SECTION 2.2
CENTRIFUGAL PUMP CONSTRUCTION

2.2.1
CENTRIFUGAL PUMPS: MAJOR COMPONENTS

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CLASSIFICATION AND NOMENCLATURE

A centrifugal pump consists of a set of rotating vanes enclosed within a housing or casing that is used to impart energy to a fluid through centrifugal force. Thus, stripped of all refinements, a centrifugal pump has two main parts: (1) a rotating element, including an impeller and a shaft, and (2) a stationary element made up of a casing, casing cover, and bearings.

In a centrifugal pump, the liquid is forced by atmospheric or other pressure into a set of rotating vanes. These vanes constitute an impeller that discharges the liquid at its periphery at a higher velocity. This velocity is converted to pressure energy by means of a volute (see Figure 1) or by a set of stationary diffusion vanes (see Figure 2) surrounding the impeller periphery. Pumps with volute casings are generally called *volute pumps*, while those with diffusion vanes are called *diffuser pumps*. Diffuser pumps were once quite commonly called *turbine pumps*, but this term has become more selectively applied to the vertical deep-well centrifugal diffuser pumps usually referred to as *vertical turbine pumps*. Figure 1 shows the path of the liquid passing through an end-suction volute pump operating at rated capacity (the capacity at which best efficiency is obtained).

Impellers are classified according to the major direction of flow in reference to the axis of rotation. Thus, centrifugal pumps may have the following:

- Radial-flow impellers (see Figures 25, 34, 35, 36, and 37)
- Axial-flow impellers (see Figure 29)
- Mixed-flow impellers, which combine radial- and axial-flow principles (see Figures 27 and 28)

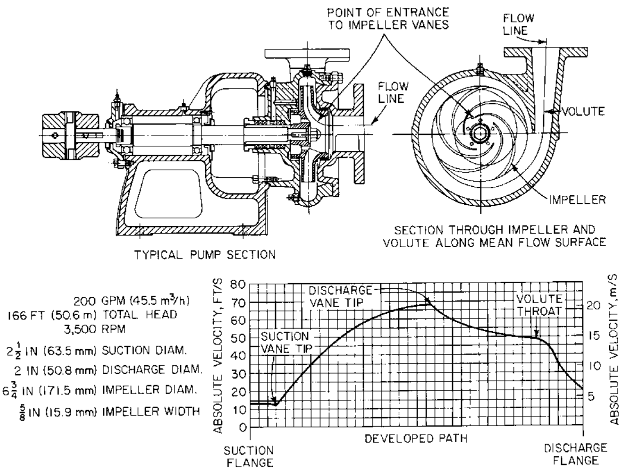


FIGURE 1 A typical single-stage, end-suction volute pump (Flowserve Corporation)

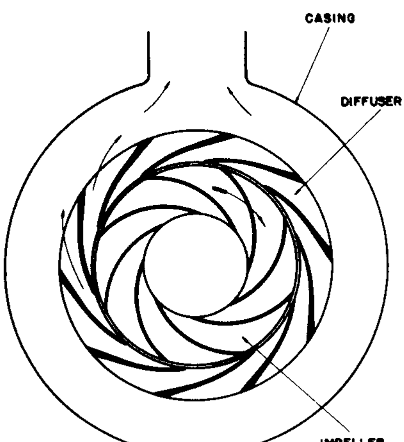


FIGURE 2 A typical diffuser-type pump

Impellers are further classified in one of two categories:

- Single-suction, with a single inlet on one side (see Figures 25, 33, and 37)
- Double-suction, with liquid flowing to the impeller symmetrically from both sides (see Figures 26, 27, and 38)

The mechanical construction of the impellers gives a still further subdivision into

- Enclosed, with shrouds or side walls enclosing the waterways (see Figures 25, 26, and 27)
- Open, with no shrouds (see Figures 29, 33, and 34)
- Semiopen or semi-enclosed (see Figure 36)

A pump in which the head is developed by a single impeller is called a *single-stage pump*. Often the total head to be developed requires the use of two or more impellers operating in a series, each taking its suction from the discharge of the preceding impeller. For this purpose, two or more single-stage pumps can be connected in a series or all the impellers can be incorporated in a single casing. The unit is then called a *multistage pump*.

The mechanical design of the casing provides the added pump classification of *axially split* or *radially split*, and the axis of rotation determines whether the pump is a horizontal or vertical unit.

Horizontal-shaft centrifugal pumps are classified still further according to the location of the suction nozzle:

- End-suction (see Figures 1, 8, 11, and 13)
- Side-suction (see Figures 7 and 10)
- Bottom-suction (see Figure 15)
- Top-suction (see Figure 22)

Some pumps operate in air with the liquid coming to and being conducted away from the pumps by piping. Other pumps, most often vertical types, are submerged in their suction supply. Vertical-shaft pumps are therefore called either *dry-pit* or *wet-pit* types. If the wet-pit pumps are axial-flow, mixed-flow, or vertical-turbine types, the liquid is discharged up through the supporting drop or column pipe to a discharge point above or below the supporting floor. These pumps are consequently designated as *aboveground discharge* or *belowground discharge units*.

Figures 3, 4, and 8 show typical constructions of a horizontal double-suction volute pump, the bowl section of a single-stage axial-flow propeller pump, and a vertical dry-pit single-suction volute pump, respectively. The names recommended by the Hydraulic Institute for various pump parts are given in Table 1.

CASINGS AND DIFFUSERS

The Volute Casing Pump This pump (refer to Figure 1) derives its name from the spiral-shaped casing surrounding the impeller. This casing section collects the liquid discharged by the impeller and converts velocity energy to pressure energy.

A centrifugal pump volute increases in area from its initial point until it encompasses the full 360° around the impeller and then flares out to the final discharge opening. The

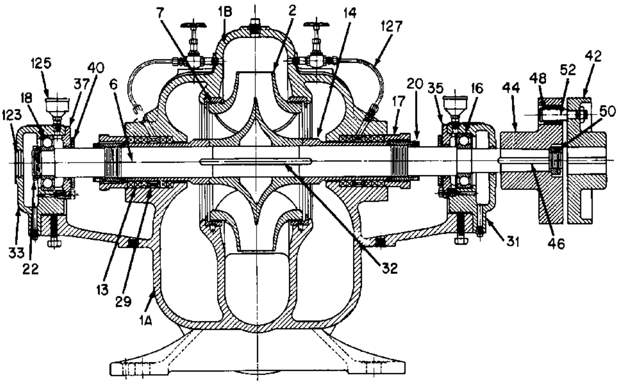


FIGURE 3 Horizontal single-stage double-suction volute pump (the numbers refer to parts listed in Table 1) (Flowserve Corporation)

TABLE 1 Recommended names of centrifugal pump parts

Item no.	Name of part	Item no.	Name of part
1	Gasing	33	Bearing housing (outboard)
1A	Gasing (lower half)	35	Bearing cover (inboard)
1B	Gasing (upper half)	36	Propeller key
2	Impeller	37	Bearing cover (outboard)
4	Propeller	39	Bearing bushing
6	Pump shaft	40	Deflector
7	Gasing ring	42	Coupling (driver half)
8	Impeller ring	44	Coupling (pump half)
9	Suction cover	46	Coupling key
11	Stuffing box cover	48	Coupling bushing
13	Packing	50	Coupling lock nut
14	Shaft sleeve	52	Coupling pin
15	Discharge bowl	59	Handhole cover
16	Bearing (inboard)	68	Shaft collar
17	Gland	72	Thrust collar
18	Bearing (outboard)	78	Bearing spacer
19	Frame	85	Shaft enclosing tube
20	Shaft sleeve nut	89	Seal
22	Bearing lock nut	91	Suction bowl
24	Impeller nut	101	Column pipe
25	Suction head ring	103	Connector bearing
27	Stuffing box cover ring	123	Bearing end cover
29	Lantern ring (seal cage)	125	Grease (oil) cup
31	Bearing housing (inboard)	127	Seal pipe (tubing)
32	Impeller key		

[a]These parts are called out in Figures 3, 4, and 8.

wall dividing the initial section and the discharge nozzle portion of the casing is called the tongue of the volute or the "cutwater." The diffusion vanes and concentric casing of a diffuser pump fulfill the same function as the volute casing in energy conversion.

In propeller and other pumps in which axial-flow impellers are used, it is not practical to use a volute casing. Instead, the impeller is enclosed in a pipe-like casing. Generally, diffusion vanes are used following the impeller proper, but in certain extremely low-head units these vanes may be omitted.

A diffuser is seldom applied to a single-stage, radial-flow pump, except in special instances where volute passages become so small that machined or precision-cast volute or diffuser-like pieces are utilized for precise flow control. Conventional diffusers are often applied to multistage pump designs in conjunction with guide vanes to direct the flow efficiently from one impeller (stage) to another in a minimum radial and axial space. Diffuser vanes are used as the primary construction method for vertical turbine pumps and single-stage, low-head propeller pumps (see Figure 4).

Radial Thrust In a *single-volute pump casing* design (see Figure 5), uniform or near uniform pressures act on the impeller when the pump operates at design capacity (which coincides with the best efficiency). At other capacities, the pressures around the impeller are not uniform (see Figure 6) and there is a resultant radial reaction (F). A detailed discussion of the radial thrust and of its magnitude is presented in Subsection 2.3.1. Note that the unbalanced radial thrust increases as capacity decreases from that at the design flow.

For any percentage of capacity, this radial reaction is a function of total head and of the width and diameter of the impeller. Thus, a high-head pump with a large impeller diameter will have a much greater radial reaction force at partial capacities than a low-head

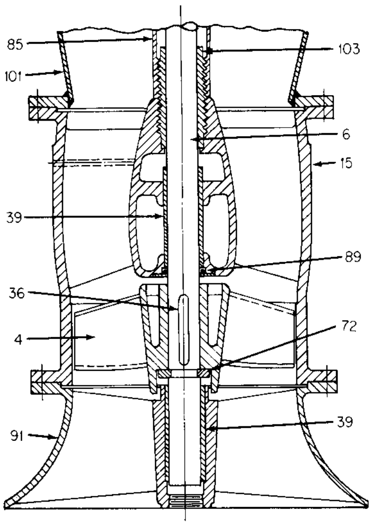


FIGURE 4 Vertical wet-pit diffuser pump bowl (the number refers to parts listed in Table 1) (Flowserve Corporation)

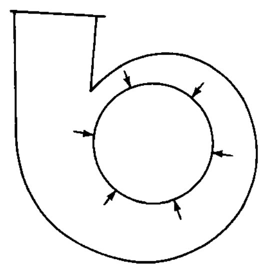


FIGURE 5 Uniform casing pressures exist at design capacity, resulting in zero radial reaction.

pump with a small impeller diameter. A zero radial reaction is not often realized; the minimum reaction occurs at design capacity.

Although the same tendency for unbalance exists in the diffuser-type pump, the reaction is limited to a small arc repeated all around the impeller. As a result, the individual reactions cancel each other out as long as flow is constantly removed from around the periphery of the diffuser discharge. If flow is not removed uniformly around its periphery, a pressure imbalance may occur around the diffuser discharge that will be transmitted back through the diffuser to the impeller, resulting in a radial reaction on the shaft and bearing system.

In a centrifugal pump design, shaft diameter as well as bearing size can be affected by the allowable deflection as determined by the shaft span, impeller weight, radial reaction forces, and torque to be transmitted.

Formerly, standard designs compensated for radial reaction forces encountered at capacities in excess of 50 percent of the design capacity for the maximum-diameter impeller of the pump. For sustained operations at lower capacities, the pump manufacturer, if properly advised, would supply a heavier shaft, usually at a much higher cost. Sustained operations at extremely low flows without the manufacturer being informed at the time of purchase are a much more common practice today. This can result in broken shafts, especially older designs, on high-head units.

Because of the increasing application of pumps that must operate at reduced capacities, it has become desirable to design standard units to accommodate such conditions. One solution is to use heavier shafts and bearings. Except for low-head pumps in which only a small additional load is involved, this solution is not economical. The only practical answer is a casing design that develops a much smaller radial reaction force at partial capacities. One of these is the *double-volute* casing design, also called the *twin-volute* or *dual-volute* design.

The application of the double-volute design principle to neutralize radial reaction forces at reduced capacity is illustrated in Figure 7. Basically, this design consists of two 180° volutes, and a passage external to the second joins the two into a common discharge. Although a pressure unbalance exists at partial capacity through each 180° arc, the two

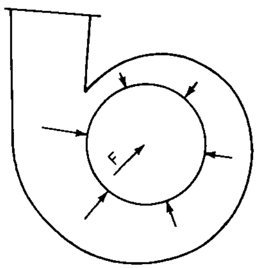


FIGURE 6 At reduced capacities, uniform pressures do not exist in a single-volute casing, resulting in a radial reaction *F*.

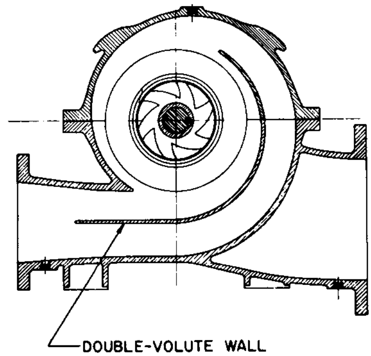


FIGURE 7 Transverse view of a double-volute casing pump.

forces are approximately equal and opposite. Thus, little if any radial force acts on the shaft and bearings. Subsection 2.3.1 also covers this topic.

The double-volute design has many hidden advantages. For example, in large-capacity medium- and high-head single-stage vertical pumps, the rib forming the second volute that separates it from the discharge waterway of the first volute strengthens the casing (see Figure 8).

The individual stages of a multistage pump can be made double volute, as illustrated in Figure 9. The kinetic energy of the pumped liquid discharged from the impeller must be transformed into pressure energy and then must be turned 180° to enter the impeller of the next stage. The double volute therefore also acts as a return channel. The back view in Figure 9 shows this as well as the guide vanes used to straighten the flow into the next stage. An alternative to the volute design for multistage pumps is the diffuser and its return vanes that channel the flow from the discharge of the diffuser vanes back into the impeller of the next stage.

Solid and Split Casings *Solid casing* implies a design in which the discharge waterways leading to the discharge nozzle are all contained in one casting or fabricated piece. The casing must have one side open so that the impeller can be introduced into it. Because the sidewalls surrounding the impeller are actually part of the casing, a solid casing, strictly speaking, cannot be used, and designs normally called solid casing are really radially split (refer to Figure 1 and see Figures 11, 12, and 13).

A *split casing* is made of two or more parts fastened together. The term *horizontally split* had regularly been used to describe pumps with a casing divided by a horizontal plane through the shaft centerline or axis (see Figure 10). The term *axially split* is now preferred. Because both the suction and discharge nozzles are usually in the same half of the casing, the other half may be removed for inspection of the interior without disturbing the bearings or the piping. Like its counterpart horizontally split, the term *vertically split* is poor terminology. It refers to a casing split in a plane perpendicular to the axis of rotation. The term *radially split* is now preferred.

End-Suction Pumps Most end-suction, single-stage pumps are made of one-piece solid casings. At least one side of the casing must be open so that the impeller can be assembled in the pump. Thus, a cover is required for that side. If the cover is on the suction side, it becomes the casing sidewall and contains the suction opening (refer to Figure 1). This is called the *suction cover* or *casing suction head*. Other designs are made with casing covers (see Figure 12) and still others have both casing suction covers and casing covers (refer to Figure 8 and see Figure 13).

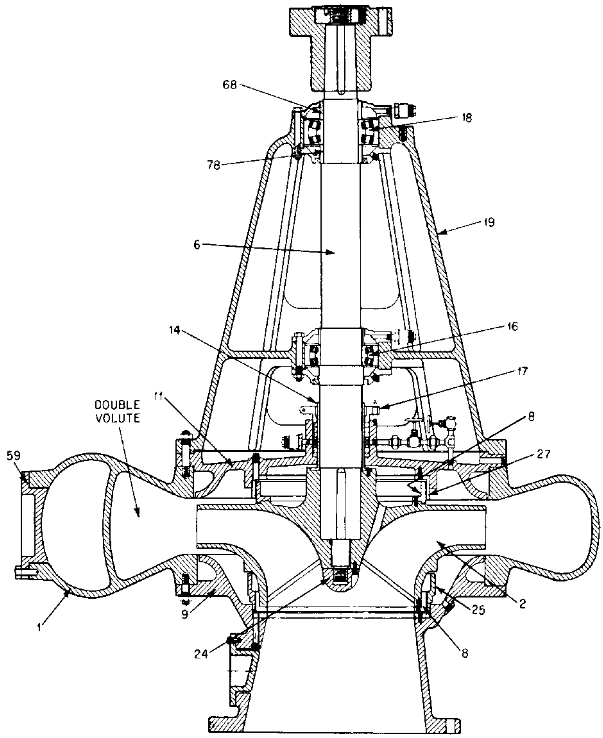


FIGURE 8 The sectional view of a vertical-shaft, end-suction pump with a double-volute casing (the numbers refer to parts listed in Table 1) (Flowserve Corporation)

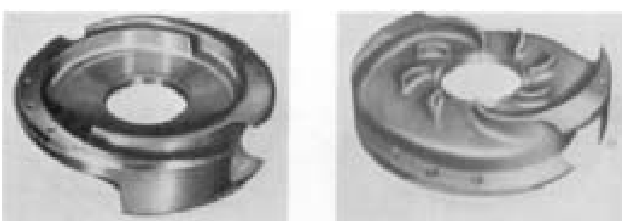


FIGURE 9 The double volute of a multistage pump, front view (left) and back view (right) (Flowserve Corporation)

For general service, the end-suction, single-stage pump design is extensively used for small pumps with a 4- or 6-in (102 or 152 mm) discharge size for both motor-mounted and coupled types. In these pumps, the small size makes it feasible to cast the volute and one side integrally. Whether or not the seal chamber side or the suction side is made integrally with the casing is usually determined by the most economical pump design.

For larger pumps, especially those for special services such as sewage handling, there is a demand for pumps of both rotations. A design with separate suction and seal chamber heads permits the use of the same casing for either rotation if the flanges on the two sides

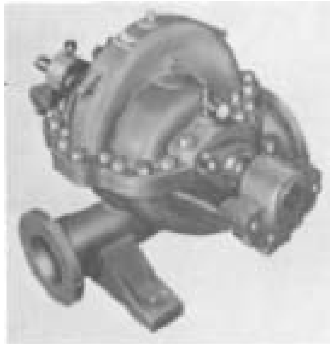


FIGURE 10 An axially split casing, horizontal-shaft, double-suction volute pump (Flowserve Corporation)

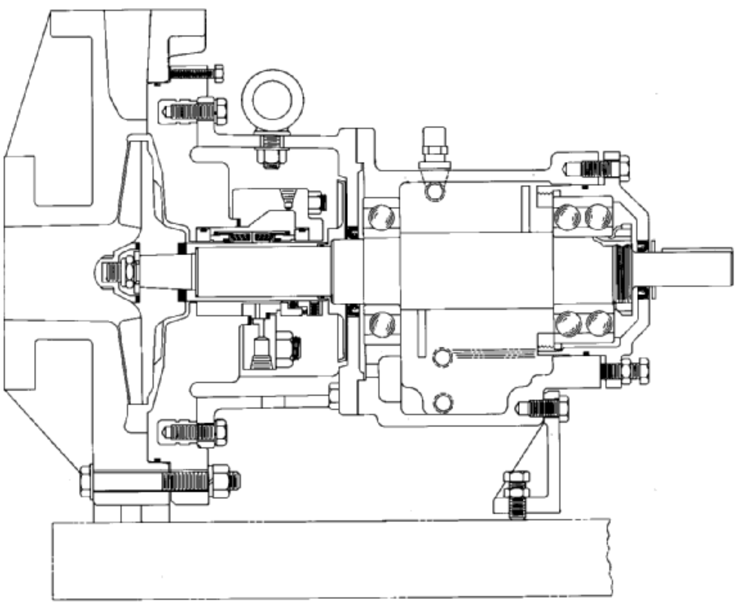


FIGURE 11 End-suction pump with semi-open impeller (Flowserve Corporation)

are made identical. There is also a demand for vertical pumps that can be disassembled by removing the rotor and bearing assembly from the top of the casing. Many horizontal applications of the pumps of the same line, however, require partial dismantling from the suction side. Such lines are most adaptable when they have separate suction and casing covers.

Casing Construction for Open- and Semiopen-Impeller Pumps In the open- or semiopen impeller pump, the impeller rotates within close clearance of the pump casing or suction cover (refer to Figure 11). If the intended service is abrasive, a side plate is mounted within the casing to provide a renewable close-clearance guide to the liquid flow-

FIGURE 12 End-suction pump with semi-open impeller, inducer and renewable side plate (Flowserve Corporation)

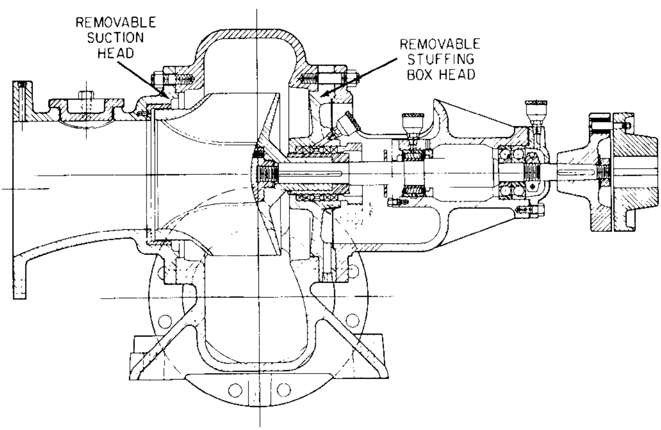


FIGURE 13 End-suction pump with removable suction and stuffing box heads (Flowserve Corporation)

ing through the impeller (refer to Figure 12). One of the advantages of using side plates is that abrasion-resistant material, such as stainless steel, can be used for the impeller and side plate, while the casing itself may be of a less costly material. Although double-suction, semiopen-impeller pumps are seldom used today, they were common in the past and were generally made with side plates.

In order to maintain pump efficiency, a close running clearance is required between the front unshrouded face of the open or semiopen impeller and the casing, suction cover, or side plate. Pump designs provide either jackscrews or shims to adjust the position of the thrust bearing housing (and, as a result, the axial position of the shaft and impeller) relative to the bearing frame.

Pre-rotation and Stop Pieces Improper entrance conditions and inadequate suction approach shapes may cause the liquid column in the suction pipe to spiral for some distance ahead of the impeller entrance. This phenomenon is called *pre-rotation*, and it is attributed to various operational and design factors in both vertical and horizontal pumps.

Pre-rotation is usually harmful to pump operation because the liquid enters between the impeller vanes at an angle other than that allowed in the design. This frequently lowers the net effective suction head and the pump efficiency. Various means are used to avoid pre-rotation both in the construction of the pump and in the design of the suction approaches.

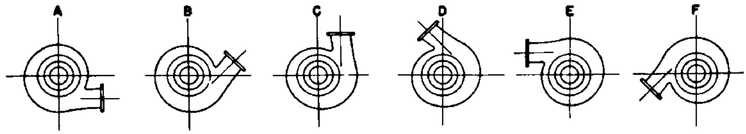


FIGURE 14 Possible positions of discharge nozzles for a specific design of an end-suction, solid-casing, horizontal-shaft pump. The rotation illustrated is counterclockwise from suction end.

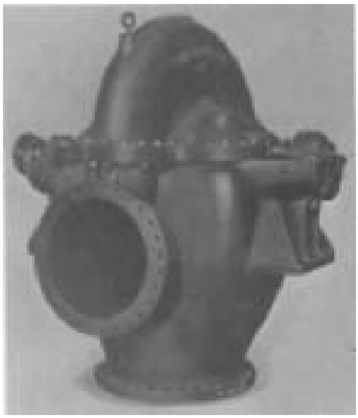


FIGURE 15 A bottom-suction, axially split casing single-stage pump (Flowserve Corporation)

Practically all horizontal, single-stage, double-suction pumps and most multistage pumps have a suction volute that guides the liquid in a streamline flow to the impeller eye. The flow comes to the eye at right angles to the shaft and separates unequally on the two sides of the shaft. Moving from the suction nozzle to the impeller eye, the suction waterways are reduced in area, meeting in a projecting section of the sidewall dividing the two sections. This dividing projection is called a *stop piece*. To minimize pre-rotation in end-suction pumps, a radial-fin stop piece projecting toward the center is sometimes cast into the suction nozzle wall.

Nozzle Locations The discharge nozzle of end-suction, single-stage horizontal pumps is usually in a top-vertical position (refer to Figures 1, 11, and 12). However, other nozzle positions can be obtained, such as top-horizontal, bottom-horizontal, or bottom-vertical. Figure 14 illustrates the flexibility available in discharge nozzle locations. Sometimes the pump frame, bearing bracket, or baseplate may interfere with the discharge flange, prohibiting a bottom-horizontal or bottom-vertical discharge nozzle position. In other instances, solid casings cannot be rotated for various nozzle positions because the seal chamber connection would become inaccessible.

Practically all double-suction, axially split casing pumps have a side discharge nozzle and either a side- or a bottom-suction nozzle. If the suction nozzle is placed on the side of the pump casing with its axial centerline (refer to Figure 10), the pump is classified as a *side-suction pump*. If its suction nozzle points vertically downward (see Figure 15), the pump is called a *bottom-suction pump*. Single-stage, bottom-suction pumps are rarely made in sizes below a 10-in (254 mm) discharge nozzle diameter.

Special nozzle positions can sometimes be provided for double-suction, axially split casing pumps to meet special piping arrangements, such as a radically split casing with bottom suction and top discharge in the same half of the casing. Such special designs are generally costly and should be avoided.

Centrifugal Pump Rotation Because suction and discharge nozzle locations are affected by pump rotation, it is important to understand how the direction of rotation is defined. According to Hydraulic Institute standards, rotation is defined as clockwise or counterclockwise by looking at the driven end of a horizontal pump or looking down on a vertical pump. To avoid misunderstanding, clockwise or counterclockwise rotation should always be qualified by including the direction from which one looks at the pump.

The terms *inboard end* or *drive end* (the end of the pump closest to the driver) and *outboard end* or *nondrive end* (the end of the pump farthest from the driver) are used only with horizontal pumps. The terms lose their significance with dual-driven pumps. Any centrifugal pump casing pattern can be arranged for either clockwise or counterclockwise rotation, except for end-suction pumps, which have integral heads on one side. These require separate directional patterns.

Casing Handholes Casing handholes are furnished primarily on pumps handling sewage and stringy materials that may become lodged on the impeller suction vane edges or on the tongue of the volute. The holes permit removal of this material without completely dismantling the pump. End-suction pumps used primarily for liquids of this type are provided with handholes or access to the suction side of the impellers. These access points are located on the suction head or in the suction elbow. Handholes are also provided in drainage, irrigation, circulating, and supply pumps if foreign matter may become lodged in the waterways. On very large pumps, manholes provide access to the interior for both cleaning and inspection.

Mechanical Features of Casings Most single-stage centrifugal pumps are intended for service at moderate pressures and temperatures. As a result, pump manufacturers usually design a special line of pumps for high operating pressures and temperatures rather than make their standard line unduly expensive by making it suitable for too wide a range of operating conditions.

If axially split casings are subject to high pressure, they tend to “breathe” at the split joint, leading to misalignment of the rotor and, even worse, leakage. For such conditions, internal and external ribbing is applied to casings at the points subject to the greatest stress. In addition, whereas most pumps are supported by feet at the bottom of the casing, high temperatures require centerline support so that, as the pump becomes heated, expansion will not cause misalignment.

Series Units For large-capacity medium-high-head conditions, two single-stage, double-suction pumps can be connected in a series on one baseplate with a single driver. Such an arrangement was at one time very common in waterworks applications for heads of 250 to 400 ft (76 to 122 m). One series arrangement uses a double-extended shaft motor in the middle, driving two pumps connected in a series by external piping. In a second type, a standard motor is used with one pump having a double-extended shaft. This latter arrangement may have limited applications because the shaft of the pump next to the motor must be strong enough to transmit the total pumping horsepower. If the total pressure generated by such a series unit is relatively high, the casing of the second pump could require ribbing. Higher heads per stage are becoming more and more common, and series units are generally used in only very high ranges of total head.

