treatment. Final magnetic particle inspection takes place after the overspeed test.
- Dimensional checks are made after initial welding and after completion of the manufacturing and test process.
- Hardness and tensile tests are performed on the test bars to ensure specification compliance. Hardness checks are made of the impeller cover and disk to verify compliance with specifications and agreement with test bars.

12.11 ROTOR DYNAMIC CONSIDERATIONS

In addition to meeting the aerodynamic expectations, it is equally important that the compression equipment meet the rotor dynamic requirements and be able to operate in a satisfactory and stable manner. Most large equipment today is furnished with noncontacting vibration transducers installed near the bearings. These allow both user and manufacturer to measure rotor vibration amplitudes and frequencies. Maximum acceptable vibration levels at these locations at speed are generally defined in line with applicable API or other industry recommendations. One such guideline essentially allows $(12{,}000/\text{max. cont. speed})^{1/2}$. In addition, manufacturers have also generally established their own minimum margin requirements between operating speeds and critical speeds. These requirements basically follow those set forth in API 670. The majority of compressors operate with flexible rotors, running between their first and second critical speeds. They exhibit shaft vibration amplitudes as indicated by the GOOD range in Fig. 12.114.

The location of the critical speeds plays a large part in the design of a compressor. A manufacturer has to ensure that the compressor will be able to operate over the full speed range intended; this insurance comes from rigorous computer analysis of the critical speeds, unbalance sensitivities, and stability of a system. It is generally in the best interest of overall reliability to make the system as stiff as possible (i.e., to minimize the distance between the journal bearings).



Notes:

1. Operation in the "unsafe" region may lead to near-term failure of the machinery.
2. When operating in the "marginal" region, it is advisable to implement continuous monitoring and to make plans for early problem correction.
3. Periodic monitoring is recommended when operating in the "acceptable" range. Observe trends for amplitude increases at relevant frequencies.
4. The above limits are based on Mr. Zierau's experience. They refer to the typical proximity probe installation close to and supported by the bearing housing and assume that the main vibration component is *1 × rpm frequency*. The seemingly high allowable vibration levels above 20,000 rpm reflect the experience of high-speed air compressors (up to 50,000 rpm) and jet-engine-type gas turbines, with their light rotors and light bearing loads.
5. Readings must be taken on machined surfaces, with runout less than 0.5 mil up to 12,000 rpm, and less than 0.25 mil above 12,000 rpm.
6. Judgment must be used, especially when experiencing frequencies in multiples of operating rpm on machines with standard bearing loads. Such machines cannot operate at the indicated limits for frequencies higher than 1 × rpm. In such cases, enter onto the graph the predominant frequency of vibration instead of the operating speed.

FIGURE 12.114 Turbomachinery shaft vibration chart. (From H. P. Bloch and F. K. Geitner, *Machinery Failure Analysis and Troubleshosssoting*, 3rd ed., Gulf Publishing Company, Houston, Tex., 1998.)

The effort to minimize bearing span manifests itself in various ways in compressor design. These include:

- Positioning the bearings as far inboard in the case end enclosures (heads) as possible
- Minimizing end seal lengths
- Using variable stage spacing

In the past, many casings were designed based on a fixed axial length per stage, usually determined by the largest impeller that the manufacturer anticipated would be used. When smaller impellers were used in these stages, wasted axial space resulted. With variable-stage spacing, each impeller gets only the room it requires. More impellers can thus be accommodated.

The location of the second critical speed can be greatly influenced by overhung weight, which is the weight outboard of the journal bearings at each end. This typically includes the thrust disk and coupling(s). Relocating the thrust-bearing inboard of the journal bearings on a drive-through unit increases the second critical speed. This permits a higher operating speed or increased bearing span, resulting in operating speeds farther removed from the first critical speed.

A typical rotor dynamic analysis starts with a definition of the rotor configuration. The shaft is divided into sections of a given diameter and length, and the impellers, thrust disk, balance piston, and coupling(s) are represented as weights or forces.