MULTISTAGE PUMP CASINGS

Although the majority of single-stage pumps are of the volute-casing type, both volute and diffuser casings are used in multistage pump construction. Because a volute casing gives rise to radial thrust, axially split multistage casings generally have staggered volutes so that the resultant of the individual radial thrusts is balanced out (see Figures 16 and 17). Both axially and radially split casings are used for multistage pumps. The choice between the two designs is dictated by the design pressure, with 2000 lb/in^2 (138 bar)° being typical for 3600-rpm axially split casing pumps.

°1 bar = 10^5 Pa

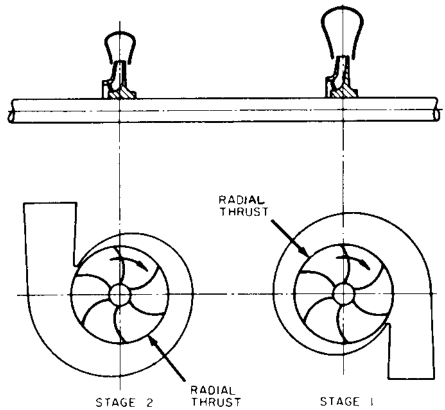


FIGURE 16 An arrangement of a multistage volute pump for radial-thrust balance

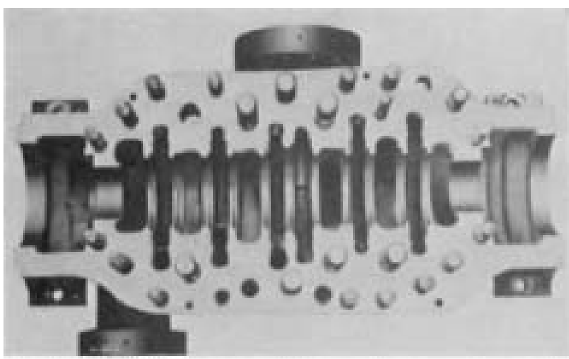


FIGURE 17 The horizontal flange of an axially split casing, six-stage pump (Flowserve Corporation)

Axially Split Casings Regardless of the arrangement of the stages in the casing, it is necessary to connect the successive stages of a multistage pump. In the low and medium pressure and capacity ranges, these interstage passages are cast integrally with the casing proper (see Figures 18 and 19). As the pressures and capacities increase, the desire to maintain as small a casing diameter as possible, coupled with the necessity of avoiding sudden changes in the velocity or the direction of the flow, leads to the use of external interstage passages cast separately from the pump casing. They are formed in the shape of a loop, bolted or welded to the casing proper (see Figure 20).

Interstage Construction for Axially Split Casing Pumps A multistage pump inherently has adjoining chambers subjected to different pressures, and means must be made available to isolate these chambers from one another so that the leakage from high to low pressure will take place only at the clearance joints formed between the stationary and rotating elements of the pump. Thus, the leakage will be kept to a minimum. The isolating wall used to separate two adjacent chambers of a multistage pump is called a *stage piece*, *diaphragm*, or *inter-stage diaphragm*. The stage piece may be formed of a single piece, or it may be fitted with a renewable stage piece bushing at the clearance joint between the stationary stage piece and the part of the rotor immediately inside the former. The stage pieces, which are usually solid, are assembled on the rotor along with impellers, sleeves, bearings,

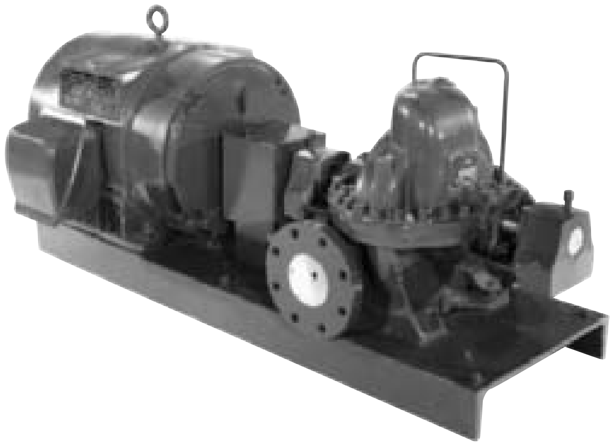


FIGURE 18 Two-stage axially split casing volute pump for small capacities and pressures up to 450 lb/in² (31 bar) (Flowserve Corporation)

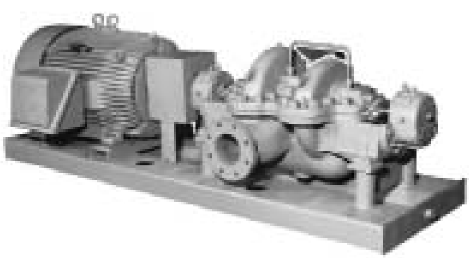


FIGURE 19 Two-stage axially split casing volute pump for pressures up to 500 lb/in² (34 bar) (Flowserve Corporation

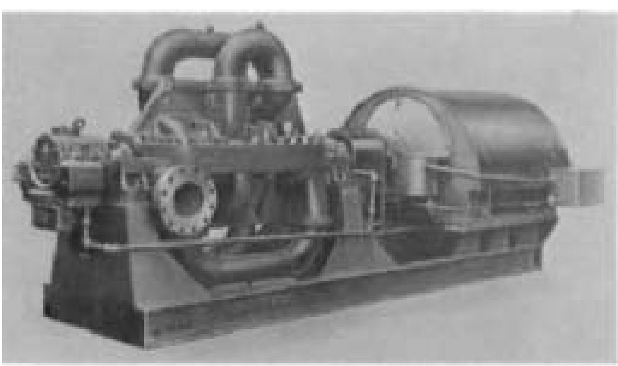


FIGURE 20 A six-stage, axially split casing volute pump for pressures up to 1,300 lb/in² (90 bar) (Flowserve Corporation)

and similar components. To prevent the stage pieces from rotating, a locked tongue-and-groove joint is provided in the lower half of the casing. Clamping the upper casing half to the lower half securely holds the stage piece and prevents rotation.

The problem of seating a solid stage piece against an axially split casing is one that has given designers much trouble. First, there is a three-way joint and, second, this seating must make the joint tight and leakproof under a pressure differential without resorting to bolting the stage piece directly to the casing.

To overcome this problem, it is wise to make a pump that has a small-diameter casing so that when the casing bolting is pulled tight, there is a seal fitting of the two casing halves adjacent to the stage piece. The small diameter likewise helps to eliminate the possibility of a stage piece cocking and thereby leaving a clearance on the upper-half casing when it is pulled down. No matter how rigidly the stage piece is located in the lower-half casing, there must be a sliding fit between the seat face of the stage piece and that in the upper-half casing so that the upper-half casing can be pulled down. Each stage piece, furthermore, must be arranged so that the pressure differential developed by the pump will tend to seat the piece tightly against the casing rather than open up the joint.

We have said that axially split casing pumps are typically used for working pressures of up to 2,000 lb/in^2 (138 bar). High-pressure piping systems, of which these pumps form a part, are inevitably made of steel because this material has the valuable property of yielding without breaking. Considerable piping strains are unavoidable, and these strains, or at least a part thereof, are transmitted to the pump casing. The latter consists essentially of a barrel that is split axially, flanged at the split, and fitted with two necks that serve as inlet and discharge openings. When piping stresses exist, these necks, being the weakest part of the casing, are in danger of breaking off if they cannot yield. Steel is therefore the safest material for pump casings whenever the working pressures in the pump are in excess of 1,000 lb/in^2 (70 bar).

This brings up an important feature in the design of the suction and discharge flanges. Although raised-face flanges are perfectly satisfactory for steel-casing pumps, their use is extremely dangerous with cast-iron pumps. This danger arises from the lack of elasticity in cast iron, which leads to flange breakage when the bolts are being tightened, the fulcrum of the bending moment being located inward from the bolt circle. As a result, it is essential to avoid raised-face flanges with cast-iron casings as well as the use of a raised-face flange pipe directly against a flat-face cast-iron pump flange. Suction flanges should obviously be suitable for whatever hydrostatic test pressure is applied to the pump casing.

The location of the pump casing feet is not critical in smaller pumps operating at discharge pressures below 275 lb/in^2 (19 bar) and at moderate temperatures of up to 300°F (150°C). Since the unit is relatively small, very little distortion is likely to occur. However, for larger units operating at higher pressures and temperatures, it is important that the casing be supported at the horizontal centerline or immediately below the bearings (refer to Figures 19 and 20).

Radially Split Double-Casing Pumps The oldest form of radially split casing multistage pump is commonly called the *ring-section*, *ring-casing*, or the *doughnut type*. When more than one stage was found necessary to generate higher pressures, two or more single-stage units of the prevalent radially split casing type were assembled and bolted together.

In later designs, the individual stage sections and separate suction and discharge heads were held together with large throughbolts. These pumps, still an assembly of bolted-up sections, can present serious dismantling and reassembly problems because suction and discharge connections have to be broken each time the pump is serviced. The double-casing pump retains the advantages of the radially split casing design and minimizes the dismantling problem.

The basic principle consists of enclosing the working parts of a multistage centrifugal pump in an inner casing and building a second casing around this inner casing. The space between the two casings is maintained at the discharge pressure of the last pump stage. The construction of the inner casing follows one of two basic principles: (1) axial splitting (see Figure 21) or (2) radial splitting (see Figure 22).

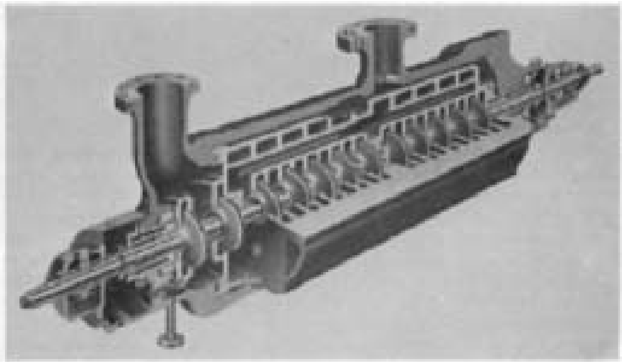


FIGURE 21 Double-casing multistage pump with axially split inner casing (Allis-Chalmers)

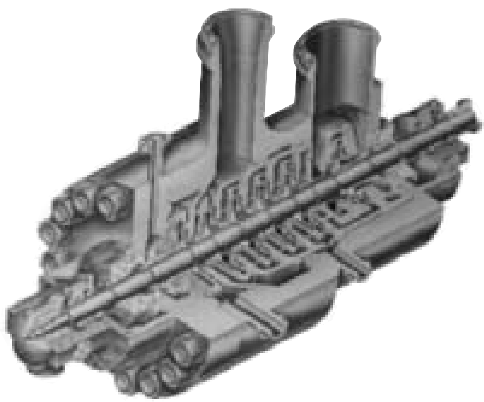


FIGURE 22 Double-casing multistage pump with radially split inner casing (Flowserve Corporation)

The double-casing pump with radially split inner casing is an evolution of the ring-casing pump with added provisions to ease dismantling. The inner unit is generally constructed exactly as a ring-casing pump. After assembly, it is inserted and bolted inside a cylindrical casing that supports it and leaves it free to expand under temperature changes. In Figure 23, the inner assembly of such a pump is being inserted into the outer casing. Figure 24 shows the external appearance of this type of pump. The suction and discharge nozzles form an integral part of the outer casing, and the internal assembly of the pump can be withdrawn without disturbing the piping connections.

Hydrostatic Pressure Tests It is standard practice for the manufacturer to conduct hydrostatic tests for the parts of a pump that contain fluid under pressure. This means that the pump casing and, where applicable, parts like suction or casing covers are assembled with the internal parts, removed, and are then subjected to a hydrostatic test, generally for a minimum of 30 minutes. Such a test demonstrates that the casing containment is sound and that there is no leakage of fluid to the exterior.

Additional tests may be conducted by the pump manufacturer to determine the soundness of internal partitions separating areas of the pump operating under different pressures.

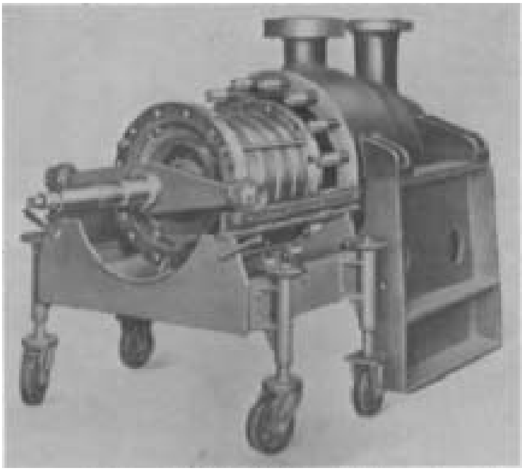


FIGURE 23 Rotor assembly of a radially split, double-casing pump being inserted into its outer casing (Flowserve Corporation)

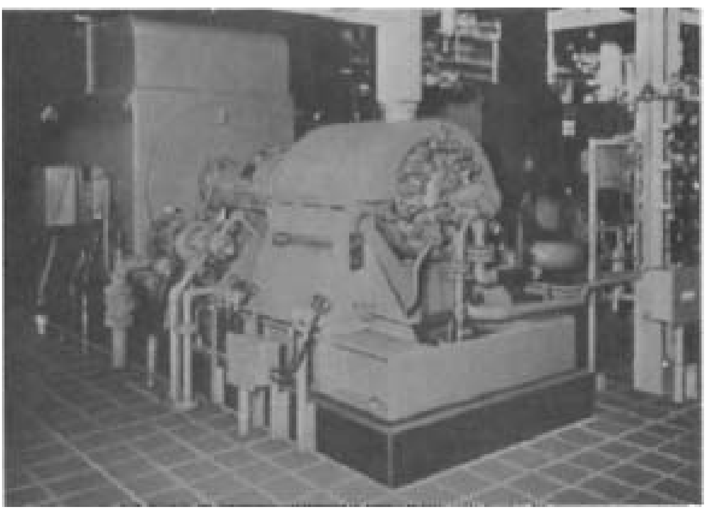


FIGURE 24 Multistage radially split, double-casing pump (Flowserve Corporation)

The definition of the applicable hydrostatic pressure for these tests varies. The most generally accepted definition is that given by the Hydraulic Institute Standards. Each part of the pump that contains fluid under pressure shall be capable of withstanding a hydrostatic test at not less than the greatest of the following:

- 150 percent of the pressure that will occur in that part when the pump is operated at rated conditions for the given application of the pump, except thermoset parts
- 125 percent of the pressure that would occur in that part when the pump is operating at rated speed for a given application, but with the pump discharge valve closed

IMPELLERS

In a single-suction impeller, the liquid enters the suction eye on one side only. A double-suction impeller is, in effect, two single-suction impellers arranged back to back in a single casing. The liquid enters the impeller simultaneously from both sides, while the two casing suction passageways are connected to a common suction passage and a single suction nozzle.

For the general service single-stage, axially split casing design, a double-suction impeller is favored because it is theoretically in an axial hydraulic balance and because the greater suction area of a double-suction impeller permits the pump to operate with less net absolute suction head. For small units, the single-suction impeller is more practical for manufacturing reasons, as the waterways are not divided into two very narrow passages. It is also sometimes preferred for structural reasons. End-suction pumps with single-suction overhung impellers have both first-cost and maintenance advantages unobtainable with double-suction impellers. Most radially split casing pumps therefore use single-suction impellers. Because an overhung impeller does not require the extension of a shaft into the impeller suction eye, single-suction impellers are preferred for pumps handling suspended matter, such as sewage. In multistage pumps, single-suction impellers are almost universally used because of the design and first-cost complexity that double-suction staging introduces.

Impellers are called radial vane or radial flow when the liquid pumped is made to discharge radially to the periphery. Impellers of this type usually have a specific speed below 4200 (2600) if single-suction and below 6000 (3700) if double-suction. Specific speed is discussed in detail in Subsection 2.3.1. The units for specific speed used here are in USCS rpm, gallons per minute, and feet. In SI, they are rpm, liters per second, and meters. Impellers can also be classified by the shape and form of their vanes:

- The straight-vane impeller (see Figures 25, 34, 35, 36, and 37)
- The Francis-vane or screw-vane impeller (see Figures 26 and 27)
- The mixed-flow impeller (see Figure 28)
- The propeller or axial-flow impeller (see Figure 29)

In a *straight-vane radial impeller*, the vane surfaces are generated by straight lines parallel to the axis of rotation. These are also called *single-curvature vanes*. The vane surfaces of a *Francis-vane radial impeller* have a double curvature. An impeller design that has both a radial-flow and an axial-flow component is called a *mixed-flow impeller*. It is generally restricted to single-suction designs with a specific speed above 4200 (2600). Types with lower specific speeds are called *Francis-vane impellers*. Mixed-flow impellers with a small radial-flow component are usually referred to as propellers. In a true propeller, or *axial-flow impeller*, the flow strictly parallels the axis of rotation. In other words, it moves only axially.

An inducer is a low-head, axial-flow impeller with few blades that is placed in front of a conventional impeller. The hydraulic characteristics of an inducer are such that it requires considerably less NPSH than a conventional impeller. Both inducer and impeller

FIGURE 25 Straight-vane, radial, single-suction closed impeller (Flowserve Corporation)

FIGURE 26 Francis-vane, radial, double-suction closed impeller (Flowserve Corporation)

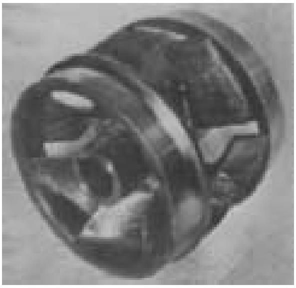


FIGURE 27 High-specific-speed, Francis-vane, radial, double-suction closed impeller (Flowserve Corporation)

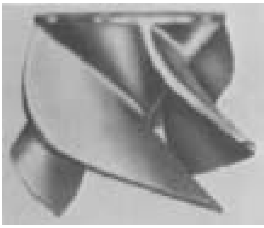


FIGURE 28 Open mixed-flow impeller (Flowserve Corporation)

FIGURE 29 Axial-flow impeller (Flowserve Corporation)

are mounted on the same shaft and rotate at the same speed (see Figure 30). The main purpose of the inducer is not to generate an appreciable portion of the total pump head, but to increase the suction pressure to a conventional impeller. Inducers are therefore used to reduce the NPSH requirements of a given pump or to permit the pump to operate at higher speeds with a given available NPSH. For a further discussion of inducers, see Section 2.1 and Subsection 2.3.1.

The relation of single-suction impeller profiles to specific speed is shown in Figure 31. The classification of impellers according to their vane shape is naturally arbitrary inasmuch as there is much overlapping in the types of impellers used in the different types of pumps. For example, impellers in single- and double-suction pumps of low specific speeds have vanes extending across the suction eye. This provides a mixed flow at the impeller entrance for low pickup losses at high rotative speeds but enables the discharge portion of the impeller to use the straight-vane principle. In pumps of higher specific speed operating against low beads, impellers have double-curvature vanes extending over the full vane surface. They are therefore full Francis-type impellers. The mixed-flow impeller, usually a single-suction type, is essentially one-half of a double-suction, high-specific-speed, Francis-vane impeller.

In addition, many impellers are designed for specific applications. For instance, the conventional impeller design with sharp vane edges and restricted areas is not suitable for handling liquids containing rags, stringy materials, and solids like sewage because it will become clogged. Special nonclogging impellers with blunt edges and large waterways have been developed for such services (see Figure 32). For pumps up to the 12- to 16-in (305- to 406-mm) discharge size, these impellers have only two vanes. Larger pumps normally use three or four vanes.

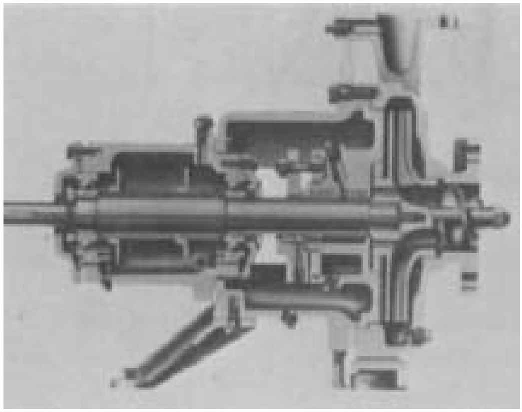


FIGURE 30 A centrifugal pump with a conventional impeller preceded by an inducer (Flowserve Corporation)

Another impeller design used for paper pulp pumps (see Figure 33) is fully open and nonclogging; it has screw and radial streamlined vanes. The screw-conveyor end projects far into the suction nozzle, permitting the pump to handle high-consistency paper pulp stock.

Impeller Mechanical Types Mechanical design also determines impeller classification. Accordingly, impellers may be completely open, semiopen, or closed.

Strictly speaking, an *open impeller* (see Figures 34 and 35) consists of nothing but vanes attached to a central hub for mounting on the shaft without any form of sidewall or shroud. The disadvantage of this impeller is structural weakness. If the vanes are long, they must be strengthened by ribs or a partial shroud. Generally, open impellers are used in small, low energy pumps. One advantage of open impellers is that they are better suited for handling liquids containing stringy materials. It is also sometimes claimed that they are better suited for handling liquids containing suspended matter because the solids in such matter are more likely to be clogged in the space between the rotating shrouds of a closed impeller and the stationary casing walls. It has been demonstrated, however, that closed impellers do not clog easily, thus disproving the claim for the superiority of the open-impeller design. In addition, the open impeller is much more sensitive to wear than the closed impeller and therefore its efficiency may deteriorate rather rapidly.

The open impeller rotates between two side plates, between the casing walls of the volute, or between the casing cover and the suction head. The clearance between the impeller vanes and the sidewalls enables a certain amount of water slippage. This slippage increases as wear increases. To restore the original efficiency, both the impeller and the side plate(s) must be replaced. This, incidentally, involves a much larger expense than would be entailed in closed impeller pumps where simple rings form the leakage joint.

The *semiopen impeller* (see Figure 36) incorporates a single shroud, usually at the back of the impeller. This shroud may or may not have *pump-out vanes*, which are vanes located at the back of the impeller shroud (see Figure 37). This function reduces the pressure at the back hub of the impeller and prevents foreign matter from lodging in back of the impeller that would interfere with the proper operation of the pump and the seal chamber.

The *closed impeller* (refer to Figures 25 through 27), which is almost universally used in centrifugal pumps handling clear liquids, incorporates shrouds or sidewalls that totally enclose the impeller waterways from the suction eye to the periphery. Although this design prevents the liquid slippage that occurs between an open or semiopen impeller and its side plates, a running joint must be provided between the impeller and the casing to separate the discharge and suction chambers of the pump. This running joint is usually formed by a relatively short cylindrical surface on the impeller shroud that rotates within a slightly larger stationary cylindrical surface. If one or both surfaces are made renewable, the leakage joint can be repaired when wear causes excessive leakage.

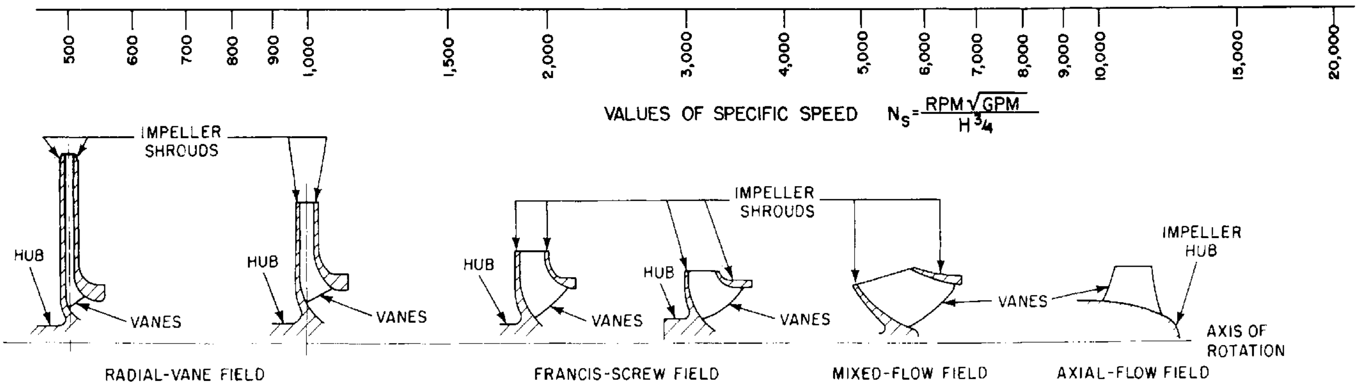


FIGURE 31 Variations in impeller profiles with specific speeds and approximate ranges of specific speeds for the various types. [Universal specific speed $\Omega_s = N_s/2733$. N_q (in rpm, m^3/s, m) $= N_s/51.65$].

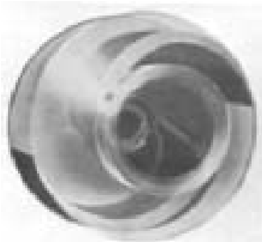


FIGURE 32 Phantom view of a radial-vane nonclogging impeller (Flowserve Corporation)

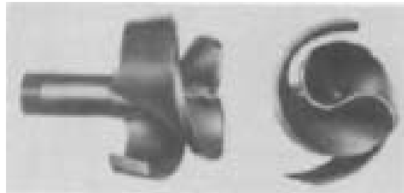


FIGURE 33 Paper plup impeller (Flowserve Corporation)

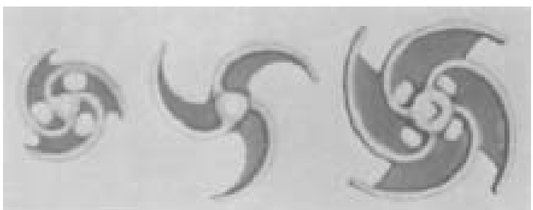


FIGURE 34 Open impellers. Notice that the impellers at left and right are strengthened by a partial shroud (Flowserve Corporation).

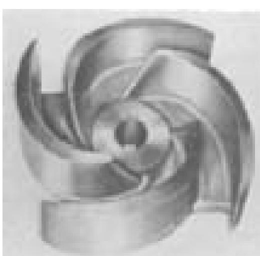


FIGURE 35 An open impeller with partial shroud (Flowserve Corporation)

FIGURE 36 Semiopen impeller (Flowserve Corporation)

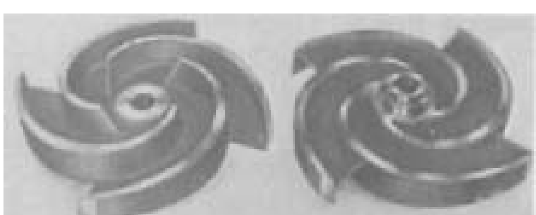


FIGURE 37 The front and back views of an open impeller with a partial shroud and pump-out vanes on the back side (Flowserve Corporation)

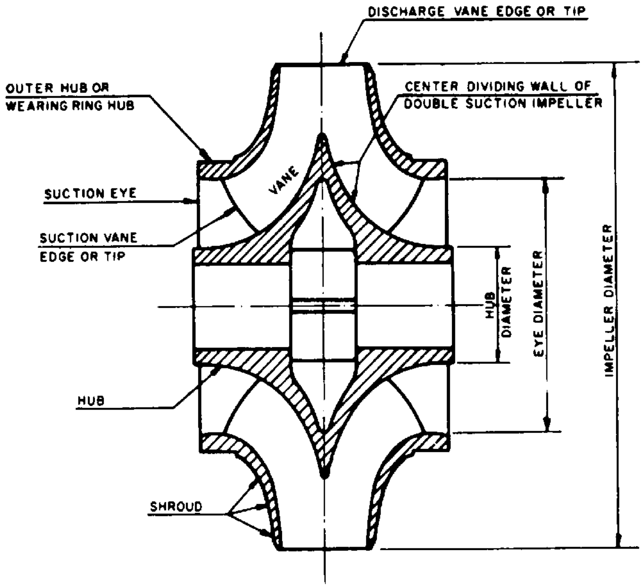


FIGURE 38 Parts of a double-suction impeller

If the pump shaft terminates at the impeller so that the latter is supported by bearings on one side, the impeller is called an *overhung impeller*. This type of construction is the best for end-suction pumps with single-suction impellers.

Impeller Nomenclature The inlet of an impeller just before the section where the vanes begin is called the suction eye (see Figure 38). In a closed-impeller pump, the suction eye diameter is taken as the smallest inside diameter of the shroud. In determining the area of the suction eye, the area occupied by the impeller shaft hub is deducted.

The *hub* is the central part of the impeller that is bored out to receive the pump shaft. The term, however, is also frequently used for the part of the impeller that rotates in the casing fit or in the casing wearing ring. It is then referred to as the *outer impeller hub* or the *wearing-ring hub* of the shroud.

WEARING RINGS

Wearing rings provide an easily and economically renewable leakage joint between the impeller and the casing. A leakage joint without renewable parts is illustrated in Figure 39. To restore the original clearances of such a joint after wear occurs, the user must either (1) build up the worn surfaces by welding or metal spraying, or (2) buy new parts.

The new parts are not very costly in small pumps, especially if the stationary casing element is a simple suction cover. This is not true for larger pumps or where the stationary element of the leakage joint is part of a complicated casting. If the first cost of a pump is of prime importance, it is more economical to provide for remachining both the stationary parts and the impeller. Renewable casing and impeller rings can then be installed (see Figures 40, 41, and 42). The nomenclature for the casing or stationary part forming the leakage joint surface is as follows: (1) *casing ring* (if mounted in the casing), (2) *casing ring* or *suction head ring* (if mounted in a suction cover or head), and (3) *casing cover ring* or *head ring* (if mounted in the casing cover or head). Some engineers like to identify the part

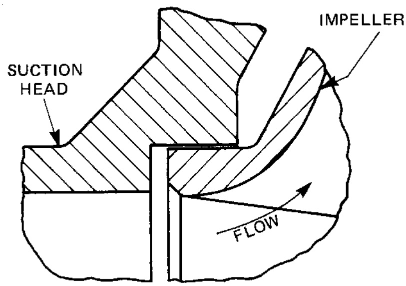


FIGURE 39 A plain flat leakage joint with no rings

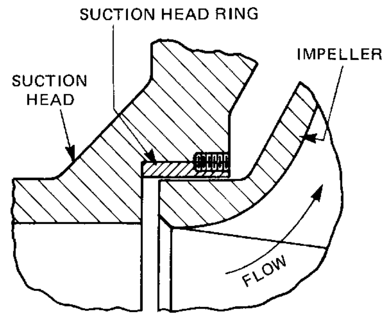


FIGURE 40 A single flat casing ring construction.

further by adding the word *wearing*, such as *casing wearing ring*. A renewable part for the impeller wearing surface is called the *impeller ring*. Pumps with both stationary and rotating rings are said to have *double-ring construction*.

Wearing Ring Types Various types of wearing ring designs, and the selection of the most desirable type depends on the liquid being handled, the pressure differential across the leakage joint, the rubbing speed, and the particular pump design. In general, centrifugal pump designers use the ring construction that they have found to be most suitable for each particular pump service.

The most common ring constructions are the flat type (see Figures 40 and 41) and the L type. The leakage joint in the former is a straight, annular clearance. In the L-type ring (see Figure 43), the axial clearance between the impeller and the casing ring is large, so the velocity of the liquid flowing into the stream entering the suction eye of the impeller is low. The L-type casing rings shown in Figures 43 and 44 have the additional function of guiding the liquid into the impeller eye; they are called *nozzle rings*. Impeller rings of the L type shown in Figure 44 also furnish protection for the face of the impeller wearing ring hub.

Some designers favor labyrinth-type rings (see Figures 45 and 46) that have two or more annular leakage joints connected by relief chambers. In leakage joints involving a single unbroken path, the flow is a function of both the area and the length of the joint as well as of the pressure differential across the joint. If the path is broken by relief chambers (see to Figure 42, 45, and 46), the velocity energy in the jet is dissipated in each relief chamber, increasing the resistance. As a result, with several relief chambers and several leakage joints for the same actual flow through the joint, the area and hence the clearance between the rings can be greater than for an unbroken, shorter leakage joint.

The single labyrinth ring with only one relief chamber (refer to Figure 45) is often called an *intermeshing ring.* The *step-ring type* (refer to Figure 42) utilizes two flat-ring elements of slightly different diameters over the total leakage joint width with a relief chamber between the two elements. Other ring designs also use some form of relief chamber. For example, one commonly used in small pumps has a flat joint similar to that in Figure 40, but with one surface broken by a number of grooves. These act as relief chambers to dissipate the jet velocity head, thereby increasing the resistance through the joint and decreasing the leakage.

For raw water pumps in waterworks service and for larger pumps in sewage services in which the liquid contains sand and grit, *water-flushed rings* have been used (see Figure 47). Clear water under a pressure greater than that on the discharge side of the rings is piped to the inlet and distributed by the cored passage, the holes through the stationary ring, and the groove to the leakage joint. Ideally, the clear water should fill the leakage joint with some flow to the suction and discharge sides to prevent any sand or grit from getting into the clearance space. Wearing-ring flush is also employed in some process pumps when pumping solids or abrasives to minimize ring wear by injecting a compatible clean liquid between the rings.

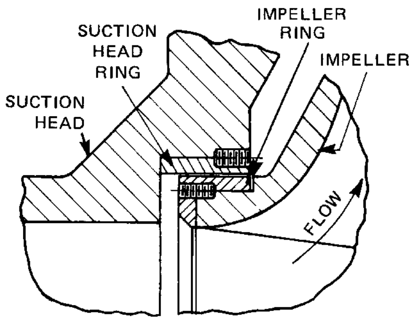


FIGURE 41 A double flat ring construction

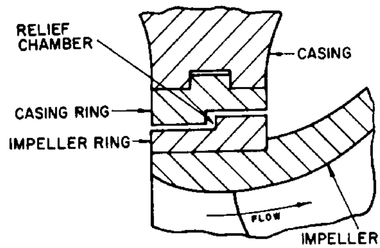


FIGURE 42 A step-type leakage joint with double rings

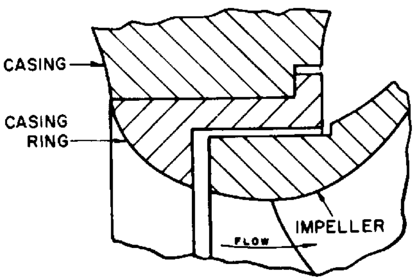


FIGURE 43 An L-type nozzle casing ring

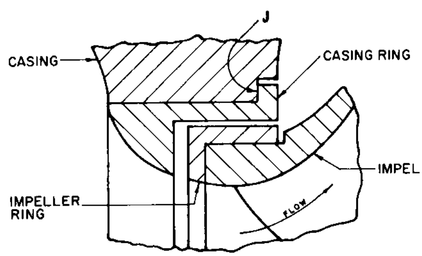


FIGURE 44 Double rings, both of L type

In large pumps (with roughly a 36-in [900-mm] or larger discharge size), particularly vertical, end-suction, single-stage volute pumps, size alone permits some refinements not found in smaller pumps. One example is the inclusion of inspection ports for measuring ring clearance (see Figure 48). These ports can be used to check the impeller centering after the original installation as well as to observe ring wear without dismantling the pump.

The lower rings of large vertical pumps handling liquids containing sand and grit in intermittent services are highly susceptible to wear. During shutdown periods, the grit and sand settle out and naturally accumulate in the region where these rings are installed, as it is the lowest point on the discharge side of the pump. When the pump is started again, this foreign matter is washed into the joint all at once and causes wear. To prevent this action in medium and large pumps, a dam-type ring is often used (see Figure 49). Periodically, the pocket on the discharge side of the dam can be flushed out.

One problem with the simple water-flushed ring is the failure to get uniform pressure in the stationary ring groove. If pump size and design permit, two sets of wearing rings arranged in tandem and separated by a large water space (see Figure 50) provide the best solution. The large water space enables a uniform distribution of the flushing water to the full 360° degrees of each leakage joint. Because ring 2 is shorter and because a greater clearance is used there than at ring 1, equal flow can take place to the discharge pressure side and to the suction pressure side. This design also makes it easier to harden or coat the surfaces.

For pumps handling gritty or sandy water, the ring construction should provide an apron on which the stream leaving the leakage joint can impinge, as sand or grit in the jet will erode any surface it hits. Thus, a form of L-type casing ring similar to that shown in Figure 49 should be used.

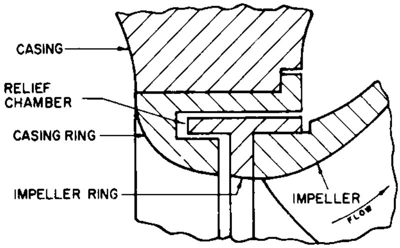


FIGURE 45 A single-labyrinth, intermeshing type. It is a double-ring construction with a nozzle-type casing ring.

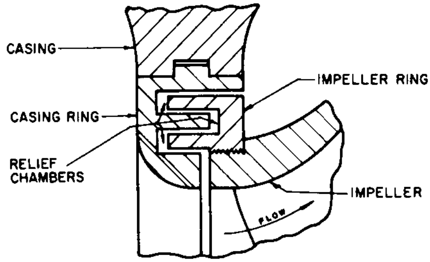


FIGURE 46 Labyrinth-type rings in a double-ring construction

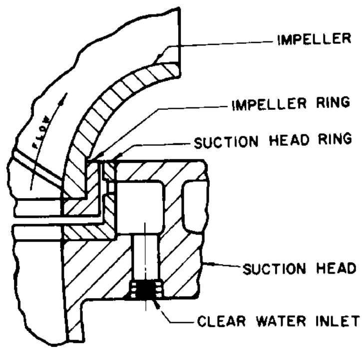


FIGURE 47 Water-flushed wearing ring

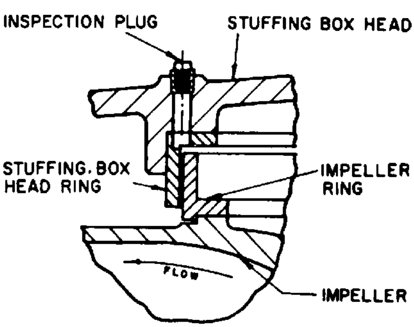


FIGURE 48 Wearing ring with an inspection port for checking clearance

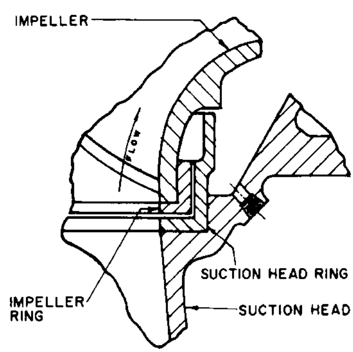


FIGURE 49 Dam-type ring construction

Wearing-Ring Location In some designs, leakage is controlled by an axial clearance (see Figure 51). Generally, this design requires a means of adjusting the shaft position for proper clearance. Then, if uniform wear occurs over the two surfaces, the original clearance can be restored by adjusting the position of the impeller. A limit exists to the amount of wear that can be compensated, because the impeller must be nearly central in the casing waterways.

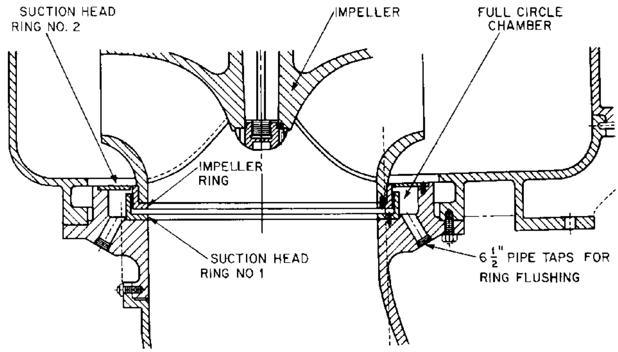


FIGURE 50 Two sets of rings with space between for flushing water (Flowserve Corporation)

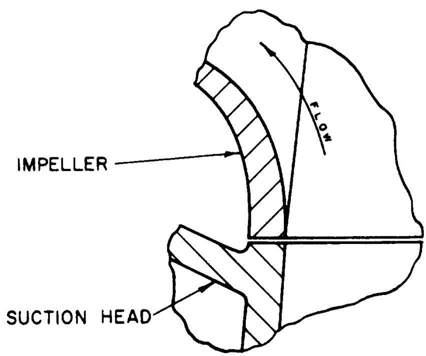


FIGURE 51 A leakage joint with axial clearance

Leakage joints with axial clearance are not popular for double-suction pumps because a very close tolerance is required in machining the fit of the rings in reference to the centerline of the volute waterways. Joints with radial clearances, however, enable some shifting of the impeller for centering. The only adverse effect is a slight inequality in the lengths of the leakage paths on the two sides.

So far, this discussion has treated only those leakage joints located adjacent to the impeller eye or at the smallest outside shroud diameter. Designs have been made where the leakage joint is at the periphery of the impeller. In a vertical pump, this design is advantageous because the space between the joint and the suction waterways is open and so sand or grit cannot collect. Because of rubbing speed and because the impeller diameters used in the same casing vary over a wide range, the design is impractical in regular pump lines.

Mounting Stationary Wearing Rings In small single-suction pumps with suction heads, a stationary wearing ring is usually pressed into a bore in the head and may or may not be further locked by several set screws located half in the head and half in the ring (refer to Figure 41). Larger pumps often use an L-type ring with the flange held against a face on the head. In axially split casing pumps, the cylindrical casing bore (in which the casing ring will be mounted) should be slightly larger than the outside diameter of the ring. Unless some clearance is provided, distortion of the ring may occur when the two casing halves are assembled. However, the joint between the casing ring and the casing must be tight enough to prevent leakage. This is usually provided by a radial metal-to-metal joint (refer to Figure 43) arranged so that the discharge pressure presses the ring against the casing surface.

As it is not desirable for the casing ring of an axially split casing pump to be pinched by the casing, the ring will not be held tightly enough to prevent its rotation unless special provisions are made to keep it in place. One means of accomplishing this is to place a pin in the casing that will project into a hole bored in the ring or, conversely, to provide a pin in the ring that will fit into a hole bored in the casing or into a recess at the casing split joint.

Another method is to have a tongue on the casing ring that extends around 180° and engages a corresponding groove in one-half of the casing. This method can be used with casing rings having a central flange by making the diameter of the flange larger for 180° and cutting a deeper groove in that half of the casing.

Many methods are used for holding impeller rings on the impeller. Probably the simplest is to rely on a press fit of the ring on the impeller or, if the ring is of proper material, on a shrink fit. Designers do not usually feel that a press fit is sufficient and often add several machine screws or set screws located half in the ring and half in the impeller, as in Figure 41.

An alternative to axial machine screws being located half in the ring and half in the impeller is the radial pin, which goes through the center of the ring into the impeller (or from the inside of the impeller eye outward into the ring). This pinning method avoids having to drill and tap holes half in a hardened wearing ring and half in a softer impeller hub. For higher speeds, installing the radial pins from the inside of the impeller eye outward into the wearing ring captures the pins and protects against pin loss, especially at higher operating speeds. Generally, some additional locking method is used rather than relying solely on friction between the ring and the impeller.

In the design of impeller rings, consideration has to be given to the stretch of the ring caused by centrifugal force, especially if the pump is of a high-speed design for the capacity involved. For example, some pumps operate at speeds that would cause the rings to become loose if only a press fit is used. For such pumps, shrink fits should be used or, preferably, impeller rings should be eliminated.

Wearing-Ring Clearances Typical clearance and tolerance standards for nongalling wearing-joint metals in general service pumps are shown in Figure 52. They apply to the

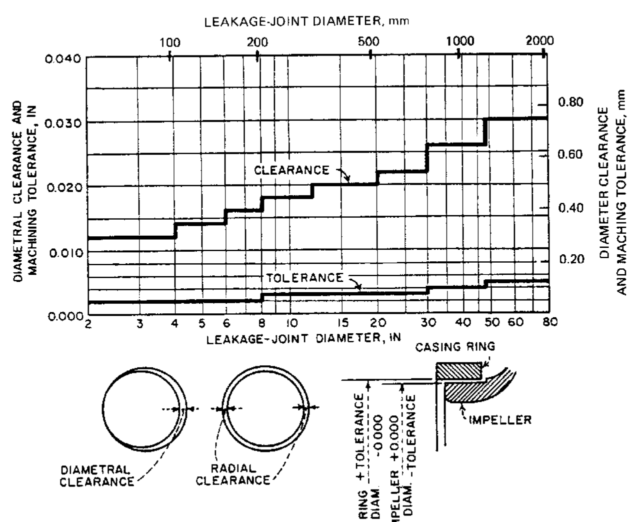


FIGURE 52 Wearing-ring clearances for single-stage pumps using nongalling materials

following combinations: (1) bronze with a dissimilar bronze, (2) cast-iron with bronze, (3) steel with bronze, (4) Monel metal with bronze, and (5) cast-iron with cast iron. If the metals gall easily (like the chrome steels), the values given should be increased by 0.002 to 0.004 in.

The tolerance indicated is positive for the casing ring and negative for the impeller hub or impeller ring.

In a single-stage pump with a joint of nongalling components, the correct machining dimension for a casing-ring diameter of 9.000 in would be 9.000 plus 0.003 and minus 0.000 in. For the impeller hub or ring, the values would be 9.000 minus 0.018 (or 8.982) plus 0.000 and minus 0.003 in. Diametral clearances would be between 0.018 and 0.024 in. Obviously, clearances and tolerances are not translated into SI units by merely using conversion multipliers; they are rounded off as they are in the USCS system. The SI values for the previous USCS examples given are outlined here.

For a single-stage pump with a casing ring diameter of 230 millimeters, the machining dimensions would be as follows:

Casing ring	230 plus 0.08 and minus 0.00 mm
Impeller hub	230 minus 0.50 (229.5) plus 0.00 and minus 0.08 mm
Diametral clearances	Between 0.50 and 0.66 mm

Naturally, the manufacturer's recommendation for ring clearances and tolerances should be followed.

AXIAL THRUST

Axial Thrust in Single-Stage Pumps with Closed Impellers The pressures generated by a centrifugal pump exert forces on both stationary and rotating parts. The design of these parts balances some of these forces, but separate means may be required to counterbalance others.

An axial hydraulic thrust on an impeller is the sum of the unbalanced forces acting in the axial direction. Because reliable, large-capacity thrust bearings are readily available, an axial thrust in single-stage pumps remains a problem only in larger, higher speed units. Theoretically, a double-suction impeller is in hydraulic axial balance, with the pressures on one side equal to and counterbalancing the pressures on the other (see Figure 53). In practice, this balance may not be achieved for the following reasons:

1. The suction passages to the two suction eyes may not provide equal or uniform flows to the two sides.
2. External conditions, such as an elbow located too close to the pump suction nozzle, may cause unequal flow to the two suction eyes.

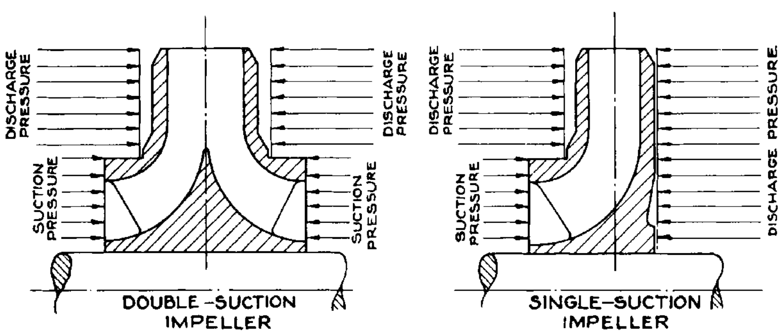


FIGURE 53 The origin of pressures acting on impeller shrouds to produce an axial thrust

3. The two sides of the discharge casing waterways may not be symmetrical, or the impeller may be located off-center. These conditions will alter the flow characteristics between the impeller shrouds and the casing, causing unequal pressures on the shrouds.
4. Unequal leakage through the two leakage joints can upset the balance.

Combined, these factors can create an axial unbalance. To compensate for this, all centrifugal pumps, even those with double-suction impellers, incorporate thrust bearings.

The ordinary single-suction, closed, radial-flow impeller with the shaft passing through the impeller eye (refer to Figure 53) is subject to an axial thrust because a portion of the front wall is exposed to suction pressure and thus relatively more backwall surface is exposed to discharge pressure. If the discharge chamber pressure is uniform over the entire impeller surface, the axial force acting toward the suction would be equal to the product of the net pressure generated by the impeller and the unbalanced annular area.

Actually, pressure on the two single-suction, closed impeller walls is not uniform. The liquid trapped between the impeller shrouds and casing walls is in rotation, and the pressure at the impeller periphery is appreciably higher than the impeller hub. Although we need not be concerned with the theoretical calculations for this pressure variation, Figure 54 describes it qualitatively. Generally speaking, the axial thrust toward the impeller suction may be about 20 to 30 percent less than the product of the net pressure and the unbalanced area.

To eliminate the axial thrust of a single-suction impeller, a pump can be provided with both front and back wearing rings. To equalize the thrust areas, the inner diameter of both rings is made the same (see Figure 55). Pressure approximately equal to the suction pressure is maintained in a chamber located on the impeller side of the back wearing ring by

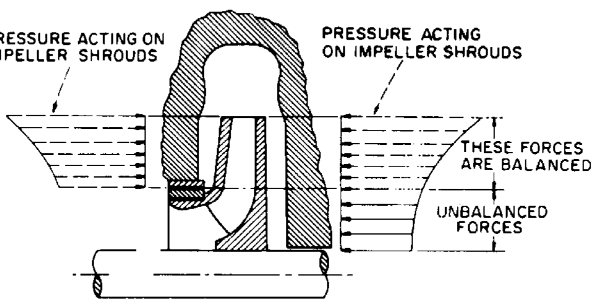


FIGURE 54 Pressure distribution on the front and back shrouds of the single-suction impeller with a shaft through the impeller eye

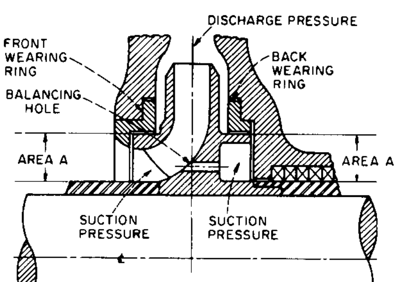


FIGURE 55 Balancing the axial thrust of a single-suction impeller by means of wearing rings on the back side and balancing holes

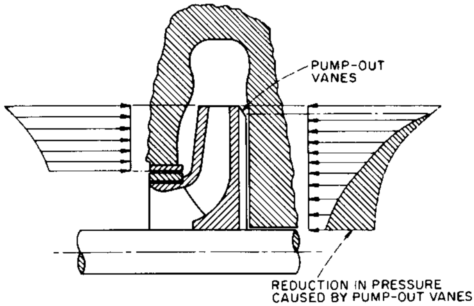


FIGURE 56 Pump-out vanes used in a single-suction impeller to reduce axial thrust

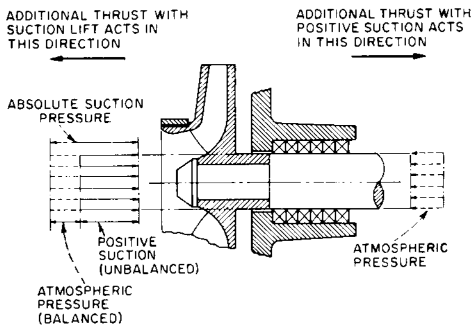


FIGURE 57 An axial thrust problem with a single-suction, overhung impeller and a single stuffing box or seal chamber

drilling balancing holes through the impeller. Leakage past the back wearing ring is returned to the suction area through these holes. However, with large single-stage, single-suction pumps, balancing holes are considered undesirable because leakage back to the impeller suction opposes the main flow, creating disturbances. In such pumps, a piped connection to the pump suction replaces the balancing holes.

Another way to eliminate or reduce an axial thrust in single-suction impellers is to use pump-out vanes on the back shroud. The effect of these vanes is to reduce the pressure acting on the back shroud of the impeller (see Figure 56). This design, however, is generally used only in pumps handling dirty liquids where it keeps the clearance space between the impeller back shroud and the casing free of foreign matter.

So far, our discussion of axial thrust has been limited to single-suction, closed impellers with a shaft passing through the impeller eye that are located in pumps with two seal chambers, one on either side of the impeller. In these pumps, suction-pressure magnitude does not affect the resulting axial thrust.

On the other hand, axial forces acting on an overhung impeller with a single seal chamber (see Figure 57) are definitely affected by suction pressure. In addition to the unbalanced force found in a single-suction, two-seal chamber design (refer to Figure 54), there is an axial force equivalent to the product of the shaft area through the seal chamber and the difference between suction and atmospheric pressure. This force acts toward the impeller suction when the suction pressure is less than atmospheric or in the opposite direction when it is higher than atmospheric.

When an overhung impeller pump handles a suction lift, the additional axial force is very low. For example, if the shaft diameter through the stuffing box is 2 in (50.8 mm) (area = 3.14 in^2 or 20.26 cm^2), and if the suction lift is 20 ft (6.1 m) of water equivalent to an absolute pressure of 6.06 lb/in^2 (0.42 bar abs), the axial force caused by the overhung impeller and acting toward the suction will be only 27 lb (121 N). On the other hand, if the suction pressure is 100 lb/in^2 (6.89 bar), the force will be 314 lb (1405 N) and will act in the opposite direction. Therefore, because the same pump may be used under many conditions over a wide range of suction pressures, the thrust bearing of pumps with single-suction, overhung impellers must be arranged to take the thrust in either direction. They must also be selected with a sufficient thrust capacity to counteract forces set up under the maximum suction pressure established for that particular pump.

This extra thrust capacity can become quite significant in certain special cases, such as with boiler circulating pumps. These are usually of the single-suction, single-stage, overhung impeller type and may be exposed to suction pressures as high as 2800 lb/in^2 gage (193 bar). This is approximately the vapor pressure of 685°F (363°C) boiler water. If such a pump has a shaft diameter of 6-in (15.24 cm), the unbalanced thrust would be as much as 77,500 lb (346,770 N), and the thrust bearing has to be capable of counteracting this.

Axial Thrust of Single-Suction Semiopen Radial-Flow Impellers The axial thrust generated in semiopen impellers is higher than that in closed impellers. This is illustrated in Figure 58, which shows that the pressure on the open side of the impeller varies from essentially the discharge pressure at the periphery (at diameter D_2) to the suction pressure at the impeller eye (at diameter D_1). The pressure distribution on the back shroud is essentially the same as that illustrated in Figure 54, varying from discharge pressure at the periphery to some portion of this pressure at the impeller hub. This latter pressure is, of course, substantially higher than the suction pressure. The unbalanced portion of the axial thrust on the impeller is represented by the cross-hatched area in Figure 58.

One of the means available for partially balancing this increased axial thrust is to provide the back shroud with pump-out vanes, as in Figures 37 and 56.

Fully open impellers or semiopen impellers with a portion of the back shroud removed produce an axial thrust somewhat higher than closed impellers and somewhat lower than semiopen impellers.

Axial Thrust of Mixed-Flow and AxiaL-Flow Impellers The axial thrust in radial impellers is produced by the static pressures on the impeller shrouds. Axial-flow impellers have no shrouds, and the axial thrust is created strictly by the difference in pressure on the two faces of the impeller vanes. In addition, a difference may exist in pressure acting on the two shaft hub ends, one generally subject to discharge pressure and the other to suction pressure.

With mixed-flow impellers, axial thrust is a combination of forces caused by the action of the vanes on the liquid and those arising from the difference in the pressures acting on the various surfaces. Wearing rings are often provided on the back of mixed-flow impellers, with either balancing holes through the impeller hub or an external balancing pipe leading back to the suction.

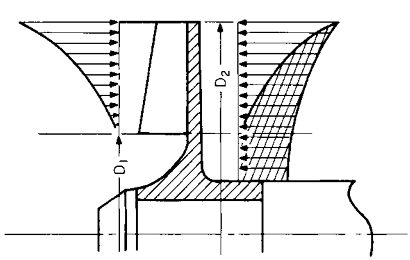


FIGURE 58 Axial thrust in a semiopen single-suction impeller

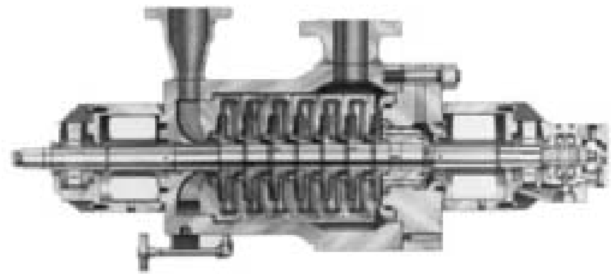


FIGURE 59 Multistage pump with single-suction impellers facing in one direction and hydraulic balancing device (Flowserve Corporation)

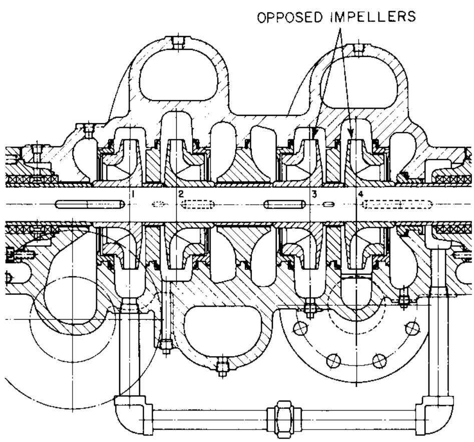


FIGURE 60 A four-stage pump with opposed impellers (Flowserve Corporation)

Except for very large units and in certain special applications, the axial thrust developed by mixed-flow and axial-flow impellers is carried by thrust bearings with the necessary load capacity.