Lateral critical speed calculations are then made, with the aid of a computer. This analysis gives the approximate location of the rigid critical speeds, and with the input of different support or bearing stiffnesses, criticals are calculated as a function of stiffness. The resulting plot is known as a *critical speed map* (Fig. 12.115). The mode shapes determined by these calculations are of value in determining the location of the unbalance required for exciting a particular critical speed in the response analysis that follows.

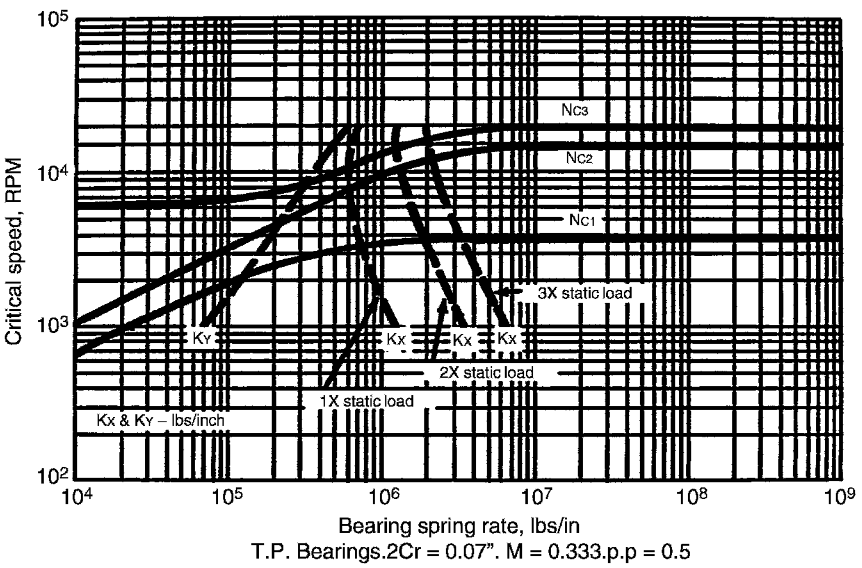


FIGURE 12.115 Critical speed map. *(Dresser-Rand Company, Olean, N.Y.)*

There also exist programs that compute the performance characteristics of the bearings, including oil flow, horsepower loss, oil temperatures, effective eccentricity, and load per pad, along with oil film stiffness and damping coefficients in the vertical and horizontal directions as a function of speed. Most often, these results are cross-plotted on the critical speed map, as shown in Fig. 12.115. The curve labeled K_y is the horizontal bearing stiffness and the curves labeled K_x are the vertical bearing stiffness. The K_x curves are for one, two, and three times the bearing static load. The two- and three-times curves represent $1g$ and $2g$ dynamic or unbalance loads added to the weight load. Although not shown, dynamic loads would similarly influence the K_y stiffness. The intersection of the bearing stiffness and rotor mode curves are the undamped critical speeds. It is seen that the horizontal and vertical critical speeds can be different and that dynamic or unbalance loads influence the critical speed results. Also, as bearing or support stiffness increases, the critical speeds approach the critical speeds of the rotor on rigid or simple support as a limit.

The next step makes use of a rotor response analysis program that calculates the unbalance response of a rotor in fluid film bearings. It is able to predict rotor synchronous vibration behavior at all speeds for a selected unbalance distribution. Bearing stiffness and damping characteristics are part of the input. The motion of the rotor is treated as two-dimensional. The output gives the major and minor axes of an ellipse formed by the locus of the shaft center. Dynamic bearing forces can also be determined for the bearings. Examples of typical results from such an analysis are shown in Figs. 12.116 (amplitude vs. speed) and 12.117 (bearing force vs. speed). This program has proven to be a most valuable tool for predicting the synchronous vibration behavior of rotating machinery.

The most recent analytical tool, developed during the mid-1970s, is rotor stability analysis. This technique has been used to increase the understanding of various rotor instability phenomena that became apparent in compressors designed for extreme-high-pressure applications.