Axial-Thrust in Multistage Pumps Most multistage pumps are built with single-suction impellers in order to simplify the design of the interstage connections. Two obvious arrangements are possible for the single-suction impellers:

1. Several single-suction impellers can be mounted on one shaft, each having its suction inlet facing in the same direction and its stages following one another in ascending order of pressure (see Figure 59). The axial thrust is then balanced by a hydraulic balancing device.
2. An even number of single-suction impellers can be used, one-half facing in one direction and the other half facing in the opposite direction. With this arrangement, an axial thrust on the first half is compensated by the thrust in the opposite direction on the other half (see Figure 60). This mounting of single-suction impellers back to back is frequently called *opposed impellers.*

An uneven number of single-suction impellers can be used with this arrangement, provided the correct shaft and interstage bushing diameters are used to give the effect of a hydraulic balancing device that will compensate for the hydraulic thrust on one of the stages.

It is important to note that the opposed impeller arrangement completely balances an axial thrust only under the following conditions:

- The pump must be provided with two seal chambers.
- The shaft must have a constant diameter.
- The impeller hubs must not extend through the interstage portion of the casing separating adjacent stages.

Except for some special pumps that have an internal and enclosed bearing at one end, and therefore only one seal chamber, most multistage pumps fulfill the first condition. Because of structural requirements, however, the last two conditions are not practical. A slight residual thrust is usually present in multistage opposed impeller pumps and is carried on the thrust bearing.

HYDRAULIC BALANCING DEVICES

If all the single-suction impellers of a multistage pump face the same direction, the total theoretical hydraulic axial thrust acting toward the suction end of the pump will be the sum of the individual impeller thrusts. The thrust magnitude will be approximately equal to the product of the net pump pressure and the annular unbalanced area. Actually, the axial thrust turns out to be about 70 to 80 percent of this theoretical value.

Some form of hydraulic balancing device must be used to balance this axial thrust and to reduce the pressure on the seal chamber adjacent to the last-stage impeller. This hydraulic balancing device may be a balancing drum, a balancing disk, or a combination of the two.

Balancing Drums The balancing drum is illustrated in Figure 61. The balancing chamber at the back of the last-stage impeller is separated from the pump interior by a drum that is usually keyed to the shaft and rotates with it. The drum is separated by a small radial clearance from the stationary portion of the balancing device, called the *balancing-drum head*, or balancing sleeve, which is fixed to the pump casing.

The balancing chamber is connected either to the pump suction or to the vessel from which the pump takes its suction. Thus, the back pressure in the balancing chamber is only slightly higher than the suction pressure, the difference between the two being equal to the friction losses between this chamber and the point of return. The leakage between

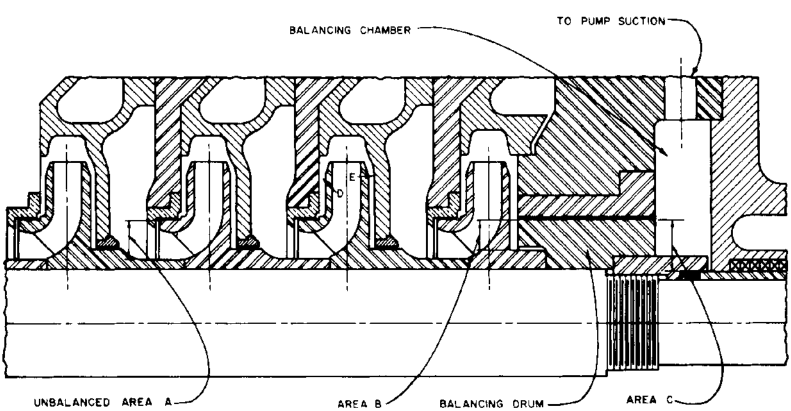


FIGURE 61 Balancing drum

the drum and the drum head is, of course, a function of the differential pressure across the drum and of the clearance area.

The forces acting on the balancing drum in Figure 61 are the following:

- *Toward the discharge end*: the discharge pressure multiplied by the front balancing area (area *B)* of the drum
- *Toward the suction end*: the back pressure in the balancing chamber multiplied by the back balancing area (area *C)* of the drum

The first force is greater than the second, thereby counterbalancing the axial thrust exerted upon the single-suction impellers. The drum diameter can be selected to balance the axial thrust completely or within 90 to 95 percent, depending on the desirability of carrying any thrust-bearing loads.

It has been assumed in the preceding simplified description that the pressure acting on the impeller walls is constant over their entire surface and that the axial thrust is equal to the product of the total net pressure generated and the unbalanced area. Actually, this pressure varies somewhat in the radial direction because of the centrifugal force exerted upon the liquid by the outer impeller shroud (refer to Figure 54). Furthermore, the pressures at two corresponding points on the opposite impeller faces *(D* and *E* in Figure 61) may not be equal because of a variation in clearance between the impeller wall and the casing section separating successive stages. Finally, a pressure distribution over the impeller wall surface may vary with head and capacity operating conditions.

This pressure distribution and design data can be determined quite accurately for any one fixed operating condition, and an effective balancing drum could be designed on the basis of the forces resulting from this pressure distribution. Unfortunately, varying head and capacity conditions change the pressure distribution, and as the area of the balancing drum is necessarily fixed, the equilibrium of the axial forces can be destroyed.

The objection to this is not primarily the amount of the thrust, but rather that the direction of the thrust cannot be predetermined because of the uncertainty about internal pressures. Still it is advisable to predetermine normal thrust direction, as this can influence external mechanical thrust-bearing design. Because 100 percent balance is unattainable in practice and because the slight but predictable unbalance can be carried on a thrust bearing, the balancing drum is often designed to balance only 90 to 95 percent of the total impeller thrust.

The balancing drum satisfactorily balances the axial thrust of single-suction impellers and reduces pressure on the discharge-side stuffing box. It lacks, however, the virtue of automatic compensation for any changes in axial thrust caused by varying impeller reaction characteristics. In effect, if the axial thrust and balancing drum forces become unequal, the rotating element will tend to move in the direction of the greater force. The thrust bearing must then prevent excessive movement of the rotating element. The balancing drum performs no restoring function until such time as the drum force again equals the axial thrust. This automatic compensation is the major feature that differentiates the balancing disk from the balancing drum.

Balancing Disks The operation of the simple balancing disk is illustrated in Figure 62. The disk is fixed to and rotates with the shaft. It is separated by a small axial clearance from the balancing disk head, or balancing sleeve, which is fixed to the casing. The leakage through this clearance flows into the balancing chamber and from there either to the pump suction or to the vessel from which the pump takes its suction. The back of the balancing disk is subject to the balancing chamber back pressure, whereas the disk face experiences a range of pressures. These vary from discharge pressure at its smallest diameter to back pressure at its periphery. The inner and outer disk diameters are chosen so that the difference between the total force acting on the disk face and that acting on its back will balance the impeller axial thrust.

If the axial thrust of the impellers should exceed the thrust acting on the disk during operation, the latter is moved toward the disk head, reducing the axial clearance between

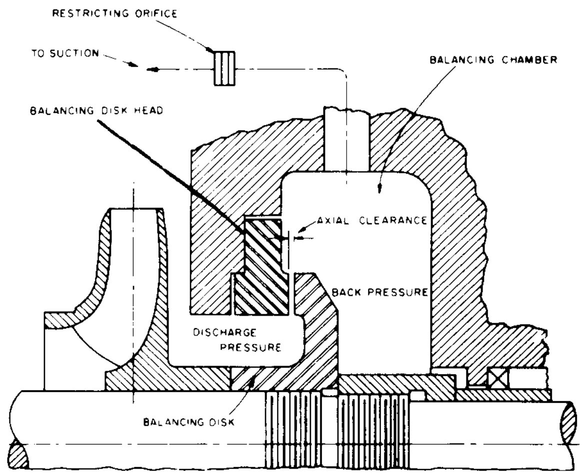


FIGURE 62 A simple balancing disk

the disk and the disk head. The amount of leakage through the clearance is reduced so that the friction losses in the leakage return line are also reduced, lowering the back pressure in the balancing chamber. This lowering of pressure automatically increases the pressure difference acting on the disk and moves it away from the disk head, increasing the clearance. Now the pressure builds up in the balancing chamber, and the disk is again moved toward the disk head until an equilibrium is reached.

To assure proper balancing in disk operation, the change in back pressure in the balancing chamber must be of an appreciable magnitude. Thus, with the balancing disk wide open with respect to the disk head, the back pressure must be substantially higher than the suction pressure to give a resultant force that restores the normal disk position. This can be accomplished by introducing a restricting orifice in the leakage return line that increases back pressure when leakage past the disk increases beyond normal. The disadvantage of this arrangement is that the pressure on the seal chamber is variable, a condition that may be injurious to the life of the seal and therefore should avoided.

Combination Balancing Disk and Drum For the reasons just described, the simple balancing disk is seldom used. The combination balancing disk and drum (see Figure 63) was developed to obviate the shortcomings of the disk while retaining the advantage of automatic compensation for axial thrust changes.

The rotating portion of this balancing device consists of a long cylindrical body that turns within a drum portion of the disk head. This rotating part incorporates a disk similar to the one previously described. In this design, radial clearance remains constant regardless of disk position, whereas the axial clearance varies with the pump rotor position. The following forces act on this device:

- *Toward the discharge end*: the sum of the discharge pressure multiplied by area *A,* plus the average intermediate pressure multiplied by area *B*
- *Toward the suction end*: the back pressure multiplied by area *C*

Whereas the position-restoring feature of the simple balancing disk required an undesirably wide variation of the back pressure, it is now possible to depend upon a variation of the intermediate pressure to achieve the same effect. Here is how it works: When the pump rotor moves toward the suction end (to the left in Figure 63) because of increased axial thrust, the axial clearance is reduced and pressure builds up in the intermediate

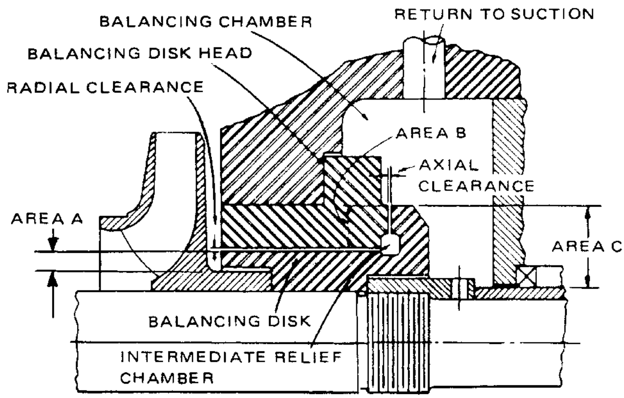


FIGURE 63 A combination balancing disk and drum

relief chamber, increasing the average value of the intermediate pressure acting on area *B*. In other words, with reduced leakage, the pressure drop across the radial clearance decreases, increasing the pressure drop across the axial clearance. The increase in intermediate pressure forces the balancing disk toward the discharge end until equilibrium is reached. Movement of the pump rotor toward the discharge end would have the opposite effect, increasing the axial clearance and the leakage and decreasing the intermediate pressure acting on area *B*.

Now numerous hydraulic balancing device modifications are in use. One typical design separates the drum portion of a combination device into two halves, one preceding and the second following the disk (see Figure 64). The virtue of this arrangement is a definite cushioning effect at the intermediate relief chamber, thus avoiding too positive a restoring action, which might result in the contacting and scoring of the disk faces.

SHAFTS AND SHAFT SLEEVES

The basic function of a centrifugal pump shaft is to transmit the torques encountered when starting and during operation while supporting the impeller and other rotating parts. It must do this job with a deflection less than the minimum clearance between rotating and stationary parts. The loads involved are (1) the torques, (2) the weight of the parts, and (3) both radial and axial hydraulic forces. In designing a shaft, the maximum allowable deflection, the span or overhang, and the location of the loads all have to be considered, as does the critical speed of the resulting design.

Shafts are usually proportioned to withstand the stress set up when a pump is started quickly, such as when the driving motor is energized directly across the line. If the pump handles hot liquids, the shaft is designed to withstand the stress set up when the unit is started cold without any preliminary warmup.

Critical Speeds Any object made of an elastic material has a natural period of vibration. When a pump rotor or shaft rotates at any speed corresponding to its natural frequency, minor unbalances will be magnified. These speeds are called the *critical speeds*.

In conventional pump designs, the rotating assembly is theoretically uniform around the shaft axis and the center of mass should coincide with the axis of rotation. This theory does not hold for two reasons. First, minor machining or casting irregularities always occur. Second, variations exist in the metal density of each part. Thus, even in vertical-shaft machines having no radial deflection caused by the weight of the parts, this eccentricity of the center of mass produces a centrifugal force and therefore a deflection when

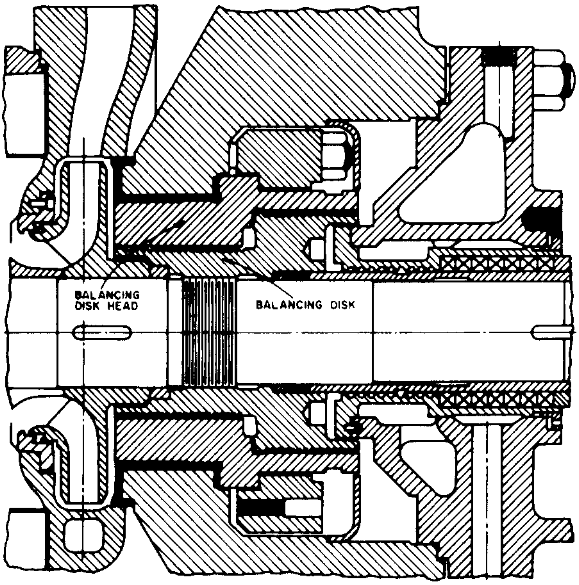


FIGURE 64 A combination balancing disk and drum with a disk located in the center portion of the drum (Flowserve Corporation)

the assembly rotates. At the speed where the centrifugal force exceeds the elastic restoring force, the rotor will vibrate as though it were seriously unbalanced. If it is run at that speed without restraining forces, the deflection will increase until the shaft fails.

Rigid and Flexible Shaft Designs The lowest critical speed is called the first critical speed, the next highest is called the second, and so forth. In centrifugal pump nomenclature, a *rigid shaft* means one with an operating speed lower than its first critical speed. A *flexible shaft* is one with an operating speed higher than its first critical speed. Once an operating speed has been selected, relative shaft dimensions must still be determined. In other words, it must be decided whether the pump will operate above or below the first critical speed.

Actually, the shaft critical speed can be reached and passed without danger because frictional forces tend to restrain the deflection. These forces are exerted by the surrounding liquid, and the various internal leakage joints acting as internal liquid-lubricated bearings. Once the critical speed is passed, the pump will run smoothly again up to the second speed corresponding to the natural rotor frequency, and so on to the third, fourth, and all higher critical speeds.

Designs rated for 1,750 rpm (or lower) are usually of the rigid-shaft type. On the other band, high-head 3,600 rpm (or higher) multistage pumps, such as those in a boiler-feed service, are frequently of the flexible-shaft type. It is possible to operate centrifugal pumps above their critical speeds for the following two reasons: (1) very little time is required to attain full speed from rest (the time required to pass through the critical speed must therefore be extremely short) and (2) the pumped liquid in the internal leakage joints acts as a restraining force on the vibration.

Experience has proved that, although it was usually assumed necessary to use shafts of such rigidity that the first critical speed is at least 20 percent above the operating speed, equally satisfactorily results can be obtained with lighter shafts with a first critical speed of about 60 to 75 percent of the operating speed. This, it is felt, is a sufficient margin to avoid any danger caused by an operation close to the critical speed.

Influence of Shaft Deflection To understand the effect of critical speed on the selection of shaft size, consider the fact that the first critical speed of a shaft is linked to its static deflection. Shaft deflection depends upon the weight of the rotating element (w), the shaft span (l), and the shaft diameter (d). The basic formula is as follows:

$$f = \frac{wl^3}{CEI}$$

where f = deflection, in (m)
w = weight of the rotating element, lb (N)
l = shaft span, in (m)
C = coefficient depending on shaft-support method and load distribution
E = modulus of elasticity of shaft materials, lb/in^2 (N/m^2)
I = moment of inertia ($\pi\, d^4/64$) in^4 (m^4)

This formula is given in its most simplified form, that is, for a shaft of constant diameter. If the shaft is of varying diameter (the usual situation), deflection calculations are much more complex. A graphical deflection analysis may then be the most practical answer.

This formula works only for static deflection, the only variable that affects critical speed calculations. The actual shaft deflection, which must be determined to establish minimum permissible internal clearances, must take into account all transverse hydraulic reactions on the rotor, the weights of the rotating element, and other external loads.

It is not necessary to calculate the exact deflection to make a relative shaft comparison. Instead, a factor can be developed that will be representative of relative shaft deflections. As a significant portion of rotor weight is in the shaft, and as methods of bearing support and the modulus of elasticity are common to similar designs, deflection f can be shown as follows:

$$f = \text{function of } \frac{(ld^2)(l^3)}{d^4}$$

$$f = \text{function of } \frac{l^4}{d^2}$$

In other words, pump deflection varies approximately as the fourth power of the shaft span and inversely as the square of shaft diameter. Therefore, the lower the l^4/d^2 factor for a given pump, the lower the unsupported shaft deflection, essentially in proportion to this factor.

For practical purposes, the first critical speed N_c can be calculated as

in USCS units $$N_c = \frac{187.7}{\sqrt{f}\,(\text{in})} \text{ rpm}$$

in SI units $$N_c = \frac{946}{\sqrt{f}\,(\text{mm})} \text{ rpm}$$

To maintain internal clearances at the wearing rings, it is usually desirable to limit the shaft deflection under the most adverse conditions to between 0.005 and 0.006 in (0.127 and 0.152 mm). It follows that a shaft design with a deflection of 0.005 to 0.006 in will have a first critical speed of 2,400 to 2,650 rpm. This is the reason for using rigid shafts for pumps that operate at 1,750 rpm or lower. Multistage pumps operating at 3,600 rpm or higher use shafts of equal stiffness (for the same purpose of avoiding wearing ring contact). However, their corresponding critical speed is about 25 to 40 percent less than their operating speed.

Lomakin Effect All the previous material refers to the behavior of a rotor and its shaft operating in air. In reality, the rotor operates immersed in the liquid being pumped, and this liquid flows through one or more of the small annular areas created by clearances separating regions in the pump under different pressures, such as at the wearing rings, interstage bushings, or balancing devices. This flow of liquid creates what is called a *hydrodynamic bearing effect* and essentially transforms the rotor from one supported at two bearings external to the pump to one with several additional internal bearings lubricated by the liquid pumped. This phenomenon is generally called the *Lomakin effect*.

The result of the Lomakin effect is that the deflection of the shaft when a pump is running is reduced somewhat from the value calculated for the shaft operating in air and the critical speed is increased. The advantage of this effect, particularly in the design of some multistage pumps, is that it permits the use of longer and more slender shafts. Whether this is sound practice remains a controversial subject. The supportive effect of the hydrodynamic hearings depends on (a) the pressure differential, which disappears completely when the pump is at rest, and (b) the clearance, which decreases substantially as the internal clearances increase with erosive or contact wear. Thus, contact between rotating and stationary parts will take place every time a pump is started if the internal clearances are initially less than the shaft deflection in air. This contact will again take place as the pump coasts down after being stopped. Furthermore, as wear takes place at the running joints, the shaft assumes a deflection closer and closer to its deflection in air, unsupported by the Lomakin effect.

In view of all these facts, it is recommended that pump users acquaint themselves not only with the calculated shaft deflections with a pump running in new condition, but also with the shaft deflections in air. This way they can compare these with the internal clearances.

Shaft Sizing Shaft diameters are usually larger than what is actually needed to transmit the torque. A factor that assures this conservative design is a requirement for ease of rotor assembly.

The shaft diameter must be stepped up several times from the end of the coupling to its center to facilitate impeller mounting (see Figure 65). Starting with the maximum diameter at the impeller mounting, there is a step down for the shaft sleeve and another for the external shaft nut, followed by several more for the bearings and the coupling. Therefore, the shaft diameter at the impellers exceeds that required for torsional strength at the coupling by at least an amount sufficient to provide all intervening step downs.

One frequent exception to shaft oversizing at the impeller occurs in units consisting of two double-suction, single-stage pumps operating in a series, one of which is fitted with a

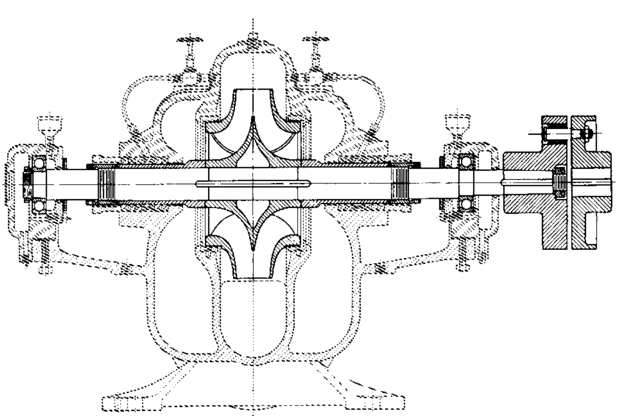


FIGURE 65 Rotor assembly of a single-stage, double-suction pump (Flowserve Corporation)

double-extended shaft. As this pump must transmit the total horsepower for the entire series unit, the shaft diameter at its inboard bearing may have to be larger than normal.

The shaft design of end-suction, overhung impeller pumps presents a somewhat different problem. One method for reducing shaft deflection at the impeller and seal chamber, where the concentricity of running fits is extremely important, is to considerably increase the shaft diameter between the bearings.

Except in certain smaller sizes, centrifugal pump shafts are protected against wear, erosion, and corrosion by renewable shaft sleeves. In small pumps, however, shaft sleeves present a certain disadvantage. As the sleeve cannot appreciably contribute to shaft strength, the shaft itself must be designed for the full maximum stress. Shaft diameter is then materially increased by the addition of the sleeve, as the sleeve thickness cannot be decreased beyond a certain safe minimum. The impeller suction area may therefore become dangerously reduced, and if the eye diameter is increased to maintain a constant eye area, the liquid pickup speed must be increased unfavorably. Other disadvantages accrue from greater hydraulic and seal losses caused by increasing the effective shaft diameter out of proportion to the pump size.

To eliminate these shortcomings, very small pumps frequently use shafts of stainless steel or some other material that is sufficiently resistant to corrosion and wear that it does not need shaft sleeves. One such pump is illustrated in Figure 66. Manufacturing costs, of course, are much less for this type of design, and the cost of replacing the shaft is about the same as the cost of new sleeves (including installation).

Shaft Sleeves Pump shafts are usually protected from erosion, corrosion, and wear at seal chambers, leakage joints, internal bearings, and in the waterways by renewable sleeves.

The most common shaft sleeve function is that of protecting the shaft from wear at packing and mechanical seals. Shaft sleeves serving other functions are given specific names to indicate their purpose. For example, a shaft sleeve used between two multistage pump impellers in conjunction with the interstage bushing to form an interstage leakage joint is called an *interstage* or *distance sleeve*.

In medium-size centrifugal pumps with two external bearings on opposite sides of the casing (the common double-suction and multistage varieties), the favored shaft sleeve construction uses an external shaft nut to hold the sleeve in an axial position against the impeller hub. Sleeve rotation is prevented by a key, usually an extension of the impeller key (see Figure 67). The axial thrust of the impeller is transmitted through the sleeve to the external shaft nut.

In larger high-head pumps, a high axial load on the sleeve is possible and a design similar to that shown in Figure 68 may be preferred. This design has the advantages of simplicity and ease of assembly and maintenance. It also provides space for a large seal chamber

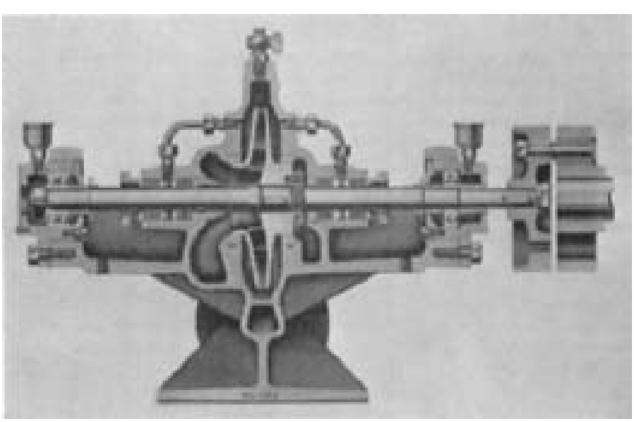


FIGURE 66 Section of a small centrifugal pump with no shaft sleeves (Flowserve Corporation)

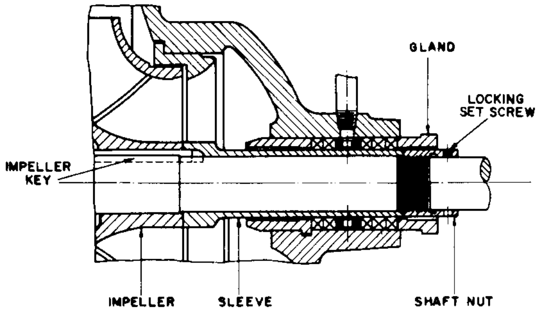


FIGURE 67 A sleeve with external locknut and impeller key extending into the sleeve to prevent rotation

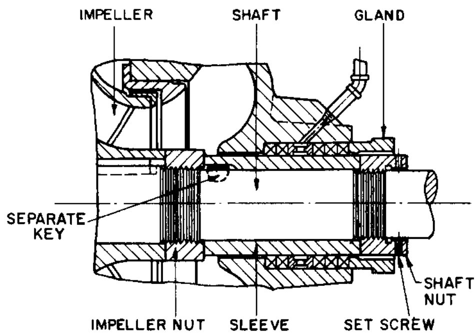


FIGURE 68 A sleeve with an internal impeller nut, external shaft-sleeve nut, and a separate key for the sleeve

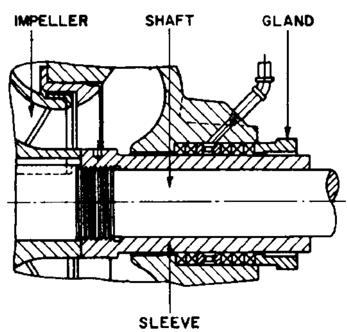


FIGURE 69 A sleeve threaded onto a shaft with no external locknut

and cartridge-type mechanical seals. When shaft sleeve nuts are used to retain the sleeves and impellers axially, they are usually manufactured with right- and left-hand threads. The friction of the pumpage and inadvertent contact with stationary parts or bushings will tend to tighten the nuts against the sleeve and impeller hub (rather than loosen them). Usually, the shaft sleeves utilize extended impeller keys to prevent rotation.

Some manufacturers favor the sleeve shown in Figure 69, in which the impeller end of the sleeve is threaded and screwed to a matching thread on the shaft. A key cannot be used with this type of sleeve, and right- and left-hand threads are substituted so that the frictional grip of the packing on the sleeve will tighten it against the impeller hub. As a safety precaution, the external shaft nuts and the sleeve itself use set screws for a locking device.

In pumps with overhung impellers, various types of sleeves are used. Most pumps use mechanical seals, and the shaft sleeve is usually a part of the mechanical seal package supplied by the seal manufacturer. Many mechanical seals are of the cartridge design, which is set and may be bench-tested for leakage prior to installation in the pump. (For a further discussion of mechanical seals, see Subsection 2.2.3.)

For overhung impeller pumps that utilize packing for sealing, the packing sleeves generally extend from the impeller hub through the seal chambers (or stuffing boxes) to protect the pump shaft from wear (see Figure 70). The sleeves are usually keyed to the shaft to prevent rotation. If a hook-type sleeve is used, the hook part of the sleeve is clamped between the impeller and a shaft shoulder to maintain the axial position of the sleeve. A hook-type sleeve used to be popular for overhung impeller pumps that operate at high temperatures because it is clamped at the impeller end and the rest of the sleeve is free to expand axially with temperature changes. But with the increased use of cartridge-type seals, the use of hook-type sleeves is diminishing.

In designs with a metal-to-metal joint between the sleeve and the impeller hub or shaft nut, a sealing device is required between the sleeve and the shaft to prevent leakage. Pumped liquid can leak into the clearance between the shaft and the sleeve when operating under a positive suction head and air can leak into the pump when operating under a negative suction head. This seal can be accomplished by means of an *O-ring*, as shown in Figure 71, or a flat gasket. For high temperature services, the sealing device must be either acceptable for the temperature to which it will be exposed, or it must be located outside the high temperature liquid environment. An alternative design used for some high-temperature process pumps is shown in Figure 72. In this arrangement, the contact surface of the hook-type sleeve and the shaft is ground at a 45-degree angle to form a metal-to-metal seal. That end of the sleeve is locked, but the other is *free* to expand with temperature changes.

When O-rings are used, any sealing surfaces must be properly finished to ensure a positive seal is achieved. All bores and changes in diameter over which O-rings must be passed should be properly radiused and chamfered to protect against damage during assembly. Guidelines for assembly dimensions and surface finish criteria are listed in O-ring manufacturers' catalogs.

Material for Packing Sleeves Packing sleeves are surrounded in the stuffing box by packing. The sleeve must be smooth so that it can turn without generating too much friction and heat. Thus, the sleeve materials must be capable of taking a very fine finish, preferably a polish. Cast-iron is therefore not suitable. A hard bronze is generally used for pumps handling clear water, but chrome or other stainless steels are sometimes pre-

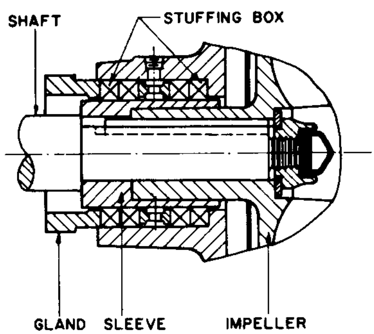


FIGURE 70 A sleeve for pumps with an overhung impeller

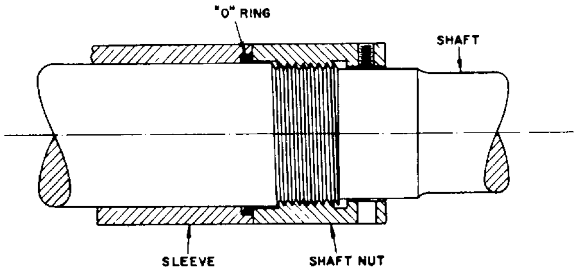


FIGURE 71 A seal arrangement for the shaft sleeve to prevent leakage along the shaft

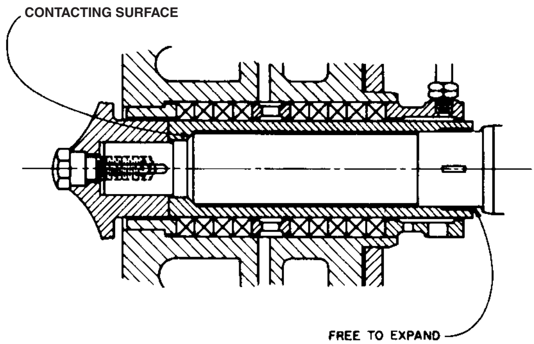


FIGURE 72 A sleeve with a 45° bevel contacting surface

ferred. For pumps subject to abrasives, hardened chrome or other stainless steels give good results. In most applications, a hardened chromium steel sleeve will be technically adequate, and the most economical choice. For severe or unusual conditions, coated sleeves are used. Ceramic coatings, applied using a plasma spray process, have also been used. Chromium oxide and aluminum oxide are the most common ceramic coatings. Both are extremely hard and resist abrasive wear well.

Ceramic coatings have been replaced in some applications by tungsten carbide coatings applied using a *high-velocity oxyfuel* (HVOF) process. The superior impact resistance and bond strength of these coatings is well documented. Another coating that is widely used on pump sleeves is a nickel-chromium-silicon-boron self-fluxing coating. This coating has a good resistance to galling and moderate resistance to abrasive wear.

Sleeves intended for coating should be machined with an undercut, so that the coating does not extend to the edge of the sleeve. This will prevent chipping at the edge, especially with the more brittle ceramic coatings.

SEAL CHAMBERS AND STUFFING BOXES

Seal chambers have the primary function of protecting the pump against leakage at the point where the shaft passes out through the pump pressure casing. If the pump handles a suction lift and the pressure at the bottom of the seal chamber (the point closest to the inside of the pump) is below atmospheric, the seal chamber function is to prevent air leakage into the pump. If this pressure is above atmospheric, the function is to prevent liquid leakage out of the pump.

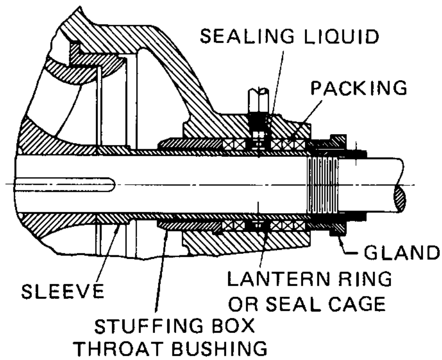


FIGURE 73 A conventional stuffing box with throat bushing

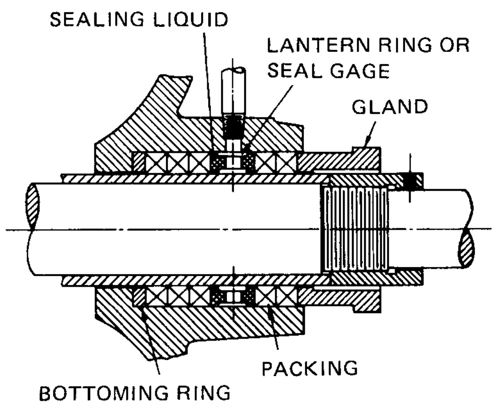


FIGURE 74 A conventional stuffing box with a bottoming ring

FIGURE 75 A lantern ring (also called a seal cage)

When sealing is accomplished by means of a mechanical seal, the seals are installed in a *seal chamber*. When sealing is accomplished by means of packing, the seal chamber is commonly referred to as a *stuffing box.* For general service pumps, a stuffing box usually takes the form of a cylindrical recess that accommodates a number of rings of packing around the shaft or shaft sleeve (see Figures 73 and 74). If sealing the box is desired, a lantern ring or seal cage (see Figure 75) is used to separate the rings of packing into approximately equal sections. The packing is compressed to give the desired fit on the shaft or sleeve by a gland that can be adjusted in an axial direction. The bottom or inside end of the box can be formed by the pump casing (refer to Figure 70), a throat bushing (see Figure 78), or a bottoming ring (see Figure 74).

For manufacturing reasons, throat bushings are widely used on smaller pumps with axially split casings. Throat bushings are always solid rather than split. The bushing is usually held from rotation by a tongue-and-groove joint locked in the lower half of the casing.

Packing Lantern Rings (Seal Cages) When a pump operates with negative suction head, the inner end of the stuffing box is under vacuum and air tends to leak into the pump. For this type of service, packing is usually separated into two sections by a lantern ring or seal cage (refer to Figure 73).

Water or some other sealing fluid is introduced under pressure into the lantern ring connection, causing a flow of sealing fluid in both axial directions. This construction is useful for pumps handling chemically active or dangerous liquids since it prevents an outflow of the pumped liquid. Lantern rings are usually axially split for ease of assembly.

Some installations involve variable suction conditions, the pump operating part of the time with suction head and part of the time with suction lift. When the operating pressure inside the pump exceeds atmospheric pressure, the liquid lantern ring becomes inoperative (except for lubrication). However, it is maintained in services so that when the pump is primed at starting, all air can be excluded.

Sealing Liquid Arrangements When a pump handles clean, cool water, sealing liquid connections are usually to the pump discharge or, in multistage pumps, to an intermediate stage. An independent supply of sealing water should be provided if any of the following conditions exist:

- A suction lift in excess of 15 ft (4.5 m)
- A discharge pressure under 10 lb/in^2 (0.7 bar)
- Hot water (over 250°F or l20°C) being handled without adequate cooling (except for boiler-feed pumps, in which lantern rings are not used)
- Muddy, sandy, or gritty water being handled
- The pump is a hot-well pump.
- The liquid being handled is other than water, such as acid, juice, molasses, or sticky liquids, without special provision in the stuffing box design for the nature of the liquid

If the suction lift exceeds 15 ft (4.5 m), excessive air infiltration through the stuffing boxes may make priming difficult unless an independent seal is provided. A discharge pressure under 10 lb/in^2 (0.7 bar) may not provide sufficient sealing pressure. Hot-well (or condensate) pumps operate with as much as a 28-in Hg (710 mm Hg) vacuum, and air infiltration would take place when the pumps are standing idle in standby service.

When sealing water is taken from the pump discharge, an external connection may be made through small-diameter piping (see Figure 76) or internal passages. In some pumps, these connections are arranged so that a sealing liquid can be introduced into the packing space through an internal drilled passage either from the pump casing or from an external source (see Figure 77). When the liquid pumped is used for sealing, the external connection is plugged. If an external sealing liquid source is required, it is connected to the external pipe tap with a socket-head pipe plug inserted at the internal pipe tap.

It is sometimes desirable to locate the lantern ring with more packing on one side than on the other. For example, in gritty-water services, a lantern ring location closer to the inner portion of the pump would divert a greater proportion of sealing liquid into the pump, thereby keeping grit from working into the box. An arrangement with most of the packing rings between the lantern ring and the inner end of the stuffing box would be applied to reduce dilution of the pumped liquid.

Some pumps handle water in which there are small, even microscopic, solids. Using water of this kind as a sealing liquid introduces the solids into the leakage path, shortening the life of the packing and sleeves. It is sometimes possible to remove these solids

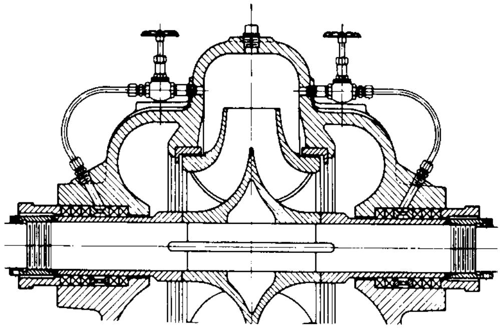


FIGURE 76 Piping connections from the pump discharge to seal cages

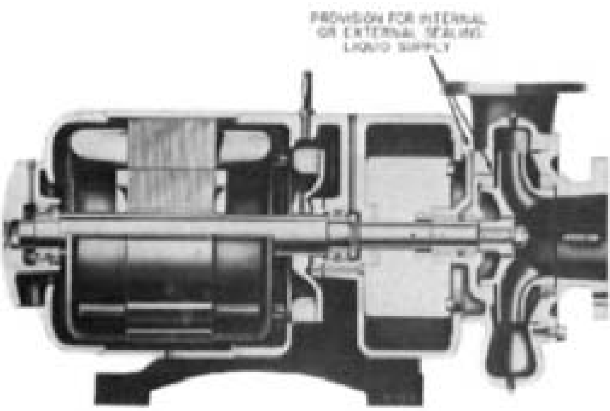


FIGURE 77 An end-suction pump with provisions for an internal or external sealing-liquid supply (Flowserve Corporation)

by installing small pressure filters in the sealing water piping from the casing to the stuffing box.

Filters ultimately get clogged, though, unless they are frequently backwashed or otherwise cleaned out. This disadvantage can be overcome by using a cyclone (or centrifugal) separator. The operating principle of the cyclone separator is based on the fact that if a liquid under pressure is introduced tangentially into a vortexing chamber, a centrifugal force will make it rotate in the chamber, creating a vortex. Particles heavier than the liquid in which they are carried will hug the outside wall of the vortexing chamber and the liquid in the center of the chamber will be relatively free of foreign matter. The action of such a separator is illustrated in Figure 78. Liquid piped from the pump discharge or from an intermediate stage of a multistage pump is piped to inlet tap A, which is drilled tangentially to the cyclone bore. The liquid containing solids is directed downward to the apex of the cone at outlet tap B and is piped to the suction or to a low-pressure point in the system. The cleaned liquid is taken off at the center of the cyclone at outlet tap C and is piped to the stuffing box.

Sand that will pass through a No. 40 sieve will be completely eliminated in a cyclone separator with supply pressures as low as 20 lb/in^2 (1.4 bar). With 100 lb/in^2 (7-bar) supply pressure, 95 percent of the particles of 5-micron size will be eliminated.

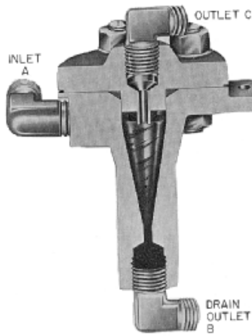


FIGURE 78 An illustration of the priniciple of cyclone separators (Flowserve Corp.)

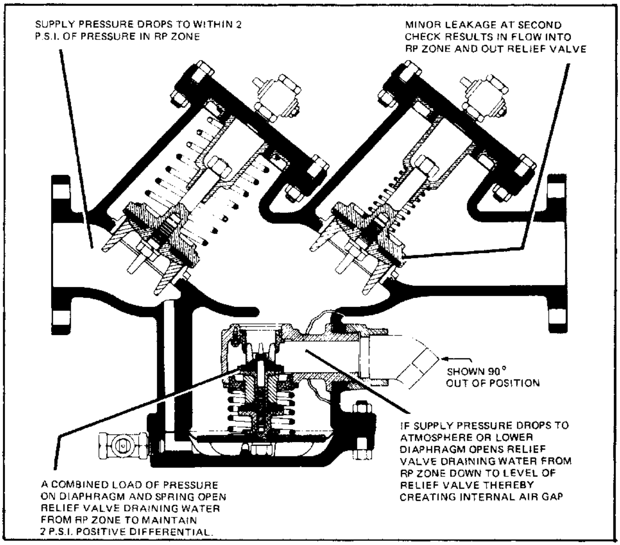


FIGURE 79 Backflow preventer (Hersey Products)

Most city ordinances require that some form of backflow preventer be interposed between city water supply lines and connections to equipment where backflow or siphoning could contaminate a drinking water supply.

This is the case, for instance, with an independent sealing supply used for stuffing boxes of sewage pumps. Quite a variety of backflow preventers are available. In most cases, the device consists of two spring-loaded check valves in a series and a spring-loaded, diaphragm-actuated, differential-pressure relief valve located in the zone between the check valves (see Figure 79).

In a normal operation, both check valves remain open as long as there is a demand for sealing water. The differential-pressure relief valve remains closed because of the pressure drop past the first check valve. If the pressure downstream of the device increases, tending to reverse the direction of the flow, both check valves close and prevent backflow. If the second check valve is prevented from closing tightly, the leakage past it increases the pressure between the two check valves, the relief valve opens, and water is discharged to the

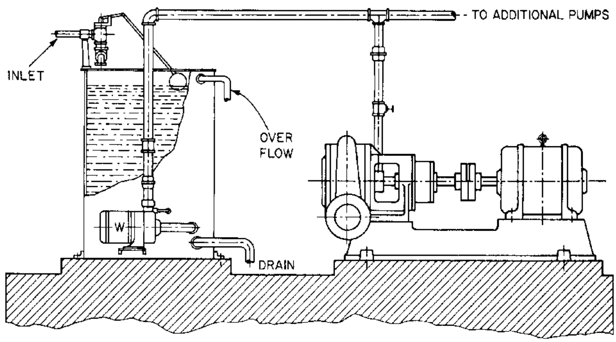


FIGURE 80 A water seal unit

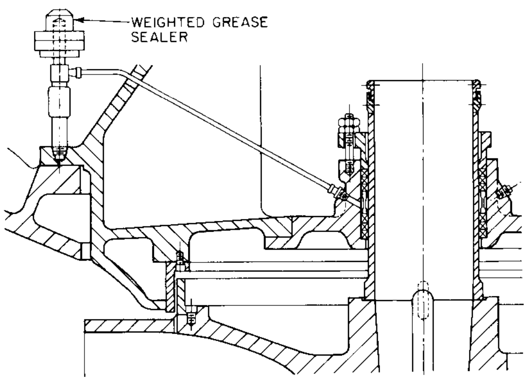


FIGURE 81 A weighted grease sealer

atmosphere. Thus, the relief valve operates automatically to maintain the pressure between the two check valves lower than the supply pressure.

Some local ordinances prohibit any connections between city water lines and a sewage or process liquid line. In such cases, an open tank under atmospheric pressure is installed into which city water can be admitted and from which a small pump can deliver the required quantity of sealing water. Such a water-sealing supply unit (see Figure 80) can be installed in a location where it can serve a number of pumps.

The tank is equipped with a float valve to feed and regulate the water level so that contamination of the city water supply is prevented. A small close-coupled pump is mounted directly on the tank and maintains a constant pressure of clear water at the stuffing box seals of the battery of pumps it serves. A small recirculation line is provided from the close-coupled pump discharge back to the tank to prevent operation at shutoff. The discharge pressure of the small supply pump is set by the maximum sealing pressure required at any of the pumps served. The supply at the individual stuffing boxes is then regulated by setting small control valves in each individual line.

If clean, cool water is not available (as with some drainage, irrigation, or sewage pumps), grease or oil seals are often used. Most pumps for sewage service have a single stuffing box subject to discharge pressure that operates with a flooded suction. It is therefore not necessary to seal these pumps against air leakage, but forcing grease or oil into the sealing space at the packing helps to exclude grit. Figure 81 shows a typical weighted grease sealer.

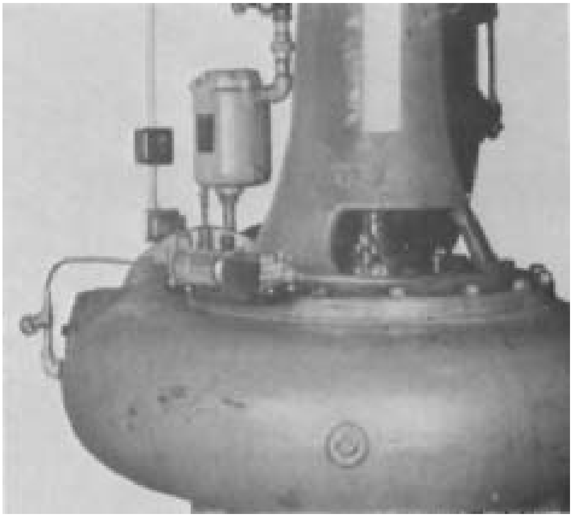


FIGURE 82 An automatic grease sealer mounted on a vertical pump (Zimmer & Francescon)

Automatic grease or oil sealers that exert pump discharge pressure in a cylinder on one side of a plunger, with light grease or oil on the other side, are available for sewage service. The oil or grease line is connected to the stuffing box seal, which is at about 80 percent of the discharge pressure. As a result, there is a slow flow of grease or oil into the pump when the unit is in operation. No flow takes place when the pump is out of service. Figure 82 shows an automatic grease sealer mounted on a vertical sewage pump.

Water-Cooled Stuffing Boxes High temperatures or pressures complicate the problem of maintaining stuffing box packing. Pumps in these more difficult services are usually provided with mechanical seals and seal support systems. When it is necessary or desirable to use packing, however, the pumps are usually equipped with jacketed, water-cooled stuffing boxes. The cooling water removes heat from the liquid leaking through the stuffing box and heat generated by friction in the box, thus improving packing service conditions. In some special cases, liquid other than water can be used in the cooling jackets. Two water-cooled stuffing box designs are commonly used. The first, shown in Figure 83, provides cored passes in the casing casting. These passages that surround the stuffing box are arranged with in-and-out connections. The second type uses a separate cooling chamber combined with the stuffing box proper, with the whole assembly inserted into and bolted to the pump casing (see Figure 84). The choice between the two is based on manufacturing preferences.

Caution is required when depending on water cooling to provide proper operation because of the danger of passage fouling and a loss of cooling effectiveness during operation. It is important that any such cooling passages be accessible for periodic inspection and cleaning to ensure that effective cooling is maintained.

Stuffing box pressure and temperature limitations vary with the pump type because it is generally not economical to use expensive stuffing box construction for infrequent high-temperature or high-pressure applications. Therefore, whenever the manufacturer's stuffing box limitations for a given pump are exceeded, the application of pressure-reducing devices ahead of the stuffing box is recommended.

Pressure-Reducing Devices Essentially, pressure-reducing devices consist of a bushing or meshing labyrinth ending in a relief chamber located between the pump interior and the stuffing box or seal chamber. The relief chamber is connected to some suitable

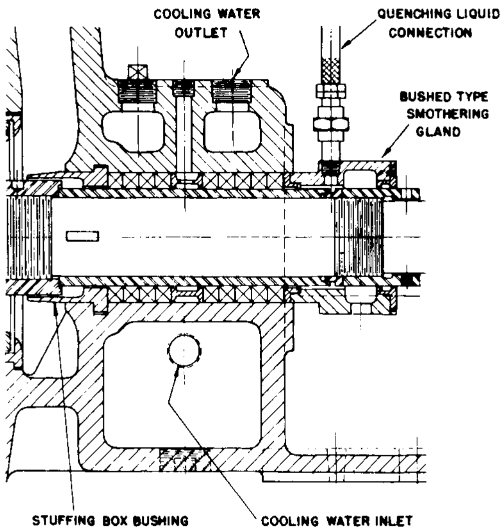


FIGURE 83 A water-cooled stuffing box with a cored water passage cast in casing

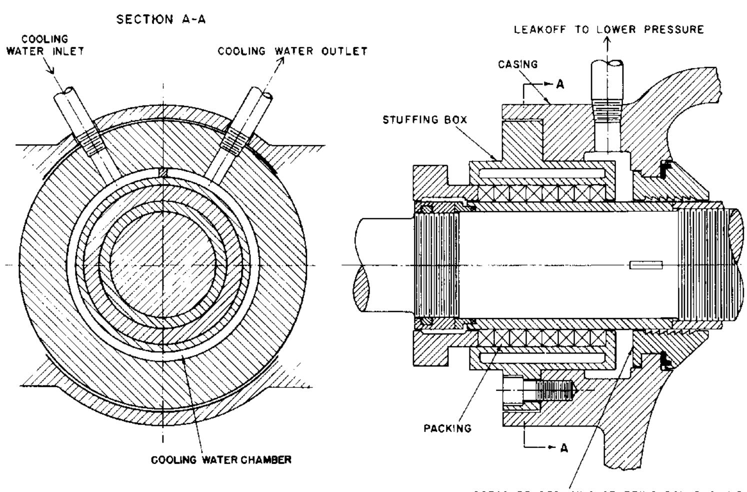


FIGURE 84 A separate water-cooled stuffing box with pressure-reducing stuffing box bushing

low-pressure point in the installation, and the leakage past the pressure-reducing device is returned to this point. If the pumped liquid must be salvaged, as with treated feedwater, it is returned to the pumping cycle. If the liquid is expendable, the relief chamber can be connected to a drain.

Many different pressure-reducing device designs exist. Figure 84 illustrates a design for limited pressures. A short serrated bushing is inserted at the bottom of the stuffing box or seal chamber, followed by a relief chamber. The leakage past the serrated bushing is bled off to a low-pressure point.

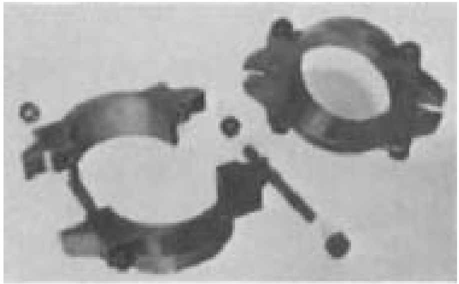


FIGURE 85 Split stuffing box gland

With relatively high-pressure units, intermeshing labyrinths can be located following the balancing device and ahead of the stuffing box or seal chamber. Piping from the chamber following pressure-reducing devices should be amply sized so that as wear increases leakage, piping friction will not increase seal chamber pressure.

Stuffing Box Packing Glands Stuffing box packing glands may assume several forms, but basically they can be classified into two groups: solid glands and split glands (see Figure 85). Split glands are made in halves so that they can be removed from the shaft without dismantling the pump, thus providing more working space when the stuffing boxes are being repacked. Split glands are desirable for pumps that have to be repacked frequently, especially if the space between the box and the bearing is restricted. The two halves are generally held together by bolts, although other methods are also used. Split glands are generally a construction refinement rather than a necessity, and they are rarely used in smaller pumps. They are commonly furnished for large single-stage pumps, for some multistage pumps, and for certain refinery pumps.

Another common refinement is the use of swing bolts in stuffing box packing glands. Such bolts may be swung to the side, out of the way, when the stuffing box is being repacked.

Stuffing box leakage into the atmosphere might, in some services, seriously inconvenience or even endanger the operating personnel. An example would be when volatile liquids are being pumped at vaporizing temperatures or temperatures above their flash point. As this leakage cannot always be cooled sufficiently by a water-cooled stuffing box, smothering glands are used (refer to Figure 83). Provisions are made in the gland to introduce a liquid, either water or another compatible liquid at a low temperature, that mixes intimately with the leakage, lowering its temperature or, if the liquid is volatile, absorbing it.

Stuffing box packing glands are usually made of bronze, although cast-iron or steel may be used for all iron-fitted pumps. Iron or steel glands are generally bushed with a nonsparking material like bronze in hazardous process services to prevent the ignition of flammable vapors by the glands sparking against a ferrous metal shaft or sleeve.

Stuffing Box Packing See Subsection 2.2.2.

MECHANICAL SEALS VERSUS PACKING

Sealing with a packed stuffing box is impractical for many conditions of service. In an ordinary packed stuffing box, the sealing between the rotating shaft or shaft sleeve and the stationary portion of the stuffing box (or seal chamber) is accomplished by means of rings of packing forced between the two surfaces and held tightly in place by a packing gland. The leakage around the shaft is controlled by merely tightening or loosening the packing gland nuts (or bolts). The actual sealing surfaces consist of the axial rotating surface of the shaft or shaft sleeve and the stationary packing. Attempts to reduce or eliminate all leakage from a conventional packed stuffing box increase the compressive load of the packing gland on the packing. The packing, being semiplastic, forms more closely to the shaft or shaft sleeve and tends to reduce the leakage. After a certain point, however, the leakage between the packing and the rotating shaft or shaft sleeve becomes inadequate to carry away the heat generated by the packing rubbing on the rotating surface, and the packing fails to function. This failure can result in burned packing, packing "blow out," and severely damaged shaft or shaft sleeve surfaces. If the sealing surface is coated, this coating may be destroyed. Even before this condition is reached, the shaft or shaft sleeve may be severely worn and scored by the packing, so that it becomes impossible to pack the stuffing box satisfactorily.

These undesirable characteristics prohibit the use of packing as a sealing method if some leakage of the pumpage to the atmosphere is not acceptable. Packing is limited in its application pressure and temperature range (see Section 2.2.2), and it is usually not acceptable for any flammable or hazardous pumping services. To address these limitations, the mechanical seal was developed (see Section 2.2.3). The mechanical seal has found general acceptance in nearly all pumping applications. Packing is still used in certain low-pressure, low-temperature applications where leakage of the pumpage is not a problem and a history of satisfactory, economical service exists.

Mechanical seals are not always the solution to every sealing situation. Seals are still subject to failure, and their failure may be more rapid and abrupt than that of packing. If packing fails, the pump can many times be kept running by temporary adjustments until it is convenient to shut it down. If a mechanical seal fails, most often the pump must be shut down immediately. As both packed stuffing boxes and conventional mechanical face seals are subject to wear, both are subject to failure. Whether one or the other should be used depends on the specific application and the experience of the user. In some cases, both give good service and the choice becomes a matter of personal preference or cost. Table 2

TABLE 2 Comparison of packing and mechanical seals

Advantages	Disadvantages
Packing	
1. Lower initial cost	1. Relatively high leakage
2. Easily installed as rings and glands are split	2. Requires regular maintenance
3. Good reliability to medium pressures and shaft speeds	3. Wear of shaft of shaft sleeve can be relatively high
4 Can handle large axial movements (thermal expansion of stuffing box versus shaft)	4. Power losses may be high
5. Can be used in rotating or reciprocating applications	
6. Leakage tends to increase gradually, giving warning of impending breakdown	
Mechanical seals	
1. Very low leakage/no leakage	1. Higher initial cost
2. Require no maintenance	2. Easily installed but may require some disassembly of pump (couplings and so on)
3. Eliminate sleeve wear/shaft wear	
4. Very good reliability	
5. Can handle higher pressures and speeds	
6. Easily applied to carcinogenic, toxic, flammable, or radioactive liquids	
7. Low power loss	

Source: John Crane Inc.

adds other comments and summarizes some advantages and disadvantages of packing and mechanical seals.