The stability program integrates rotor geometry with support stiffness and damping to determine system-critical speeds, damped mode shapes, and an exponential representation

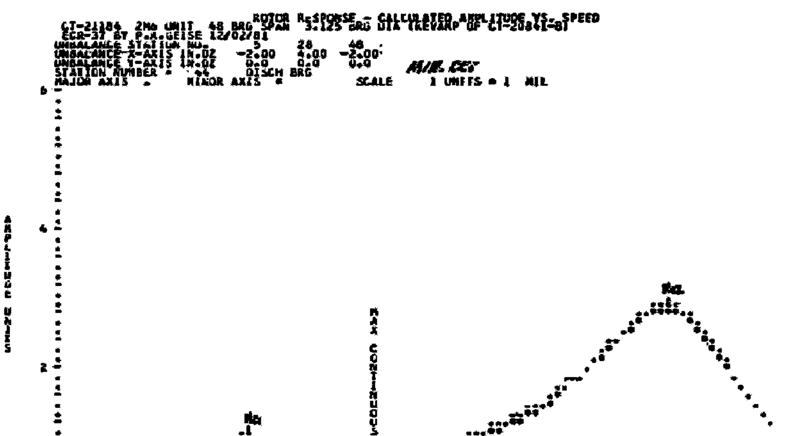


FIGURE 12.116 Amplitude vs. speed plot (unbalance response plot). *(Dresser-Rand Company, Olean, N.Y.)*

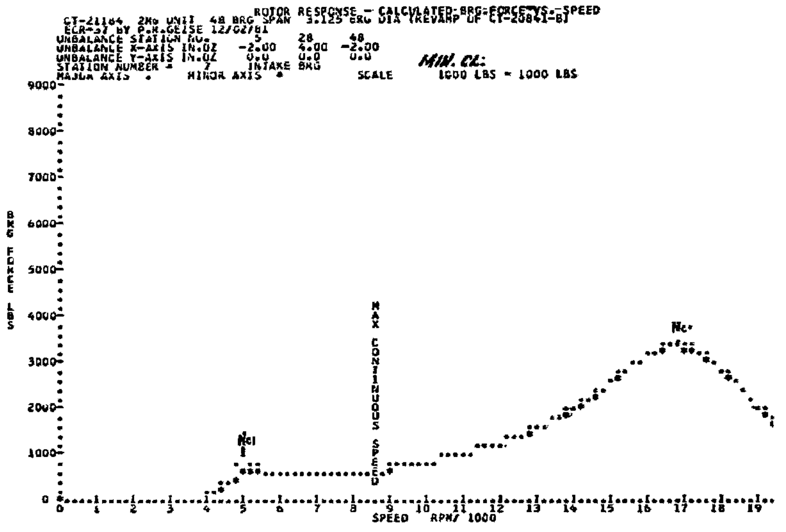


FIGURE 12.117 Bearing force vs. speed plot (unbalance response plot). *(Dresser-Rand Company, Olean, N.Y.)*

of system stability. It is complicated in that eight coefficients are used for each bearing: four stiffnesses, normal and cross-coupling, and four damping, normal and cross-coupling. These can be input for the seals if desired. In addition, the program can accept other input, such as aerodynamic effects, internal friction, and negative damping.

These analysis methods yield critical speeds and logarithmic decrements together with three-dimensional damped shaft mode shapes. The critical speed and logarithmic or *log* decrement output is plotted in the format illustrated in Fig. 12.118. It displays critical frequency in cycles per minute vs. shaft speed in revolutions per minute so that synchronous excitation will be a 45° line as shown. These curves represent the vertical and horizontal modes of the first and second critical frequencies, respectively. In essence, these curves indicate how the critical frequency varies with shaft speed, and the intersections with synchronous excitation are the critical speeds. Log decrement values offer a relative means in ranking the ability of a system to cope with undesirable excitation. The higher the value, the greater the excitation required to make the system unstable.

Torsional studies represent another type of analysis sometimes performed on a rotor system. The torsional natural frequencies are calculated by dividing the system into a number of mass moments of inertia separated by torsional spring constants for the various shaft sections. An example is given in Fig. 12.119. Results are obtained by use of a computer program.

For the majority of systems, only shaft rotational frequency is considered as potential excitation. A system is considered satisfactory if the torsional natural frequencies are 10% or more away. If closer than 10%, the system is tuned. An interference diagram is shown in Fig. 12.120. The sloped line represents excitation. Note how the natural frequencies are well removed from the operating range in this example.

Although steady-state torsional problems are rare, a potentially troublesome transient torsional problem exists in conjunction with synchronous motor drivers during startup. Widely different motor startup characteristics are possible, and their effect on the system must be analyzed. Computerized procedures that can accurately predict alternating and peak torque levels have been developed. System design integrity can thus be ascertained.

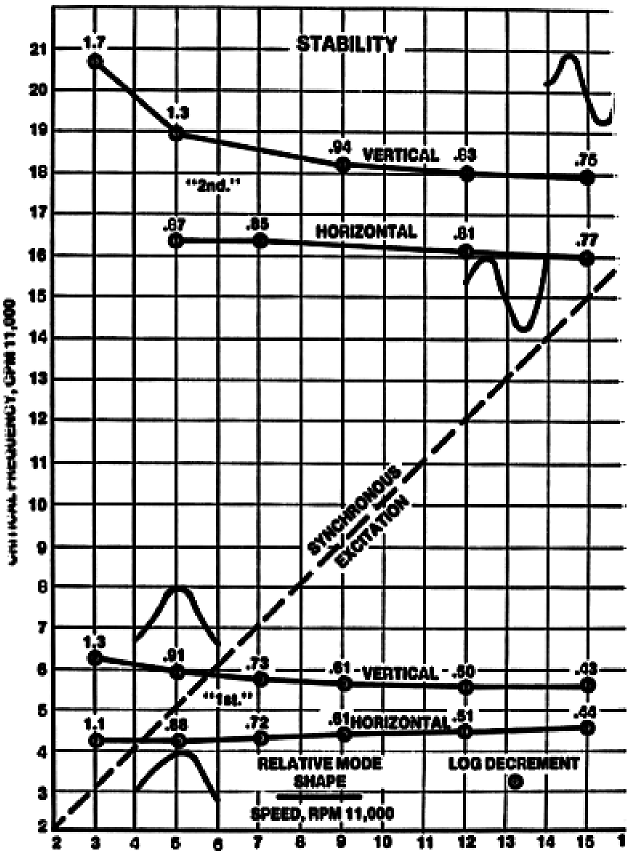


FIGURE 12.118 Rotor stability (logarithmic decrement) plot. *(Dresser-Rand Company, Olean, N.Y.)*

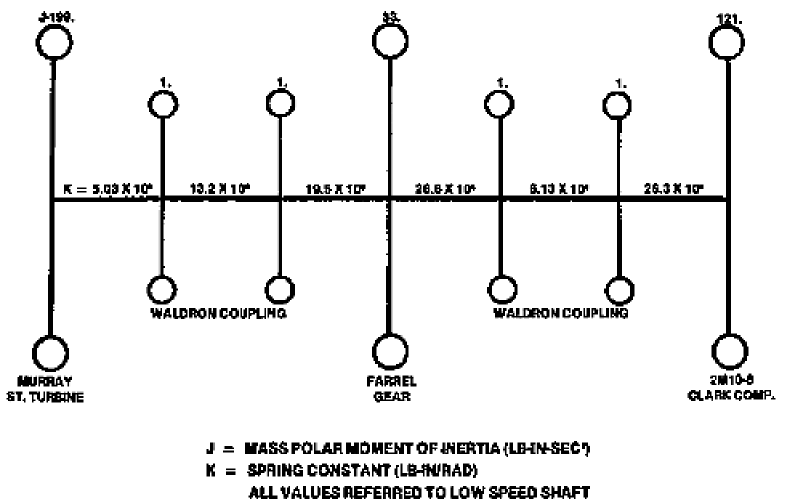


FIGURE 12.119 Torsional natural frequency plot. *(Dresser-Rand Company, Olean, N.Y.)*