Principles and Construction of Mechanical Seals See Subsection 2.2.3.

Injection-Type Shaft Seals See Subsection 2.2.4.

BEARINGS

The function of bearings in centrifugal pumps is to keep the shaft or rotor in correct alignment with the stationary parts under the action of radial and transverse loads. Bearings that give radial positioning to the rotor are known as radial or *line bearings,* and those that locate the rotor axially are called *thrust bearings*. In most applications, the thrust bearings actually serve both as thrust and radial bearings.

Types of Bearings Used All types of bearings have been used in centrifugal pumps. Even the same basic design of pump is often made with two or more different bearings, required either by varying service conditions or by the preference of the purchaser. In most pumps, however, either rolling element or oil film (sleeve-type) bearings are used today.

In horizontal pumps with bearings on each end, the bearings are usually designated by their location as *inboard*, or drive end, and *outboard*, or non-drive end. Inboard (drive end) bearings are located between the casing and the coupling. Pumps with overhung impellers have both bearings on the same side of the casing so that the bearing nearest the impeller is called inboard and the one farthest away outboard. In a pump provided with bearings at both ends, the thrust bearing is usually placed at the outboard end and the line bearing at the inboard end.

The bearings are mounted in a housing that is usually supported by brackets attached or integral to the pump casing. The housing also serves the function of containing the lubricant necessary for proper operation of the bearing. Occasionally, the bearings of very large pumps are supported in housings that form the top of pedestals mounted on soleplates or on the pump bedplate. These are called *pedestal bearings*.

Because of the heat generated by the bearing or the heat in the liquid being pumped, some means other than radiation to the surrounding air must occasionally be used to keep the bearing temperature within proper limits. If the bearings have a force-fed lubrication system, cooling is usually accomplished by circulating the oil through a separate water-to-oil or air-to-oil cooler. Otherwise, a jacket through which a cooling liquid is circulated is usually incorporated as part of the housing.

Pump bearings may be rigid or self-aligning. A self-aligning bearing will automatically adjust itself to a change in the angular position of the shaft. In babbitted or sleeve bearings, the name *self-aligning* is applied to bearings that have a spherical fit of the sleeve in the housing. In rolling element bearings, the name is applied to bearings, the outer race of which is spherically ground or the housing of which provides a spherical fit.

Although double-suction pumps are theoretically in hydraulic balance, this balance is rarely realized in practice, and so even these pumps are provided with thrust bearings. A centrifugal pump, being a product of the foundry, is subject to minor irregularities that may cause differences in the eddy currents set up on the two sides of the impeller. As this disturbance can create an axial hydraulic thrust, some form of thrust bearing that is capable of taking a thrust in either direction is necessary to maintain the rotor in its proper position.

The thrust capacity of the bearing of a double-suction pump is usually far in excess of the probable imbalance caused by irregularities. This provision is made because (1) unequal wear of the rings and other parts may cause an imbalance and (2) the flow of the liquid into the two suction eyes may be unequal and cause an imbalance because of an improper suction-piping arrangement.

Rolling Element Bearings The most common rolling element bearings used on centrifugal pumps are the various types of ball bearings. Roller bearings are used less often, although the spherical roller bearing (see Figure 86) is used frequently for large shaft

FIGURE 86 Self-aligning spherical roller bearing (SKF USA, Inc.)

sizes, for which there is a limited choice of ball bearings. As most roller bearings are suitable only for radial loads, their use on centrifugal pumps tends to be limited to applications in which they are not required to carry a combined radial and thrust load.

Ball Bearings As the coefficient of rolling friction is less than that of sliding friction, one must not consider a ball bearing in the same light as a sleeve bearing. In the former, the load is carried on a point contact of the ball with the race, but the point of contact does not rub or slide over the race and no appreciable heat is generated. Furthermore, the point of contact is constantly changing as the ball rolls in the race, and the operation is practically frictionless. In the sleeve bearing, a constant rubbing of one surface over another occurs, and the friction must be reduced by the use of a lubricant.

Ball bearings that operate at an absolutely constant speed theoretically require no lubricant. No speed can be called absolutely constant, however, for the conditions affecting the speed always vary slightly. For instance, a motor with a full-load speed rated at 3,510 rpm might vary in speed over the course of a minute from 3,505 to 3,515 rpm. Each variation in speed causes the balls in a ball bearing to lag or lead the race because of their inertia. Consequently, a very slight, almost immeasurable sliding action takes place. Another limiting condition is that the hardest of metals suffers minute deformations on carrying loads, thus upsetting perfect point contacts and adding another slight sliding action. For these reasons, ball bearings must be given some lubrication.

Ball thrust bearings are built to carry heavy loads by pure rolling motion on an angular contact. As a thrust load is axial, it is equally distributed to all the balls around the race, and the individual load on each ball is only a small fraction of the total thrust load. In such bearings, it is essential that the balls be equally spaced, and for this purpose, a retaining cage is used between the balls and between the inner and outer races. This cage carries no load, but the contact between it and the ball produces sliding friction that requires lubrication.

Types and Applications Pump designers have a wide variety of rolling element bearings and arrangements to choose from. Ball bearings with their high-speed capabilities and low friction make them ideal for small and medium-size pumps, while roller bearings are more common in larger, slower speed pumps where a heavy capacity is required. Depending upon the specific bearing type, optional characteristics such as seals, shields, various cage materials and designs, and special internal clearances and preloads are available. Although several might be dimensionally acceptable, it is best for users to adhere to manufacturer recommendations to ensure optimum reliability.

The most common ball bearings used in centrifugal pumps are 1) single-row, deep-groove, 2) single-row, angular contact, and 3) double-row, angular contact ball bearings.

Sealed ball bearings are used in special applications such as vertical in-line pumps. Sealed prelubricated bearings require special attention if the unit in which they are

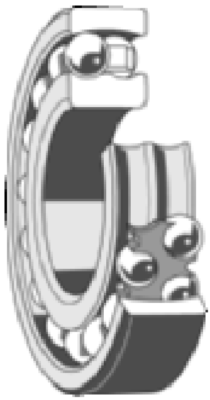


FIGURE 87 Self-aligning ball bearing (SKF USA, Inc.)

FIGURE 88 Single-row, deep-groove ball bearing (SKF USA, Inc.)

installed is not operated for long periods of time (such as stand-by units or units kept in stock or storage). The shaft should be rotated occasionally (see specific instruction manual directions) to agitate the lubricant and maintain a film coating on the bearing elements.

Self-aligning ball bearings (see Figure 87) are sometimes used for heavy loads, high speeds, long-bearing spans (large deflection angles at the bearings) and no axial thrust requirements. This bearing design acts as a pivot that compensates for misalignment and shaft deflection. For large shafts, the self-aligning spherical roller bearing (refer to Figure 86) is used instead of the self-aligning ball bearing, and it can carry both radial loads and axial thrust loads.

The single-row, deep-groove ball bearing (see Figure 88), sometimes referred to as a Conrad-type bearing, is the most commonly used bearing in centrifugal pumps, except for the larger size pumps. The Conrad-type design is recommended for use in centrifugal pumps because it can support either radial, axial, or a combination of radial and axial loads. This makes it ideal for the radial bearing in end-suction centrifugal pumps or as both the radial and thrust bearings in small pumps. The bearing design requires a careful alignment between the shaft and the housing. It is often used with seals or shields in grease-lubricated applications to help exclude dirt and retain lubricants within the bearing.

Angular contact ball bearings are commonly used in centrifugal pump applications to support axial loads or a combination of both axial and radial loads. Their axial stiffness and small operating clearances provide precise position accuracy for the shaft. Angular contact bearings are manufactured in a single-row design (see Figure 89), typically with a 40° contact angle, and also as a double-row bearing (see Figure 90), most commonly with a 30° contact angle.

Single-row, angular contact ball bearings support axial loads in only one direction when used singly. To support reversing axial loads or combined loads, single-row bearings must be mounted in a back-to-back or face-to-face arrangement where the contact angles oppose each other. Owing to its more rigid design, the back-to-back arrangement is generally recommended for centrifugal pumps, while the face-to-face arrangement is common when a slight misalignment is expected. When required to support heavy axial loads, single-row, angular contact ball bearings can be mounted in tandem where their contact angles are in the same direction. This arrangement must still be opposed with a third bearing in a back-to-back or face-to-face arrangement with the tandem pair when radial or reversing thrust loads must also be supported (see Figure 91). Depending upon the operating conditions of the pump, single-row, angular contact ball bearings typically operate with either a small clearance or a light preload.

Some applications exist where a high axial load occurs predominantly in one direction, but the thrust bearing must be capable of carrying occasional smaller axial loads in the

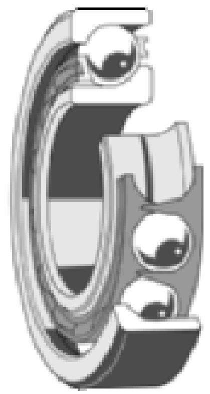


FIGURE 89 Single-row, angular contact bearing (SKF USA, Inc.)

FIGURE 90 Double-row, angular-contact bearing (SKF USA, Inc.)

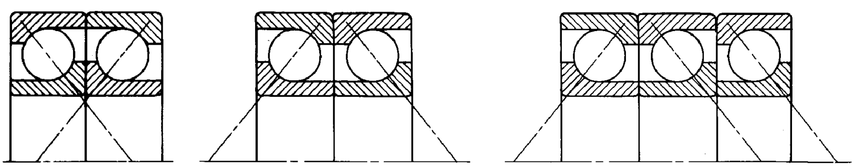


FIGURE 91 Paired bearing arrangements (SKF USA, Inc.)

reversing direction. When this occurs, a typical back-to-back angular contact bearing arrangement can result in one bearing becoming nearly completely unloaded. In the most severe cases of axial unloading of angular contact bearings, skidding of the unloaded balls within the bearing races can occur. This skidding can result in bearing heating and subsequent damage, even failure, with time. To avoid ball skidding under light load or no-load conditions, standard angular contact bearing sets can be arranged for a light preload that will result in a sufficient load on the dynamically unloaded bearing to prevent skidding. Another alternative is to install a matched set of two angular contact bearings with different contact angles (see Figure 92). By utilizing an angular contact bearing with a lower contact angle (say 15 degrees instead of the normal 40 degrees), the unloaded bearing will have a lower requirement for an axial load and be more resistant to ball skidding. This means the bearing will run at a lower temperature.

The double-row, angular contact ball bearing (see Figure 93) is similar in design to a back-to-back pair of single-row, angular contact ball bearings, but in a narrower width package. Its ease of mounting, along with its low-friction operation, high-speed capability, and seal or shield availability, make it an ideal bearing for light- to medium-duty end suction centrifugal pumps and submersible pumps.

Lubrication of Antifriction Bearing In the layout of a line of centrifugal pumps, the choice of the lubricant for the pump bearings is dictated by application requirements, by cost considerations, and sometimes by the preferences of a group of purchasers committed to the major portion of the output of that line. For example, in vertical wet-pit condenser circulating pumps, water is the lubricant of choice, in preference to grease or oil. If oil or grease is used in such pumps and the lubricant leaks into the pumping system, the condenser operation might be seriously affected because the tubes would become coated with the lubricant.

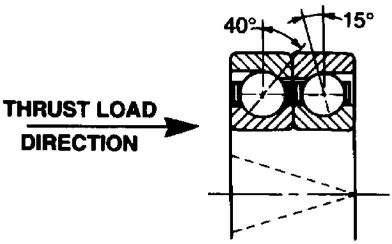


FIGURE 92 Angular contact bearings with different contact angles (SKA USA, Inc.)

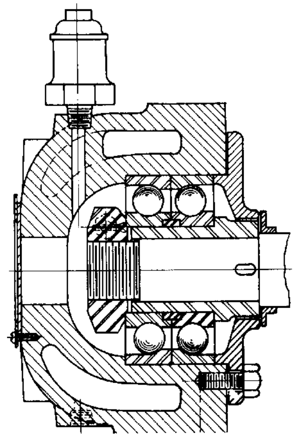


FIGURE 93 A double-row, angular-contact ball thrust bearing that is grease-lubricated and water-cooled

Most centrifugal pumps for refinery services are supplied with oil-lubricated bearings because of the insistence of refinery engineers on this feature. In the marine field, on the other hand, the preference lies with grease-lubricated bearings. For high pump operating speeds (5,000 rpm and above), oil lubrication is found to be the most satisfactory. For highly competitive lines of small pumps, the main consideration is cost, and so the most economical lubricant is chosen, depending upon the type of bearing used.

Ball bearings used in small centrifugal pumps are usually grease-lubricated, although some services use oil lubrication. In grease-lubricated bearings, the grease packed into the bearing is thrown out by the rotation of the balls, creating a slight suction at the inner race. (Even if the grade of grease is relatively light, it is still a semisolid and flows slowly. As heat is generated in the bearing, however, the flow of the grease is accelerated until the grease is thrown out at the outer race by the rotation.) As the expelled grease is cooled by contact with the housing and thus is attracted to the inner race, a continuous circulation of grease lubricates and cools the bearing. This method of lubrication requires a minimum amount of attention and has proved itself very satisfactory. A vertically mounted thrust bearing arranged for grease lubrication is shown in Figure 94.

A bearing fully packed with grease prevents proper grease circulation in itself and its housing. Therefore, as a general rule, it is recommended that only one-third of the void spaces in the housing be filled. An excess amount of grease will cause the bearing to heat up, and grease will flow out of the seals to relieve the situation. Unless the excess grease can escape through the seal or through the relief cock that is used on many large units, the bearing will probably fail early.

In oil-lubricated ball bearings, a suitable oil level must be maintained in the housing. This level should be at about the center of the lowermost ball of a stationary bearing. It can be achieved by a dam and an oil slinger to maintain the level behind the dam and thereby increase the leeway in the amount of oil the operator must keep in the housing. Oil rings are sometimes used to supply oil to the bearings from the bearing housing reservoir (see Figure 95). In other designs, a constant-level oiler is used (see Figure 96).

Because of the advantages of interchangeability, some pump lines are built with bearing housings that can be adapted to either oil or grease lubrication with minimum modifications (see Figure 97).

Oil Film or Sleeve Bearings See Subsection 2.2.5.

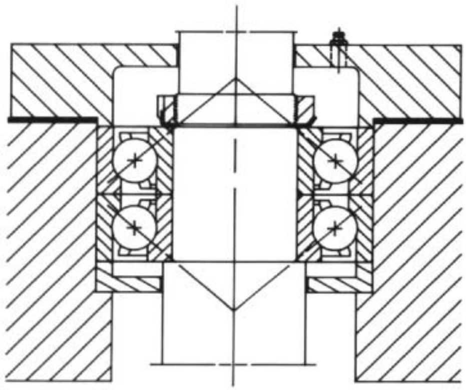


FIGURE 94 Vertically mounted thrust bearing arranged for grease lubrication (SKF USA, Inc.)

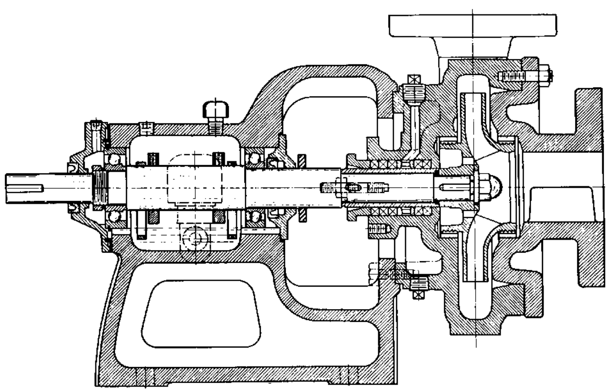


FIGURE 95 A ball bearing pump with oil rings

COUPLINGS

Centrifugal pumps are connected to their drivers through couplings of one sort or another, except for close-coupled units, in which the impeller is mounted on an extension of the shaft of the driver.

Because couplings can be used with both centrifugal and positive displacement pumps, they are discussed separately in Section 6.3.

BEDPLATE AND OTHER PUMP SUPPORTS

For very obvious reasons, it is desirable that pumps and their drivers be removable from their mountings. Consequently, they are usually bolted and doweled to machined surfaces that in turn are firmly connected to a foundation. To simplify the installation of horizontal-shaft units, these machined surfaces are usually part of a common bedplate on which either the pump or the pump and its driver have been prealigned.

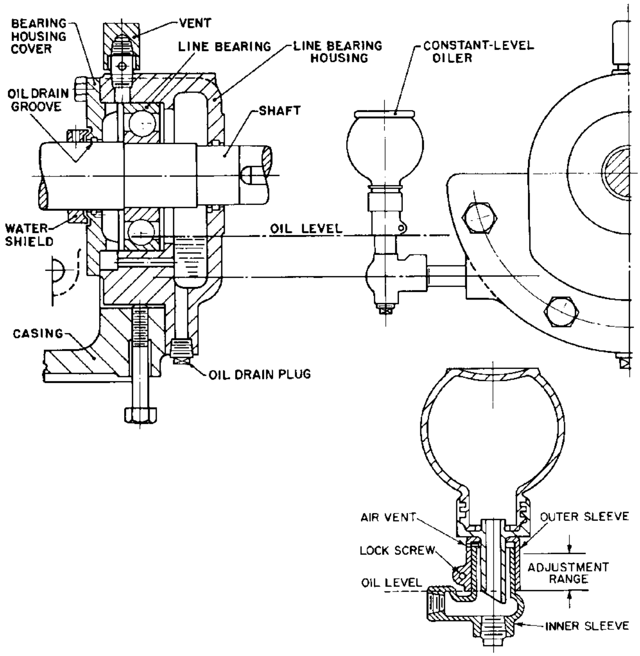


FIGURE 96 A constant-level oiler

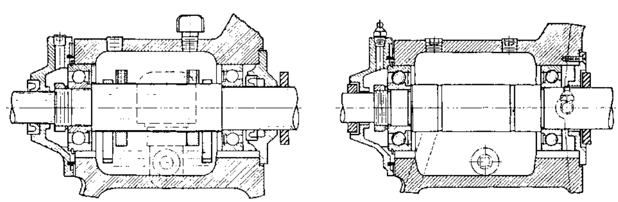


FIGURE 97 Ball bearings arranged (left) with oil rings in the housing and (right) for grease lubrication

Bedplates The primary function of a pump bedplate is to furnish mounting surfaces for the pump feet that can be rigidly attached to the foundation. Mounting surfaces are also necessary for the feet of the pump driver or drivers and for the feet of any independently mounted power transmission device. Although such surfaces could be provided by separate bedplates or by individually planned surfaces, it would be necessary to align these separate surfaces and fasten them to the foundation with the utmost care. Usually, this method requires in-place mounting in the field as well as drilling and tapping for the bolts after all the parts have been aligned. To minimize such field work, coupled horizontal-shaft pumps are usually purchased with a continuous base extending under the pump and its driver. Ordinarily, both these units are mounted and aligned at the place of manufacture.

Although such bases are designed to be quite rigid, they deflect if improperly supported. It is therefore necessary to support them on a foundation that can supply the required rigidity. Furthermore, as the base can be sprung out of shape by improper handling during transit, it is imperative that the alignment be carefully rechecked during erection and prior to starting the unit.

FIGURE 98 Horizontal shaft overhung pump and driver on a structural steel bedplate with a raised edge around the base and a tapped drainage connection (Flowserve Corporation)

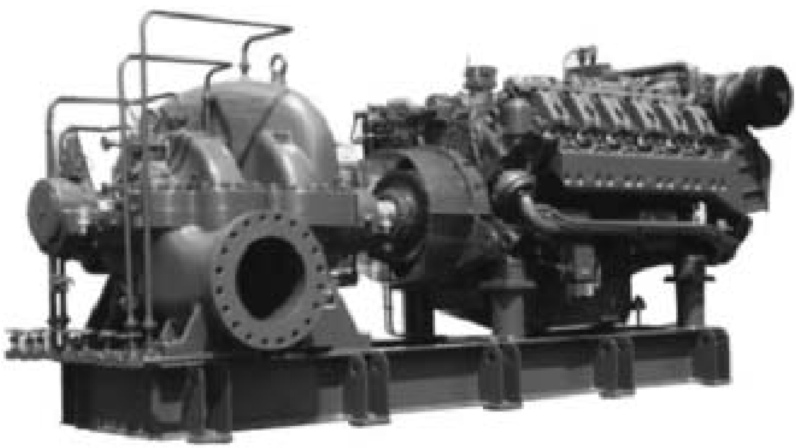


FIGURE 99 Horizontal shaft centrifugal pump and internal combustion engine mounted on a structural steel bedplate (Flowserve Corporation)

As the unit size increases, so does the size, weight, and cost of the base required. The cost of a prealigned base for most large units exceeds the cost of the field work necessary to align individual bedplates or soleplates and to mount the component parts. Such bases are therefore used only if appearances require them or if their function as a drip collector justifies the additional cost. Even in fairly small units, the height at which the feet of the pump and the other elements are located may differ considerably. A more rigid and nice-looking installation can frequently be obtained by using individual bases or soleplates and building up the foundation to various heights under the separate portions of equipment.

Baseplates are usually provided with a raised edge or raised lip around the base to prevent dripping or draining onto the floor (see Figure 98). The base itself is sloped toward

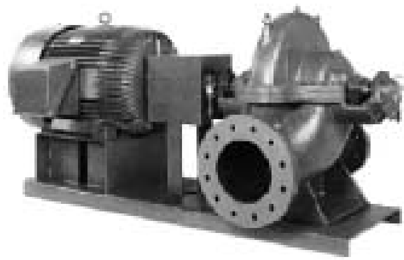


FIGURE 100 Horizontal shaft centrifugal pump and driver on a structural steel bedplate made of a simple channel shape (Flowserve Corporation)

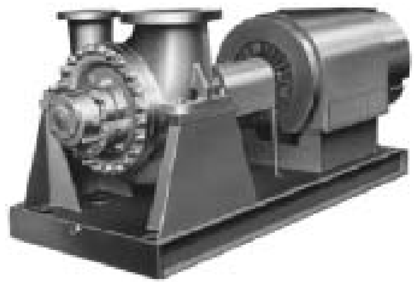


FIGURE 101 Single-stage double-suction pump with centerline support (Flowserve Corporation)

one end to collect the drainage for further disposal. A drain pocket is provided near the bottom of the slope, sometimes with a mesh screen. A tapped connection in the pocket permits piping the drainage to a convenient point.

Bedplates are usually fabricated from steel plate and structural steel shapes (see Figures 99 and 100). Even though most of these fabrications have a drain capability, because of the popular use of mechanical seals and the containment of stuffing box leakage for pumps that continue to use shaft packing (leakage is usually collected in the bearing bracket and piped to a common collecting point), the bedplate surfaces actually are seldom used to collect leakage from the pumping equipment during operation. Bedplate drain surfaces are usually employed to contain the leakage of pumpage and other liquids during pump maintenance and removal or in the event of a seal or packing failure.

Soleplates Soleplates are cast-iron or steel pads located under the feet of the pump or its driver and are embedded in the foundation. The pump or its driver is doweled and bolted to them. Soleplates are customarily used for vertical dry-pit pumps and also for some of the larger horizontal units to save the cost of the large bedplates otherwise required.

Centerline Support For operation at high temperatures, the pump casing must be supported as near to its horizontal centerline as possible in order to prevent excessive strains caused by temperature differences. Such strains might seriously disturb the alignment of the unit and eventually damage it. Centerline construction is usually employed in boiler-feed, refinery, and hot-water circulating pumps (see Figure 101). The exact temperature at which centerline support construction becomes mandatory varies from 250 to 350°F (121 to 177°C).

Horizontal Units Using Flexible Pipe Connections The previous discussion of bedplates and supports for horizontal-shaft units assumed their application would be to pumps with piping setups that do not impose hydraulic thrusts on the pumps. If flexible pipe connections or expansion joints are desirable in the suction or discharge piping of a pump (or in both), the pump manufacturer should be so advised for several reasons. First, the pump casing will be required to withstand various stresses caused by the resultant hydraulic thrust load. Although this is rarely a limiting or dangerous factor, it is best that the manufacturer have the opportunity to check the strength of the pump casing. Second, the resulting hydraulic thrust has to be transmitted from the pump casing through the casing feet to the bedplate or soleplate and then to the foundation. Usually, horizontal-shaft pumps are merely bolted to their bases or soleplates, and so any tendency to displacement is resisted only by the frictional grip of the casing feet on the base and by relatively small dowels. If flexible pipe joints are used, this attachment may not be sufficient to withstand the hydraulic thrust. If high hydraulic thrust loads are to be encountered, therefore the pump feet must be keyed to the base or supports. Similarly, the bedplate or supporting soleplates must be of a design that will permit transmission of the load to the foundation. Each of these design elements must be checked to confirm their capability to withstand hyraulic thrust loads from flexible pipe connections. Preferably expansion joints will be restrained to avoid transmitting loads to the pump nozzles.

VERTICAL PUMPS

Vertical-shaft pumps fall into two classifications: dry-pit and wet-pit. Dry-pit pumps are surrounded by air, and the wet-pit types are either fully or partially submerged in the liquid handled.

Vertical Dry-Pit Pumps Dry-pit pumps with external bearings include most small, medium, and large vertical sewage pumps, most medium and large drainage and irrigation pumps for medium and high heads, many large condenser circulating and water supply pumps, and many marine pumps. Sometimes the vertical design is preferred (especially for marine pumps) because it saves floor space. At other times, it is desirable to mount a pump at a low elevation because of suction conditions, and it is then also preferable or necessary to have the pump driver at a high elevation. The vertical pump is normally used for large capacity applications because it is more economical than the horizontal type, all factors considered.

Many vertical dry-pit pumps are basically horizontal designs with minor modifications (usually in the bearings) to adapt them for vertical-shaft drive. This is not true of small- and medium-sized sewage pumps, however. In these units, a purely vertical design is the most popular. Most of these sewage pumps have elbow suction nozzles (see Figures 102 through 104) because their suction supply is usually taken from a wet well adjacent to the

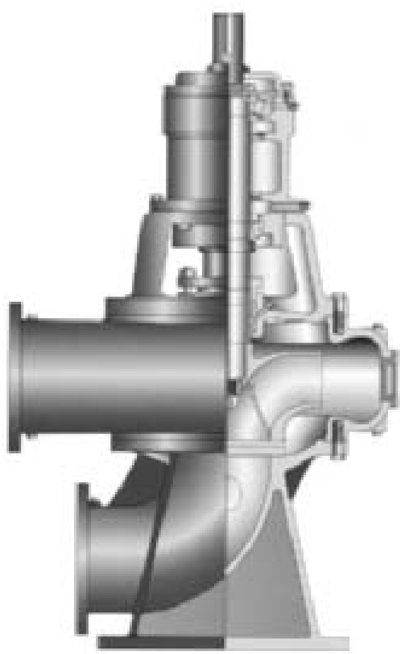


FIGURE 102 Section of a vertical sewage pump with end-suction (elbow) and side discharge (Flowserve Corporation)

FIGURE 103 An installed vertical sewage pump similar to that shown in Figure 102 (Flowserve Corporation)

FIGURE 104 Vertical sewage pump with a direct-mounted motor (Flowserve Corporation)

pit in which the pump is installed. The suction elbow usually contains a handhole with a removable cover to provide easy access to the impeller.

To dismantle one of these pumps, the stuffing box head must be unbolted from the casing after the intermediate shaft or the motor and motor stand have been removed. The rotor assembly is drawn out upward, complete with the stuffing box head, the bearing

FIGURE 105 Vertical bottom-suction volute pumps installed in a sewage pumping station (Flowserve Corporation)

housing, and the like. This rotor assembly can then be completely dismantled at a convenient location.

Vertical-shaft installations of single-suction pumps with a suction elbow are commonly furnished with either a pedestal or a base elbow (refer to Figure 102), both of which can be bolted to soleplates or even grouted in. The grouting arrangement is not desirable unless there is full assurance that the pedestal or elbow will never be disturbed or that the grouted space is reasonably regular and the grout will separate from the pump without excessive difficulty.