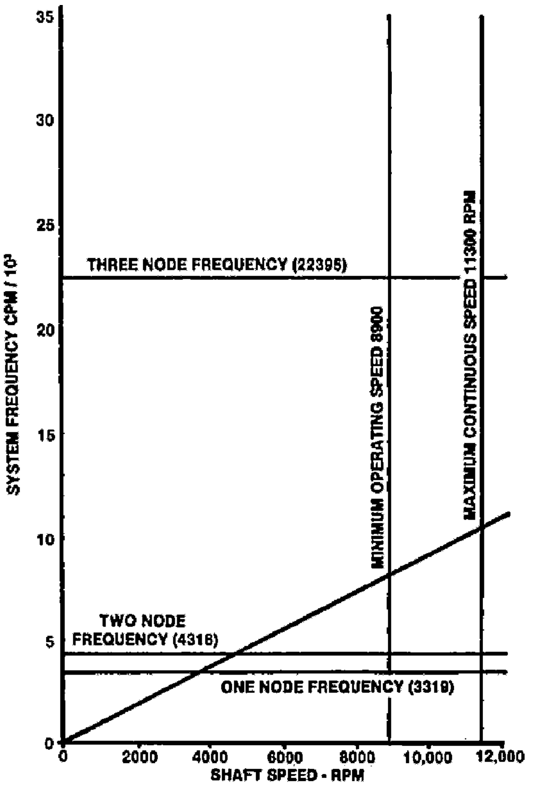


FIGURE 12.120 Interference diagram. *(Dresser-Rand Company, Olean, N.Y.)*

Steady-state torsional problems are somewhat more likely to occur in gear-driven compression systems. Here, an appropriate torsional analysis takes on added importance.

12.12 FOULING CONSIDERATIONS AND COATINGS*

Process compressors are required to run at or near peak efficiencies for long periods of time. It is on this basis that two solutions to the polymerization problem, flush liquid injection and the application of coatings, have been implemented.

The introduction of liquid into the gas stream is always process-dependent and is thus not considered within the scope of this book. However, modern coating technology should be reviewed within the context of centrifugal compressor design.

12.12.1 Polymerization and Fouling

The chemical mechanism that takes place to generate polymerization is not well understood as it applies to compressor fouling. However, what is known is that hydrocarbons inherent in the process gas, or formed during the compression process, can bond tenaciously to components and lead to significant performance loss (see Fig. 12.121). Deposits of this

* From R. Chow, B.McMordie, and R. Wiegand, Coatings limit compressor fouling, *Turbomachinery International*, Jan.–Feb. 1995. Adapted by permission of the publishers.

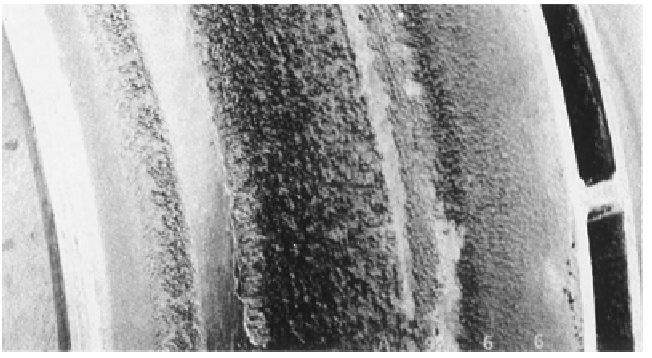


FIGURE 12.121 Centrifugal compressor impeller, showing fouling deposits. (*Turbomachinery International, Norwalk, Conn.*)

type have been found in compressors used for hydrocarbon processing, coke gas blowers, and other units where the gas contains sufficient amounts of hydrocarbons under the right conditions of pressure and temperature.

Factors that have been found empirically to be critical to the fouling process are:

- *Temperature.* Polymerization occurs above 194°F (90°C).
- *Pressure.* The extent of the fouling is proportional to pressure.
- *Surface finish.* The smoother the surface, the less apt the component is to foul.
- *Gas composition.* Fouling is proportional to the concentration of reactable hydrocarbon in the process (inlet) gas.

12.12.2 Fouling and Its Effect on Compressor Operation

Component fouling has many detrimental effects on compressor operation.

Unbalance One of the most obvious is the buildup of material on the rotor. This can lead to unbalance, which gradually builds until the unit exceeds its allowable vibration limit and has to be shut down to correct the problem. In addition, operation with significant rotor unbalance can lead to fatigue loading and a possible reduction in component life.

Abrasive Wear Deposits have also been known to reduce both the axial and radial clearances between the rotor and the stationary components. This clearance reduction has led to abrasive wear, which has severely damaged numerous impellers and labyrinth seals.

Unbalance and abbrasive wear are progressive, with the costs associated with correction showing up after longer periods of operation.

Loss of Efficiency When considering fouling that affects unit performance, the losses and associated costs are revealed very quickly, typically only months after the unit is started. This has been confirmed by actual operation. The case study that follows describes a 6% decrease in efficiency after only 17 months of operation. In addition, it was found elsewhere that the most intensive growth of the deposition layer occurred during the first 50 to 200 hours of operation. These examples reinforce the fact that fouling degradation occurs early and can cause significant losses, making it increasingly important to take corrective action from the start to assure optimum efficiency.

Fouling affects efficiency through three basic loss mechanisms:

1. Friction losses
2. Flow area reductions
3. Random changes of pressure distribution on the blade

These mechanisms affect both the stationary flow path and the rotating element. However, in the past, attempts to correct the fouling problem were limited to the diffuser and return channels, which were considered to be the most susceptible. The rotating element is less likely to foul due to the dynamic force applied to the deposits because of the dislodging effect of rotation. In addition, by design, the stationary flow paths have slightly rougher surface finishes than those of the rotating element. Today, however, fouling of both stationary and rotating flow path components needs to be addressed.

The Elliott Company has modeled the effect of fouling on compressor performance using their compressor performance prediction program. This work simulated the effect of deposits on stationary flow path surfaces on the performance of an Elliott 38M9 centrifugal compressor operating under the following conditions:

- Compressor speed = 6150 rpm
- Inlet pressure = 228 psia
- Inlet temperature = 142°F
- Gas containing a mixture of various hydrocarbons

The study evaluated the effects of diffuser passage width and surface finish on head and efficiency. It showed that losses of 10% or more could result from passage width restrictions of 10% or greater and surface finishes of 500 rpm or higher (Figs. 12.122 and 12.123). It should again be noted that compressor washing, using either water to lower gas temperature or a hydrocarbon solvent to dissolve the deposits, has been used to reduce the extent of fouling.

12.12.3 Coating Case Study

Novacor Chemicals' Ethylene 2 plant is a world-scale petrochemical plant located in Joffre, just east of Red Deer, Alberta, Canada. The main feedstock, ethane, is cracked to make ethylene and other hydrocarbon by-products. Approximately 1.8 billion pounds of ethylene is produced by Ethylene 2 annually.

Ethane is cracked in furnaces and then compressed by the cracked gas compressor string for finish processing and separation of products. The cracked gas compressor string consists of Elliott 88 M, 60 M, and 46 M compressors driven by an Elliott NV9 steam turbine (Fig. 12.124). Historical performance data show that the compressors foul during operation. The formation of polymers in the 46 M compressor is greatest, as this has the highest pressure and temperature in the compressor string.