Vertical single-suction pumps with bottom suction are commonly used for larger sewage, water supply, or condenser circulating applications. Such pumps are provided with wing feet that are bolted to soleplates grouted in concrete pedestals or piers (see Figure 105). Sometimes the wing feet may be grouted right in the pedestals. These must be suitably arranged to provide proper access to any handholes in the pump and to allow clearance for the elbow suction nozzles if these are used.

If a vertical pump is applied to a condensate service or some other service for which the eye of the impeller must be vented to prevent vapor binding, a pump with a bottom single-inlet impeller is not desirable because it does not permit effective venting. Neither does a vertical pump employing a double-suction impeller (see Figure 106). The most suitable design for such applications incorporates a top single-inlet impeller (see Figure 107).

If the driver of a vertical dry-pit pump can be located immediately above the pump, it is often supported on the pump itself (refer to Figure 104). The shafts of the pump and driver may be connected by a flexible coupling, which requires that each have its own thrust bearing. If the pump shaft is rigidly coupled to the driver shaft or is an extension of the driver shaft, a common thrust bearing is used, normally in the driver.

Although the driving motors are frequently mounted on top of the pump casing, one important reason for the use of the vertical shaft design is the possibility of locating the motors at an elevation sufficiently above the pumps to prevent the accidental flooding of the motors. The pump and its driver may be separated by an appreciable length of shafting, which may require steady bearings between the two units. Subsection 6.3.1 discusses the construction and arrangement of the shafting used to connect vertical pumps to drivers located some distance above the pump elevation.

Bearings for vertical dry-pit pumps and for intermediate guide bearings are usually antifriction grease-lubricated types to simplify the problem of retaining a lubricant in a housing with a shaft projecting vertically through it. Larger units, for which antifriction bearings are not available or desirable, use self-oiling, babbitt steady bearings with spiral

FIGURE 106 A vertical double-suction volute pump with a direct-mounted motor (Flowserve Corporation)

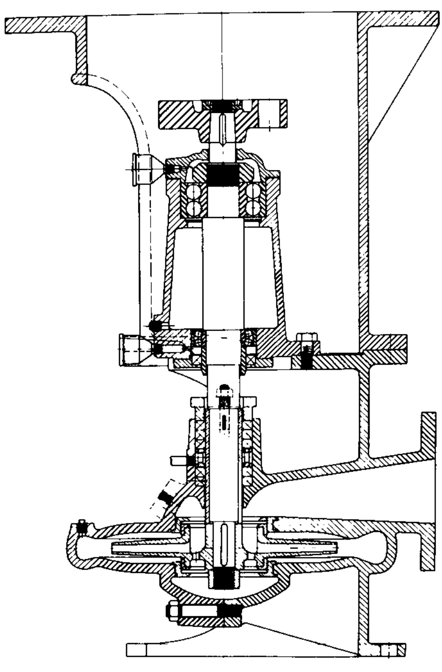


FIGURE 107 A section of a vertical pump with a top-suction inlet impeller (Flowserve Corporation)

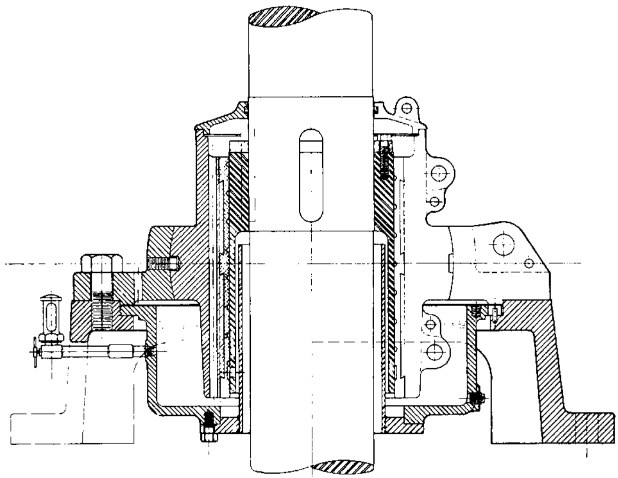


FIGURE 108 A self-oiling, babbitt steady bearing for large vertical shafting (Flowserve Corporation)

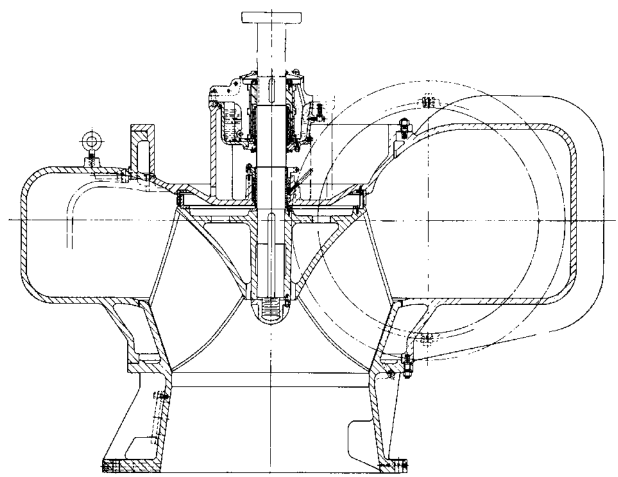


FIGURE 109 A section of a large, vertical, bottom-suction volute pump with a single sleeve bearing (Flowserve Corporation)

oil grooves (see Figures 108 and l09). Figure l09 illustrates a vertical dry-pit pump design with a single-sleeve line bearing. The pump is connected by a rigid coupling to its motor (not shown), which is provided with a line and a thrust bearing.

Vertical dry-pit centrifugal pumps are structurally similar to horizontal-shaft pumps. It is to be noted, however, that many of the large, vertical, single-stage, single-suction (usually bottom) volute pumps that are preferred for large storm water pumpage, drainage, irrigation, sewage, and water supply projects have no comparable counterpart among horizontal-shaft units. The basic U-section casing of these pumps, which is structurally weak, often requires the use of heavy ribbing to provide sufficient rigidity. In the comparable water turbine practice, a set of vanes (called a *speed ring*) is employed between the casing and the runner to act as a strut. Although the speed ring does not adversely affect the operation of a water turbine, it would function basically as a diffuser in a pump because of the inherent hydraulic limitations of that construction. Some high-head pumps of this type have been made in the twin-volute design. The wall separating the two volutes acts as a stengthening rib for the casing, thus making it easier to design a casting strong enough for the pressure involved.

Bases and Supports for Vertical Pumping Equipment Vertical-shaft pumps, like horizontal-shaft units, must be firmly supported. Depending upon the installation, the unit can be supported at one or several elevations. Vertical units are seldom supported from walls, but even that type of support is sometimes encountered.

Occasionally, a nominal horizontal-shaft pump design is arranged with a vertical shaft and a wall used as the supporting foundation. Regular horizontal-shaft units can be used for this purpose without modification, except that the bedplate is attached to a wall. Careful attention must be given to the arrangement of the pump bearings to prevent the escape of the lubricant. Installations of double-suction, single-stage pumps with the shaft in the vertical position are relatively rare, except in some marine or navy applications. Hence, manufacturers have few standard pumps of this kind arranged so that a portion of the casing forms the support (to be mounted on soleplates). Figure 106 shows such a pump, which also has a casing extension to support the driving motor.

Vertical Wet-Pit Pumps Vertical pumps intended for submerged operations are manufactured in a great number of designs, depending mainly upon the service for which they are intended. Small pumps of this type are often referred to as *sump pumps*. Wet-pit centrifugal pumps can be classified in the following manner:

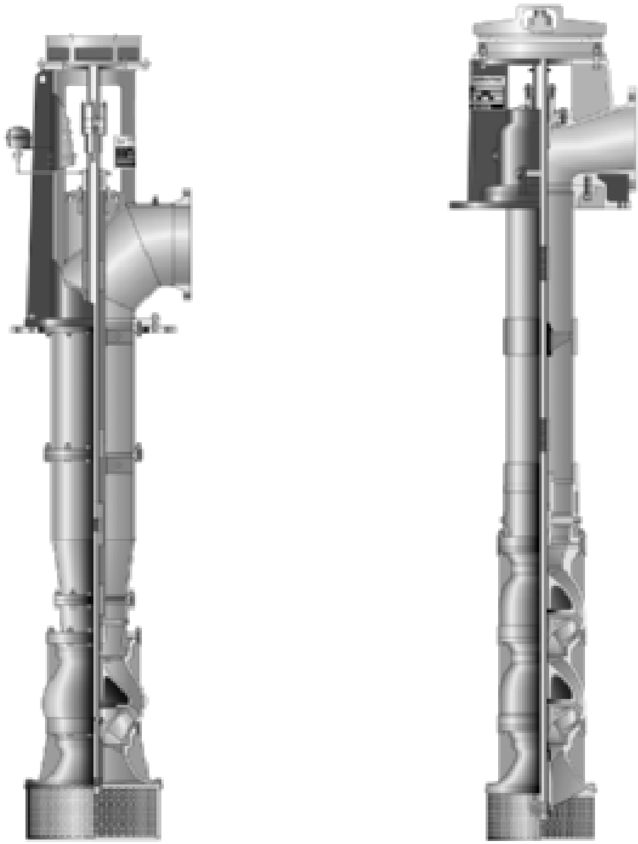
- Vertical turbine pumps
- Propeller or modified propeller pumps
- Volute pumps

Vertical Turbine Pumps Vertical turbine pumps were originally developed for pumping water from wells and have been called *deep-well pumps, turbine well pumps,* and *borehole pumps.* As their application to other fields has increased, the name *vertical turbine pumps* has been generally adopted by manufacturers. This is not too specific a designation because the term *turbine pump* has been applied in the past to any pump employing a diffuser. There is now a tendency to designate pumps using diffusion vanes as *diffuser pumps* to distinguish them from *volute pumps.* As that designation becomes more universal, applying the term *vertical turbine pumps* to the construction formerly called *turbine well pumps* will become more specific.

The largest fields of application for the vertical turbine pump are pumping from wells for irrigation and other agricultural purposes, for a municipal water supply, and for industrial water supplies, as well as for processing, circulating, refrigerating, and air conditioning. This type of pump has also been utilized for brine pumping, mine dewatering, oil field repressuring, and other purposes.

These pumps have been made for capacities as low as 10 or 15 gpm (2 or 3 m^3/h) and as high as 25,000 gpm (5700 m^3/h) or more and for heads up to 1,000 feet (300 m). Most applications naturally involve the smaller capacities. The capacity of the pumps used for bored wells is naturally limited by the size of the well as well as by the rate at which water can be drawn without lowering its level to a point of insufficient pump submergence.

Vertical turbine pumps should be designed with a shaft that can be readily raised or lowered from the top to permit proper positioning of the impeller in the bowl. An adequate thrust bearing is also necessary to support the vertical shafting, the impeller, and the hydraulic thrust developed when the pump is in service. As the driving mechanism must also have a thrust bearing to support its vertical shaft, it is usually provided with one large enough to carry the pump parts as well. For these two reasons, the hollow-shaft motor or gear is most commonly used for vertical turbine pump drives. In addition, these pumps are sometimes made with their own thrust bearings to allow for a belt drive or for a drive through a flexible coupling by a solid-shaft motor, gear, or turbine. Dual-driven pumps usually employ an angle gear with a vertical motor mounted on its top.



(a) (b)

FIGURE 110 A vertical turbine pump design with enclosed impellers and: a) enclosed line shafting and b) open-line shafting (Flowserve Corporation)

The design of vertical pumps illustrates how a centrifugal pump can be specialized to meet a specific application. Figure 110 illustrates a turbine design with closed impellers and enclosed-line shafting and another turbine design with closed impellers and open-line shafting.

The bowl assembly, or section, consists of the suction case (also called *suction head* or *inlet vane*), the impeller or impellers, the discharge bowl, the intermediate bowl or bowls (if more than one stage is involved), the discharge case, the various bearings, the shaft, and the miscellaneous parts, such as keys, impeller-locking devices, and the like. The column pipe assembly consists of the column pipe, the shafting above the bowl assembly, the shaft bearings, and the cover pipe or bearing retainers. The pump is suspended from the driving head, which consists of the discharge elbow (for above ground discharge), the motor or driver and support, and either the stuffing box (in an open-shaft construction) or the assembly for providing tension on the cover pipe and introducing a lubricant into it. Below ground discharge is taken from a tee in the column pipe, and the driving head functions principally as a stand for the driver and a support for the column pipe.

Liquid in a vertical turbine pump is guided into the impeller by the suction case or head. This may be a tapered section for the attachment of a conical strainer or suction pipe, or it may be a bell mouth.

Semiopen and enclosed impellers are both commonly used. For proper clearances in the various stages, the semiopen impeller requires more care in assembly on the impeller shaft and more accurate field adjustments of the vertical shaft position in order to obtain the best efficiency. Enclosed impellers are favored over semiopen ones because wear on the latter reduces capacity, which cannot be restored unless new impellers are installed. Normal wear on enclosed impellers does not affect impeller vanes, and worn clearances may be restored by replacing wearing rings. The thrust produced by semiopen impellers may be as much as 150 percent of that by enclosed impellers.

Occasionally, in power plants, the maximum water level that can be carried in the condenser's hot well will not give adequate NPSH for a conventional horizontal condensate pump mounted on the basement floor, especially if the unit has been installed in a space originally allotted for a smaller pump. Building a pit for a conventional horizontal condensate pump or a vertical dry-pit pump that will provide sufficient submergence involves considerable expense. Pumps of the design shown in Figure 111 have become quite popular in such applications. This is basically a vertical turbine pump mounted in a tank (often called a can) that is sunk into the floor. The length of the pump has to be such that sufficient NPSH will be available for the first-stage impeller design, and the diameter and length of the tank have

FIGURE 111 Vertical turbine can pump for condesate service (Flowserve Corporation)

FIGURE 112 Vertical propeller pump installed (Flowserve Corporation)

to allow for proper flow through the space between the pump and tank and then for a turn and flow into the bell mouth. Installing this design in an existing plant is naturally much less expensive than making a pit because the size of the hole necessary to install the tank is much smaller. The same basic design has also been applied to pumps handling volatile liquids that are mounted on the operating floor and not provided with sufficient NPSH.

Propeller Pumps Originally, the term *vertical propeller pump* was applied to vertical wet-pit diffuser or turbine pumps with a propeller or axial-flow impellers, usually for installation in an open sump with a relatively short setting (see Figures 112 and 113). Operating heads exceeding the capacity of a single-stage axial-flow impeller might call for a pump of two or more stages or a single-stage pump with a lower specific speed and a mixed-flow impeller. High enough operating heads might demand a pump with mixed-flow impellers and two or more stages. For lack of a more suitable name, such high-head designs have usually been classified as propeller pumps also.

Although vertical turbine pumps and vertical modified propeller pumps are basically the same mechanically and could even be of the same specific speed hydraulically, a basic turbine pump design is suitable for a large number of stages. A modified propeller pump design, however, is basically intended for a maximum of two or three stages.

Most wet-pit drainage, low-head irrigation, and storm water installations employ conventional propeller or modified propeller pumps. These pumps have also been used for condenser circulating services, but a specialized design dominates this field. As large power plants are usually located in heavily populated areas, they frequently have to use badly contaminated water (both fresh and salt) as a cooling medium. Such water quickly short-

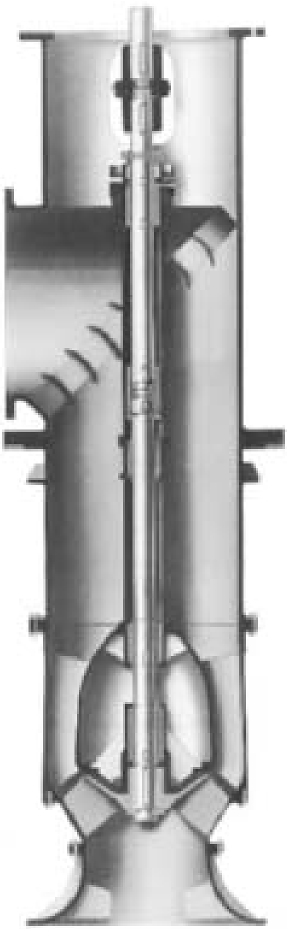


FIGURE 113 Vertical propeller pump with above ground discharge (Flowserve Corporation)

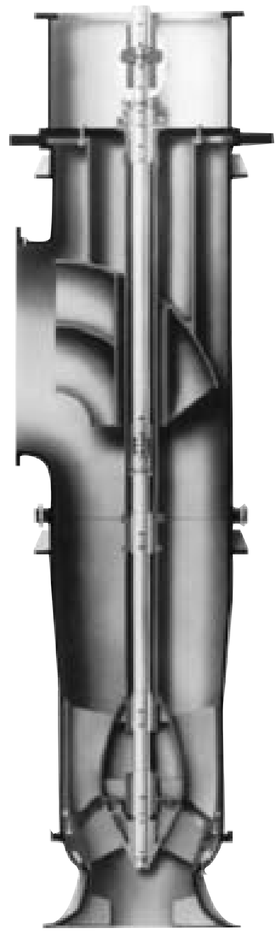


FIGURE 114 Vertical pull-out design allows the rotating element and critical non-rotating wear components to be removed for inspection and replacement without removing the complete pump (Flowserve Corporation)

ens the life of fabricated steel. Cast iron, bronze, or an even more corrosion-resistant cast metal must therefore be used for the column pipe assembly. This requirement means a very heavy pump if large capacities are involved. To avoid the necessity of lifting this large mass for maintenance of the rotating parts, some designs (one of which is illustrated in Figure 114) are built so that the impeller, diffuser, and shaft assembly can be removed from the top without disturbing the column pipe assembly. These designs are commonly designated as *pullout* designs.

Like vertical turbine pumps, propeller and modified propeller pumps have been made with both open- and enclosed-line shafting. Except for condenser circulating services, enclosed shafting, using oil as a lubricant but with a grease-lubricated tail bearing below the impeller, seems to be favored. Some pumps handling condenser circulating water use enclosed shafting but with water (often from another source) as the lubricant, thus eliminating any possibility of oil getting into the circulating water and coating the condenser tubes.

Propeller pumps have open propellers. Modified propeller pumps with mixed-flow impellers are made with both open and closed impellers.

Volute Pumps A variety of wet-pit pumps are available. The liquid pumped, be it clean water, sewage, abrasive liquids or slurries, dictates whether a semiopen or an enclosed impeller will be used, whether the shafting will be open or closed to the liquid pumped, and whether the bearings will be submerged or located above the liquid.

Figure 115 illustrates a single-volute pump with a single-suction enclosed nonclog impeller, no pump-out vanes or wearing-ring joints on the hack side of the impeller, and enclosed shafting. The pump is designed to be suspended from an upper floor by means of a drop pipe and for pumping sewage or other solid-laden liquids. To seal against leakage along the shaft at the point where it passes through the casing, a seal chamber or a stuffing box is provided. The design of a stuffing box can be either like the one shown in Figure 116, which uses rings of packing and a spring-loaded gland, or the one shown in Figure 117, which uses U-cup packing requiring no gland. The pump shown in Figure 115 uses two sleeve bearings above the impeller. The bottom bearing is grease-lubricated, and a seal is provided to prevent grease leakage as well as to keep out any grit. The upper bearing connects the shaft cover pipe to the pump-bearing bracket and the upper end of the cover pipe to the floorplate. This bearing is gravity-feed oil-lubricated. If intermediate bearings are required, they are also supported by the cover pipe and are oil-lubricated. The pump thrust is carried by the motor, which can be either a hollow-shaft or a solid-shaft construction. The latter type requires the use of a rigid coupling between the pump and motor shafts.

In most applications, these volute-type pumps have been replaced with vertical wet-pit pumps with the stuffing box/seal chamber in the discharge head (see Section 9.2, Figure 4b).

A design that uses open shafting and no seal chamber or stuffing box at the pump casing, incorporating its own thrust bearing, is shown in Figure 118. The impeller shown in Figure 118 is of the vortex type (sometimes called a *recessed impeller*, as shown in Figure 119), which is suitable for pumping heavy concentrations of solid material (such as sludges or slurries) or in certain food-processing applications, but other types of impellers can be substituted. Pumped liquid leakage from the casing is relieved back to the suction through holes in the support pipe. The seal chamber or stuffing box at the driver floor elevation is used only when gas tight construction is desired. The lower and any intermediate sleeve bearings are grease-lubricated as shown, but gravity-feed oil lubrication is also available in other designs. The upper antifriction thrust bearing is grease-lubricated. A solid shaft motor and a flexible shaft are used.

Figure 120 illustrates what is called a *cantilever-shaft pump*, which has the unique feature of having no bearings below the liquid surface. The shaft is exposed to the liquid pumped. External antifriction grease-lubricated bearings are provided above the floor and are properly spaced to support the rigid shaft. They carry both the thrust and the radial load. A flexible coupling is used between the pump and the solid-shaft motor. The stuffing box at the floorplate may be eliminated if holes are provided in the drop pipe to maintain the liquid level in the pipe even with the sump liquid level. Either semiopen or enclosed impellers may be used.

An interesting design of the wet-pit pump is shown in Figure 121. It uses a single-stage, double-suction impeller in a twin-volute casing. Because the axial thrust is balanced, the thrust bearing need carry only the weight of the rotating element. The pump requires no stuffing box or mechanical seal. The shaft is entirely enclosed, and the bearings are externally lubricated, either with oil or with water. The lower bearing receives its lubrication from an external pipe connection.

The term sump pump ordinarily conveys the idea of a vertical wet-pit pump that is suspended from a floorplate or sump cover. It could be supported by a foot on the bottom of a well, be motor-driven and automatically controlled by a float switch, and be used to remove drainage collected in a sump. The term does not indicate a specific construction, for both diffuser and volute designs are used. These may be single-stage or multistage and have open or closed impellers of a wide range of specific speeds.

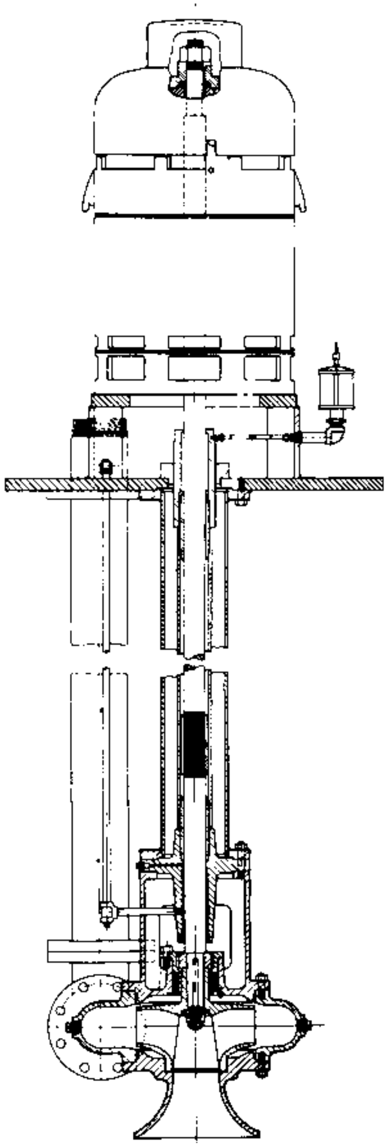


FIGURE 115 A section of a vertical wet-pit, nonclogging pump (Flowserve Corporation)

For small capacities driven by fractional-horsepower motors, *cellar drainers* can be used. These are small and usually single-stage volute pumps with single-suction impellers (either top or bottom suction) supported by a foot on the casing. The motor is supported well above the impeller by some form of column enclosing the shaft. These drainers are made as complete units, including float, float switch, motor, and strainers (see Figure 122).

The larger sump pumps are usually standardized but obtainable in any length, with covers of various sizes (on which a float switch may be mounted) and the like. Duplex

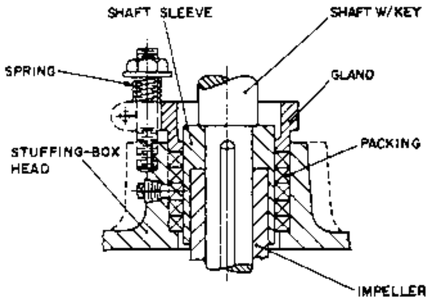
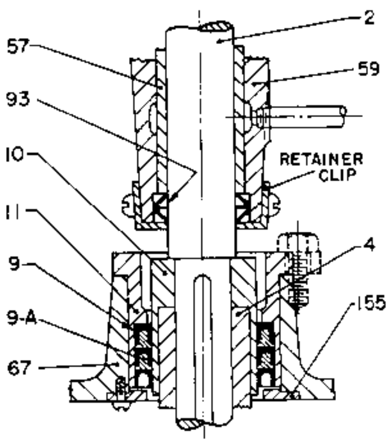


FIGURE 116 A typical stuffing box arrangement



STATIONARY PARTS		
REF. NO.	NO. OF PCS.	NAME OF PARTS
9	3	U-CUP SEAL RING
9-A	2	SPACERS
11	1	SEAL RING ADAPTER
57	1	LOWER PUMP BEARING
59	1	BEARING BRACKET
67	1	STUFFING-BOX HEAD
93	2	OIL SEAL
155	1	SEAL-RING RETAINER

ROTATING PARTS		
REF. NO	NO. OF PCS.	NAME OF PARTS
2	1	SHAFT W/KEY
4	1	IMPELLER
10	1	SHAFT SLEEVE

FIGURE 117 A stuffing box arrangement with U-cup packing (Flowserve Corporation)

units, that is, two pumps on a common sump cover (sometimes with a hole for access to the sump), are often used (see Figure 123). Such units may operate their pumps in a fixed order, or a mechanical or electric alternator may be used to equalize their operation.

The Application of Vertical Wet-Pit Pumps Like all pumps, the vertical wet-pit pump has advantages and disadvantages. One advantage is that installation does not require a separate dry pit to collect the pumped liquid. If the impeller (first-stage impeller in multistage pumps) is submerged, no priming problem exists and the pump can be automatically controlled without fear of its ever running dry. Moreover, the available NPSH is greater (except in closed tanks) and often permits a higher rotative speed for the same service conditions. A second advantage is that the motor or driver can be located at any desired height above any flood level.

It has the following mechanical disadvantages: (1) the possibility of freezing when idle, (2) the possibility of damage by floating objects if the unit is installed in an open ditch or similar installation, (3) the inconvenience of lifting out and dismantling for inspection and repairs, no matter how small, and (4) the pump bearings have a relative short life unless the water and bearing design are ideal. In summary, the vertical wet-pit pump is the best pump available for some applications. It's not ideal but can be the most economical for certain installations, a poor choice for some, and the least desirable for still others.

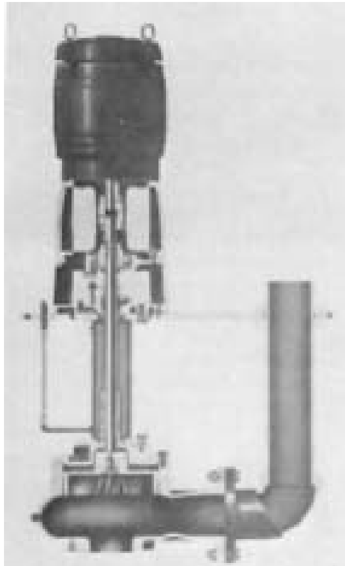


FIGURE 118 A vertical wet-pit volute pump with open shafting, no stuffing box, and its own thrust bearing (Aurora Pump)

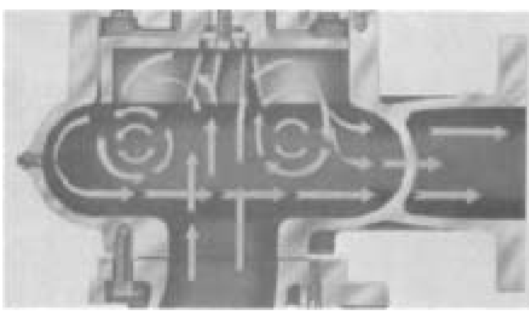


FIGURE 119 The principle operation of the vortex pump in Figure 118 (Aurora Pump)

Axial Thrust in Vertical Pumps with Single Suction Impellers The subject of axial thrust in horizontal pumps has already been discussed in this section. When pumps are installed in a vertical position, additional factors need be taken into account when determining the amount and direction of the thrust to be absorbed in the thrust bearings.

The first and most significant of these factors is the weight of the rotating parts, which is a constant downward force for any given pump, completely independent of the pump operating capacity or total head. Since in most cases the single-suction impellers of vertical pumps are mounted with the suction eye facing downward, the normal hydraulic axial thrust is exerted downward and the weight of the rotating parts is additive to this axial thrust.