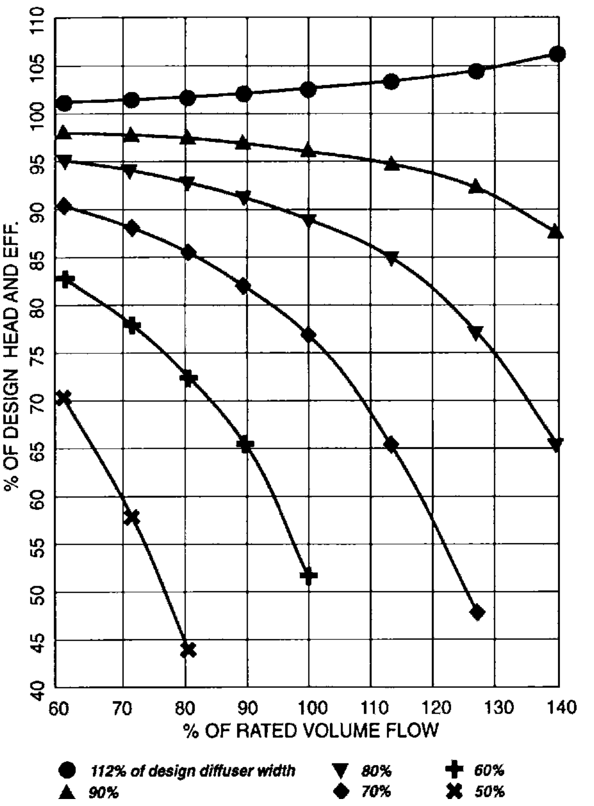


FIGURE 12.122 Effect of diffuser width change on polytropic head and efficiency. (*Turbomachinery International, Norwalk, Conn.*)

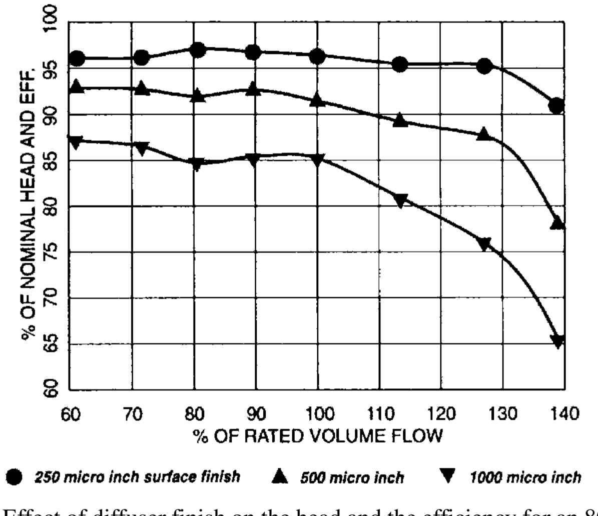


FIGURE 12.123 Effect of diffuser finish on the head and the efficiency for an 80% diffuser width. (*Turbomachinery International, Norwalk, Conn.*)

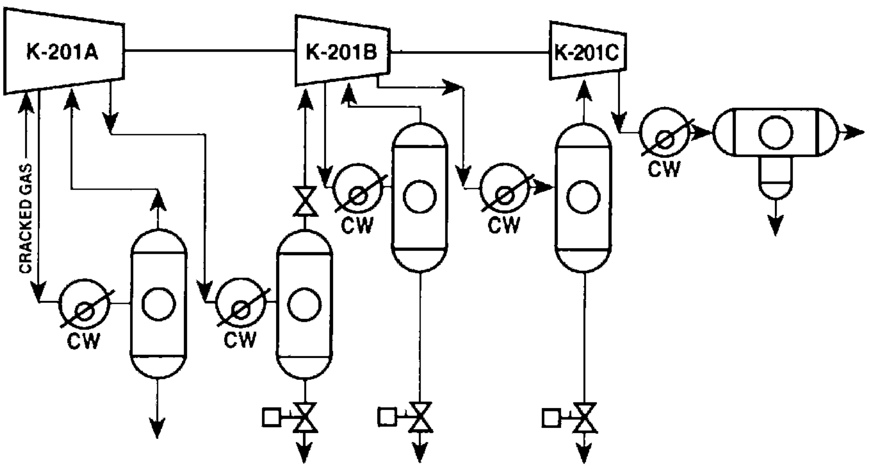


FIGURE 12.124 Compressor string at Novacor Chemicals: K-201A = Elliott 88 M, K-201B = Elliott 60 M, K-201C = Elliott 46 M. (*Turbomachinery International, Norwalk, Conn.*)

To increase reliability and improve the run time of the equipment, coating of the compressor parts was considered. A coating would have to provide three benefits:

1. A *nonstick surface*, so that fouling could not form on the surfaces and degrade performance
2. An *erosion barrier* to the current wash oil injection practice in the compressors
3. *Corrosion protection,* to maintain the finish on aerodynamic surfaces

A risk analysis of the coating showed minimal impact on the process and equipment should the coating not function as designed. The only concern was that unstacking the spare rotor was necessary to coat the compressor wheels. At the next shutdown of the plant, the rotor wheels were coated by Sermatech using the SermaLon coating system.

12.12.4 SermaLon Coating

The SermaLon coating system developed by Sermatech was designed for wet, corrosive environments. It combines the benefits and features of the three types of coatings generally used to combat corrosion of metal components: barrier coatings, inhibitive coatings, and sacrificial coatings.

The outermost layer of the coating system is a high-temperature resin film, which is a barrier against corrodants in the environment. Barrier coatings prevent corrosion by sealing the substrate from environmental effects. But once this seal is broken, corrosion proceeds unchecked at the point of the breech in the coating.

The intermediate layer of the SermaLon coating system is a durable inhibitive coating. Inhibitive coatings contain pigments (e.g., chromates or complex metalo-organic compounds) that prevent corrosion by modifying the chemistry of environmental corrodants contacting the coated surface. These reactions change the pH, reactivity, and even the molecular structure of the corrodants. Inhibitive coatings are very effective as long as the

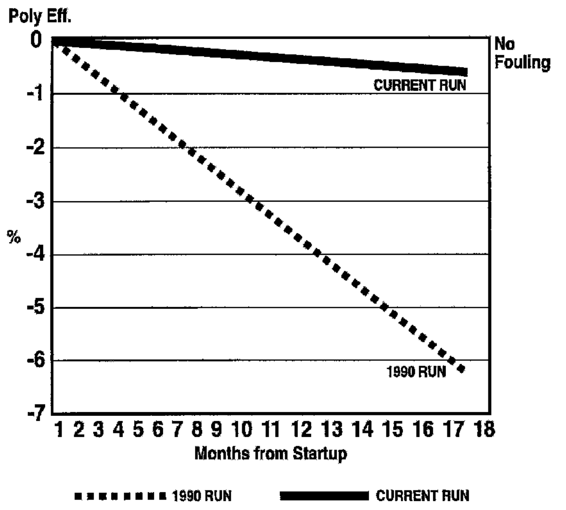


FIGURE 12.125 Fouling history of the compressor at Ethylene 2. (*Turbomachinery International, Norwalk, Conn.*)

film remains intact and inhibitive pigments remain reactive. When the coating is breeched, corrosion of the exposed substrate is slowed until active pigments are depleted.

The foundation of the coating system is a tightly adherent layer of a sacrificial aluminum-filled ceramic. Sacrificial or *galvanic* coatings prevent corrosion of structural hardware instead of the substrate. They are made of a more *active* metal, which when placed in contact with a less-active (more *noble*) metal will be consumed entirely by the environment before the more noble material begins to corrode.

12.12.5 Results

The performane of the compressor to date has shown that the efficiency has remained virtually constant. For comparison, there was a 6% reduction in efficiency during the previous run, without coatings, in the same amount of time (Fig. 12.125). The success in maintaining the performance of the compressor is attributed to the coating and washing system. However, it is impossible to quantify the effect of each alone. It is estimated that the payback for the coating of the rotor is two months.