The second factor involves the dynamic force (or change in momentum) caused by the change in the direction of flow, from vertical to either horizontal or partly horizontal, as the pumped liquid flows through the impeller. This force acts upward and balances a small portion of the hydraulic downthrust and of the rotor weight. The magnitude of this force for water having a specific weight (force) of 62.34 lb/ft^3 (9.79 kN/m^3) is

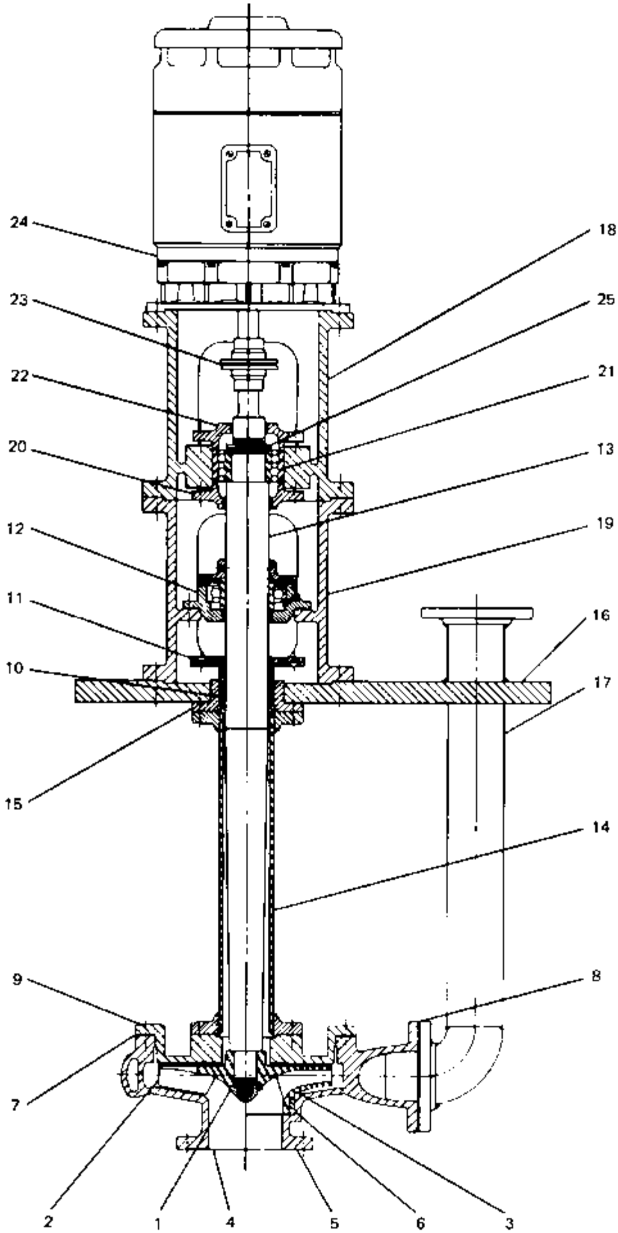


FIGURE 120 Vertical wet-pit cantilever-type volute pump: (1) impeller nut, (2) open impeller, (3) closed impeller, (4) open casing, (5) closed casing, (6) casing wearing ring, (7) casing gasket, (8) discharge flange gasket, (9) back cover, (10) packing or mechanical seal, (11) packing or seal gland, (12) radial bearing, (13) shaft, (14) supporting pipe, (15) separate stuffing box, (16) sump cover, (17) discharge pipe, (18) upper motor pedestal, (19) lower motor pedestal, (20) lower thrust bearing cover, (21) thrust bearing, (22) upper thrust bearing cover, (23) coupling, (24) motor, (25) bearing locknut and washer (Laurence Pump)

FIGURE 121 Double-suction, wet-pit pump (Flowserve Corporation)

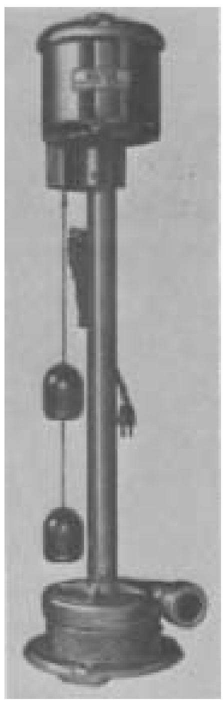


FIGURE 122 A cellar-drainer sump pump (Sta-Rite Products)

in USCS units $$F_u = \frac{KV_e^2 A_e}{2g(2.31)}$$

in SI units $$F_u = \frac{KV_e^2 A_e}{2g(1.02)}$$

where F_u = upthrust, lb (N)

K = constant related to the impeller type

V_e = velocity in the impeller eye ft/s (m/s)

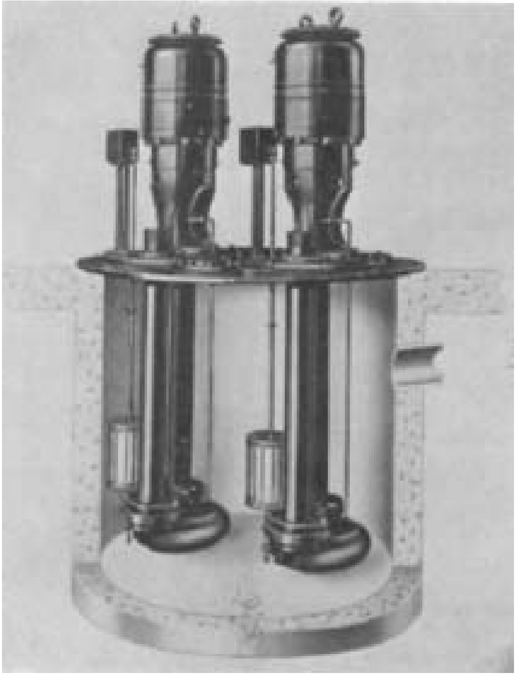


FIGURE 123 A duplex sump pump (Economy Pump)

A_e = net eye area, in^2 (cm^2)
g = 32.2 ft/s^2 (9.81 m/s^2)

The constant K is 1.0 for a fully radial-flow impeller, less than 1.0 for a mixed-flow impeller, and essentially zero for a fully axial-flow impeller.

Under normal operating conditions, the upward thrust caused by the change of momentum is hardly significant in comparison with the downward thrust caused by the unbalanced pressures acting on the single-suction impeller. Consider the example of an impeller with the following characteristics:

Capacity	2,500 gpm (568 m^3/h)
Total head	231 ft (70.4 m)
Net pressure	100 lb/in^2 (6.89 bar)
Unbalanced eye area	40 in^2 (258 cm^2)

If we neglect the effect of the pressure distribution on the shrouds of the impeller, the downward thrust is

in USCS units $$100 \times 40 = 4000 \text{ lb}$$

in SI units $$6.89 \times 10^5 \times \frac{258}{10{,}000} = 17{,}800 \text{ N}$$

The upward force caused by the change of momentum is

in USCS units
$$\frac{1.0 \times \left(\dfrac{2500 \times 0.32}{40}\right)^2 \times 40}{64.4 \times 2.31} = 107.5 \text{ lb}$$

in SI units
$$\frac{1 \times \left(\dfrac{568 \times 10{,}000}{258 \times 3600}\right)^2 \times 258}{2 \times 9.81 \times 1.02} = 481 \text{ N}$$

This is certainly negligible relative to the downward hydraulic axial thrust.

The situation is quite different, however, during the startup of a vertical pump. Although the motor may get up to full speed in just a few seconds, it takes a certain amount of time for the total head to increase from zero that corresponding to the normal operating capacity. Consequently, the pump will be operating in a very high capacity range. Since the upward force caused by the change of momentum varies as the square of the capacity while the downward axial thrust caused by pressure differences is very low, there can be a momentary net upward force, or upthrust. This means that thrust bearings intended to accommodate the axial thrust of vertical pumps must be capable of accommodating some thrust in the upward direction in addition to the normal downward thrust. This is particularly true for pumps with relatively low heads per stage and with short settings because in these units the rotor weight does not at all times compensate for the upthrust from the change in momentum.

Particular care must be exercised in defining the range of vertical movement allowed for the thrust bearings because any such movement must remain within the displacement limits of any mechanical seal used in the pump.

In rather rare cases, a close-coupled vertical pump is operated at such a high capacity that a continuous net upthrust is generated. Such operations can damage the pump because the line shaft is operated in compression and may buckle, causing vibration and bearing wear. The manufacturer's comments should be invited if such an operation is contemplated.

Shaft Elongation in Vertical Pumps The elongation of a vertical pump shaft is caused by three separate phenomena: (1) the tensile stress caused by the weight of the rotor, (2) the tensile stress caused by the axial thrust, and (3) the thermal expansion of the shaft.

In most cases, the tensile stress created by the axial thrust is several times greater than that created by the weight. In a typical example of a 16,000-gpm (3636-m^3/h) pump designed for a 175-ft (53.3 m) head and 50 ft (15.25 m) long, the elongation caused by the weight of a 1600-pound (726 kg) impeller will be of the order of 0.0033 in (0.084 mm). The elongation caused by the axial thrust will be approximately 0.0315 in (0.8 mm).

The elongation caused by thermal expansion has to be considered from two angles. First, if the shaft and the stationary parts are built of materials that have essentially the same coefficient of expansion, both will expand equally and no significant relative elongation will take place. Second, if the pumps are operated in essentially the same range of temperatures in which they were assembled, no significant relative expansion will take place, even if dissimilar coefficients of expansion are involved. Whatever the case, the pump manufacturers take these factors into consideration by providing the necessary vertical end-play between the stationary and rotating pump components.

Loads on Foundations of Vertical Pumps If the motor support is integral with the pump discharge column, as in Figures 110, 112, and 113, and if the hydraulic thrust is carried by the motor thrust bearing, this thrust is not additive to the deadweight of the pump and of its motor plus the weight of the water contained in the pump, insofar as the load on the foundations is concerned. This is because the pump and motor mounted in this fashion form a self-contained entity and all internal forces and stresses are balanced within this entity. If the pump and motor are supported separately, however, as in Figure 124, and are joined by a rigid coupling that transmits the pump hydraulic thrust to the motor thrust bearing, the foundations will carry the following loads when the pump is running:

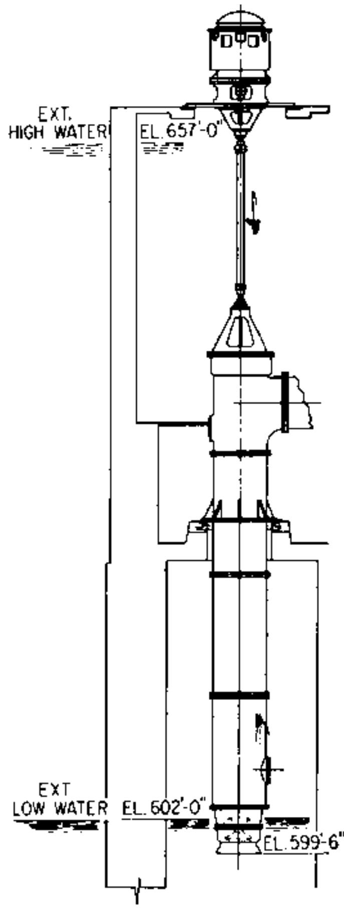


FIGURE 124 The vertical pump and driving motor supported separately (Flowserve Corporation)

- *The foundation supporting the motor*: the weight of the motor plus the weight of the pump rotor plus the axial hydraulic thrust developed by the pump
- *The foundation supporting the pump*: the weight of the pump stationary parts plus the weight of the water in the pump, including the water in the discharge pipe supported by the foundation, less the axial hydraulic thrust developed by the pump

Since when the pump is idle the foundation supporting the pump is not benefited by the reduction in load equivalent to the axial hydraulic thrust, both running and idle operating conditions must be considered in this case.

Typical Arrangements of Vertical Pumps A pump is only part of a pumping system. The hydraulic design of the system external to the pump will affect the overall economy of the installation and can easily have an adverse effect upon the performance of the pump itself. Vertical pumps are particularly susceptible because the small floor space occupied by each unit offers the temptation to reduce the size of the station by placing the units closer together. If the size is reduced, the suction arrangement may not permit the proper flow of water to the pump suction intake. Recommended arrangements for vertical pumps are discussed in Section 10.1.

SPECIAL PURPOSE PUMPS

In addition to the more or less general purpose pumps described on the preceding pages, literally hundreds of centrifugal pumps are intended for very specific applications. Although it is impossible to describe every one of these special purpose designs, many of them are discussed and illustrated in Sections 9.1 through 9.22, covering a variety of pump services. However, several specific designs are seeing rapidly increasing usage and so are discussed in greater detail here.

Submersible Motor-Driven Wet-Pit Pumps The installation of conventional vertical wet-pit pumps with the motor located above the liquid level may require a considerable length of drive shafting, particularly in the case of deep settings. The addition of this shafting, of the many line bearings, and possibly of an external lubrication system may represent a major portion of the total installed cost of the pumping unit. Furthermore, shaft alignment becomes more critical, and shaft elongation and power losses increase rapidly as the setting is increased, especially for deep-well pumps.

A great variety of submersible motors have been developed to obviate these shortcomings. They are described in Subsection 6.1.1, and a classification of the various types of such motors is presented in Figure 26 of that subsection. Submersible wet-pit pumps eliminate the need for extended shafting, shaft couplings, a mechanical seal or stuffing box, a subsurface motor stand, and, in some cases, an expensive pump house. Both vertical turbine and volute-type wet-pit pumps may be so driven.

Figures 125 and 126 illustrate, respectively, the external appearance and a cross section of a vertical turbine pump driven by a submersible motor located at the bottom of the pump. The pump suction is through a perforated strainer located between the motor and the first-stage impeller bowl. There is, of course, no shafting above the pump and the pump-and-motor unit is supported by the discharge pipe only. No external lubrication is required. The motor is completely enclosed and oil-filled and is provided with a thrust bearing to carry the pump downthrust. A mechanical seal is provided at the motor shaft extension, which is connected to the pump shaft with a rigid coupling. Only a discharge elbow and the electric cable connection are seen above the surface support plate. On occasion, this type of pump is used horizontally as a booster pump in a pipeline, and in such cases the elbow at the discharge is eliminated.

Vertical wet-pit volute-type sump pumps can be obtained with close-coupled submersible motors for drainage, sewage, process, and slurry services. Figure 127 illustrates dual submersible sewage pumps in a below ground collecting tank. The pumps (see Figure 128) are supported by guide rails that make it possible to lower and raise the pumps by means of a chain hoist. During this operation, the discharge pipe is connected and disconnected without dewatering the tank. Other arrangements use foot-supported pumps with rigid discharge piping.

Motors used for this type of construction are usually hermetically sealed, employing a double mechanically sealed oil chamber with a moisture-sensing probe to detect any influx of conductive liquid past the outer seal. Controls to start and stop the pump motors can be either an air compressor bubbler system or level-sensing switches that tilt when floated (refer to Figure 127).

Small portable pumps are available with flexible discharge hoses and built-in water-level motor control switches activated by trapped air pressure. Motors for these pumps are usually oil-filled and have a single mechanical shaft seal but are also available in a hermetically sealed design. The submersible motors are cooled by the liquid in which they are immersed and therefore should not be run dewatered, although some motors can operate for short periods (10 to 15 min.) this way.

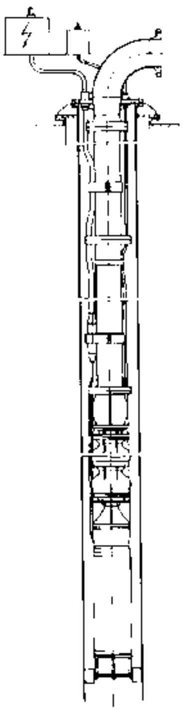


FIGURE 125 Vertical turbine pump driven by a submersible motor (Flowserve Corporation)

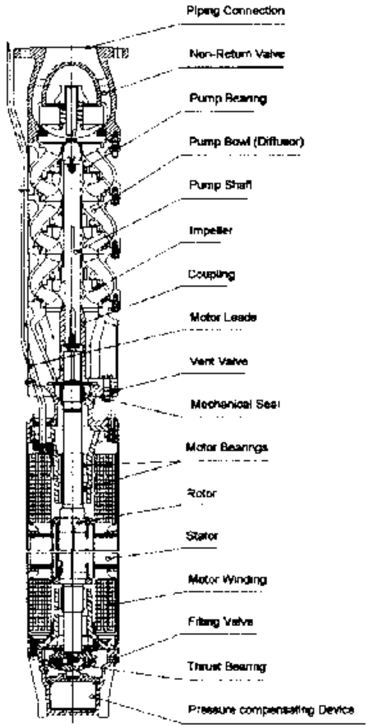


FIGURE 126 Cross-sectional view of a submersible pump (Flowserve Corporation)

FIGURE 127 Dual submersible sewage pumps in a below ground collecting tank (Flowserve Corporation)

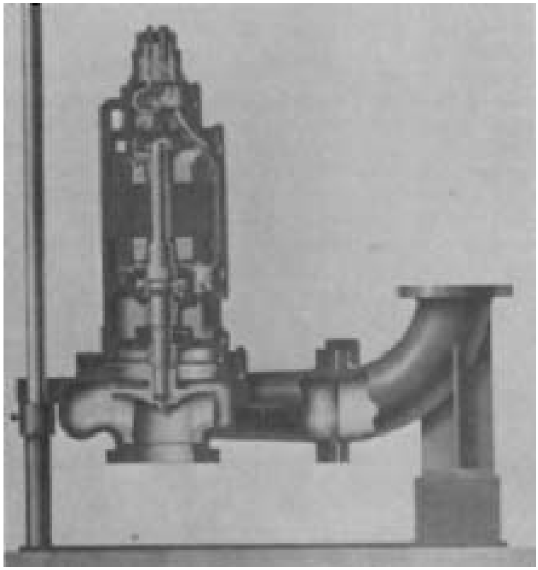


FIGURE 128 A section of a submersible sewage pump shown in Figure 127 (Flowserve Corporation)

In a pump-motor combination (see Figure 129), the motor is cooled by the pumped liquid as it moves through a passage around the sides of the motor. This design also uses a pressurized oil-seal chamber to assure positive sealing.

Sealless Pumps Completely leakproof pumps are available for pumping corrosive, volatile, radioactive, and otherwise hazardous liquids. Canned motor pumps (see Figure 130) are an assembly of a standard centrifugal pump and a squirrel-cage induction motor in a hermetically sealed unit. Modifying a recirculating flow system in a canned motor pump can allow it to be used in applications at up to 1000°F (538°C).

Magnetic drive pumps (see Figure 131) are an assembly of a rotor, an impeller, product lubricated bearings, and a magnetic carrier inside an isolation shell or diaphragm. This rotor is driven by magnets outside the shell or diaphragm. No mechanical connection exists between the driven magnets and the driving magnets. No seals exist and thus we have the term "sealless." Sealless pumps are described in detail in Subsection 2.2.7.

Straight-Radial-Vane High-Speed Pumps For handling volatile liquids at low flow rates and high heads, the straight-radial-vane impeller in a diffuser casing offers several advantages. Volatile, low-specific-gravity, poor-lubricity liquids require larger running clearances, which is not possible with conventional high-head multistage centrifugal or positive displacement pumps. High-head pumping of these liquids can be handled by operating this completely open impeller at very high speeds through an integral gear increaser and with very large impeller-to-casing clearance, typically 0.030 to 0.070 in (0.76 to 1.8 mm). Tests of one manufacturer's design have shown this clearance can increase to 0.125 in (3.2 mm) with virtually no change in performance, and consequently there is no need to provide adjustment for impeller axial clearance. A pump of this design can also run "dry," as liquid lubricity is not required to lubricate the bearings, which can be separately lubricated (providing the mechanical seal is lubricated).

Figure 132 illustrates one manufacturer's design of a radial-vane high-speed pump. The impeller rotates in a circular casing that has a single emission point leading to a conical diffusion section. The advantage of this type of casing is that the conversion to pressure occurs outside the circular housing, thus eliminating any recirculation forces that

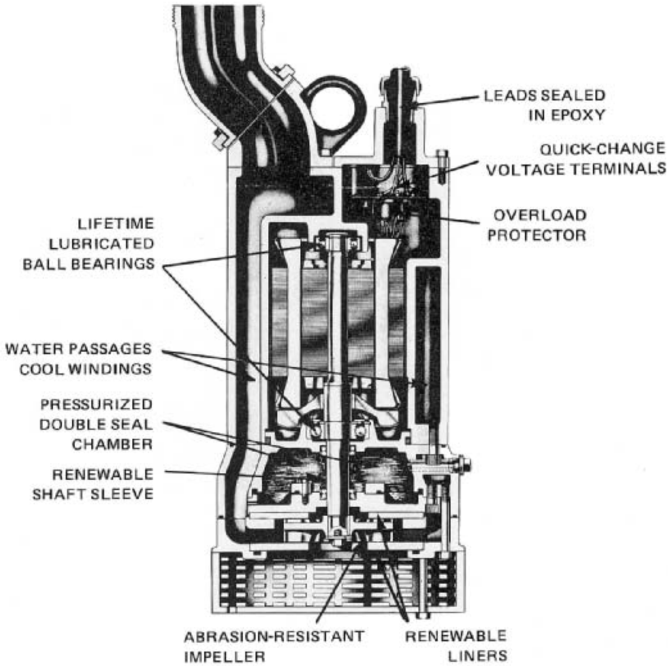


FIGURE 129 A portable submersible pump (Peabody Barnes)

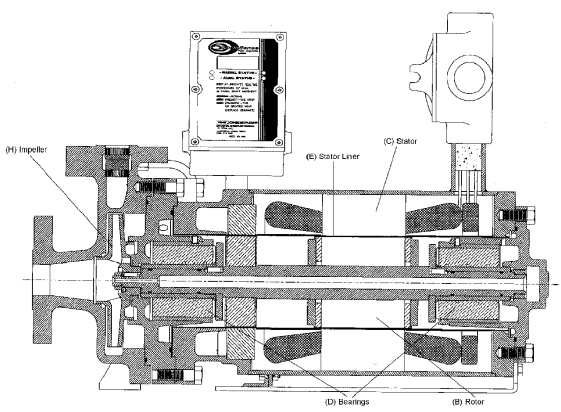


FIGURE 130 Typical canned motor pump (Ref. Subsection 2.2.7) (Crane Chempump)

would require a close clearance between the impeller and casing. For higher flow ranges, a double emission point design, as shown in Figure 133, is used. This additional emission acts like a double-volute casing in conventional centrifugal pumps.

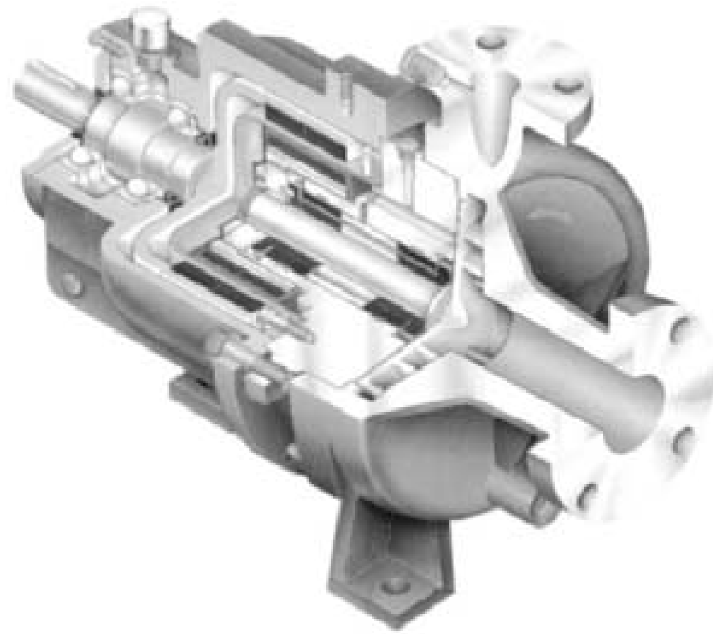


FIGURE 131 Typical magnetic drive sealless pump (Ref. Subsection 2.2.7) (Flowserve Corporation)

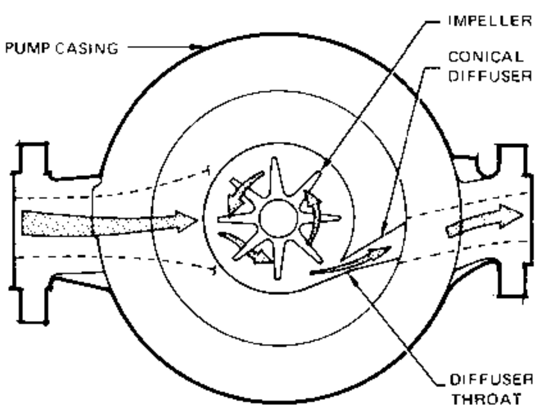


FIGURE 132 A straight-radial-vane impeller in a single emission-point, conical-diffuser circular casing (Sundyne Corporation)

Figure 134 is a sectional view of this pump with an integral speed increaser. Because of the high rotative speed, a single-stage pump can achieve the head normally associated with multistage centrifugal or positive displacement pumps. With this smaller liquid end and an integral gear box, cost and space savings can be appreciable.

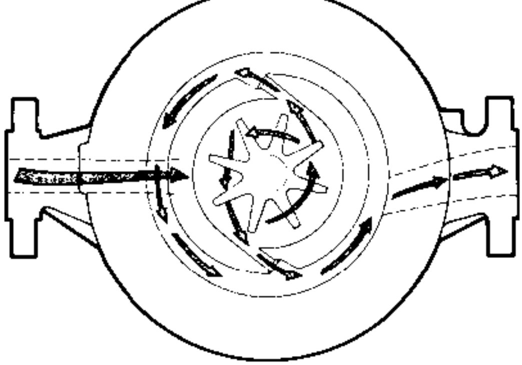


FIGURE 133 A straight-radial-vane impeller in a double-emission-point, conical-diffuser circular casing (Sundyne Corporation)

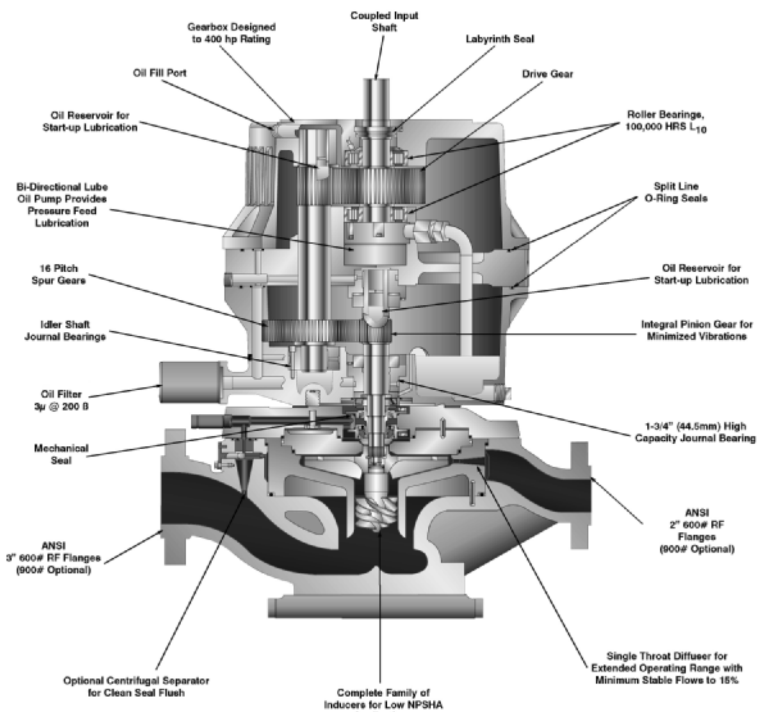


FIGURE 134 A section of a straight-radial-vane impeller and inducer in a conical-diffuser circular casing with an integral gear increaser with the following: (1) pump casing, (2) impeller, (3) impeller bolt, (4) impeller tab washer, (5) inducer, (6) diffuser, (7) diffuser cover, (8) upper throttle bushing, (9) seal housing, (10) single shaft sleeve, (11) pump seal rotating face, (12) gearbox seal rotating face, (11) pump mechanical seal, (14) gearbox mechanical seal, (15) separator orifice, (16) separator fitting, (17) gearbox output housing, (18) gearbox input housing, (19) bearing plate, (20) interconnecting shaft, (21) low-speed shaft, (22) input gear, (23) lower idler gear, (24) high-speed shaft, (25) pinion gear, (26) idler shaft, (27) O-ring packings (Sundyne Corporation).