

CHAPTER ONE

CENTRIFUGAL

PUMP

FUNDAMENTALS

Centrifugal Pump Fundamentals

In the previous chapter, we have studied different types of centrifugal and positive displacement pumps and their various distinguishing external features. In this chapter, we will learn about various types of internal components of centrifugal pumps.

The diversity among pumps does not only limit itself to the external features of the machines but also extends to its internal components. This is especially true in the case of centrifugal pumps. The basic components are essentially the same in almost every design but depending on the design and its applications, the construction features of the internal components differ to meet various requirements.

2.1 Impellers

The impeller of the centrifugal pump converts the mechanical rotation to the velocity of the liquid. The impeller acts as the spinning wheel in the pump.

It has an inlet eye through which the liquid suction occurs. The liquid is then guided from the inlet to the outlet of the impeller by vanes. The angle and shape of the vanes are designed based on flow rate. The guide vanes are usually cast with a back plate, termed shroud or back cover, and a front plate, termed front cover.

Impellers are generally made in castings and very rarely do come across fabricated and welded impellers.

Impellers can have many features on them like balancing holes and back vanes. These help in reducing the axial thrust generated by the hydraulic pressure. This is covered in Chapter 4.

In order to reduce recirculation losses and to enhance the volumetric efficiency of the impellers, they are provided with wearing rings. These rings maybe either on the front side or on both the front and backsides of the impeller. It is also possible to have an impeller without any wearing rings.

The casting process, as mentioned above, is the primary method of impeller manufacture. Smaller size impellers for clean water maybe cast in brass or bronze due to small section thickness of shrouds and blades. Recently, plastic has also been introduced as casting material.

For larger impellers and in most of the applications, cast iron is the first choice of the material. The grade used is ASTM A-48-40 (minimum tensile strength is 40 000 psi or 2720 kgs/cm²).

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This is used for a maximum peripheral speed of 55m/s and a maximum temperature of 200°C. When the temperature exceeds 200°C, carbon steel castings of the grade A-216 WCA/WCC are recommended.

The adequacy of cast steel is dependent on its usage in handling of abrasives like ash, sand, or clinker. In such cases, the impellers could also be cast in 12% Cr steels (A-743 CA15). Stainless steel castings (A-744 CF8M) are used for their high corrosion resistance and for low-temperature applications. In case of low-temperature applications (not lower than 100°C), ferritic steel castings containing 3.5% nickel can be used (A-352 LC3). For temperatures until 200°C, A-276-Type 304 castings are used.

Marine applications may demand castings made from aluminum bronze (B-148 – Alloy C 95 800). Copper bronze casting grade adopted is B 150-Alloy 63 200. Caustic Acid solutions and other corrosive liquids may demand special materials. For example, Sulphuric acid (concentration 67% and at a temperature of 60–70°C) needs hi-silicon cast iron (15% Si).

During the casting process, it is important to keep the liquid contact surfaces of the impeller as smooth as possible. Thus, the composition of the core sand mixture and the finish of the core play an important part in the casting process. Largely, the relative smoothness of the liquid path determines the efficiency of the pump.

In a closed impeller design, the contact surface area of the metal with the liquid is higher which results in high friction losses. When the impeller's diameter is large, the problem becomes more acute and so there is a higher demand for smoother surface. Friction losses are related to 5th power of the diameter.

Subsequent to the casting and surface finishing operations, the impellers are dynamically balanced. The limits of residual unbalance are generally specified in ISO 1940, or even in API, which has a stricter limit. The balance of impellers alone is insufficient. Once the pump rotor components are ready, these should be mounted assembly wise on the balancing machine and balanced to stated limits.

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2.1.1 Construction of the Impellers

There are three types of construction seen in an impeller. These are based on the presence or absence of the impeller covers and shrouds.

The three types (Figure 2.1) are:

1. Closed
2. Semi-open
3. Open

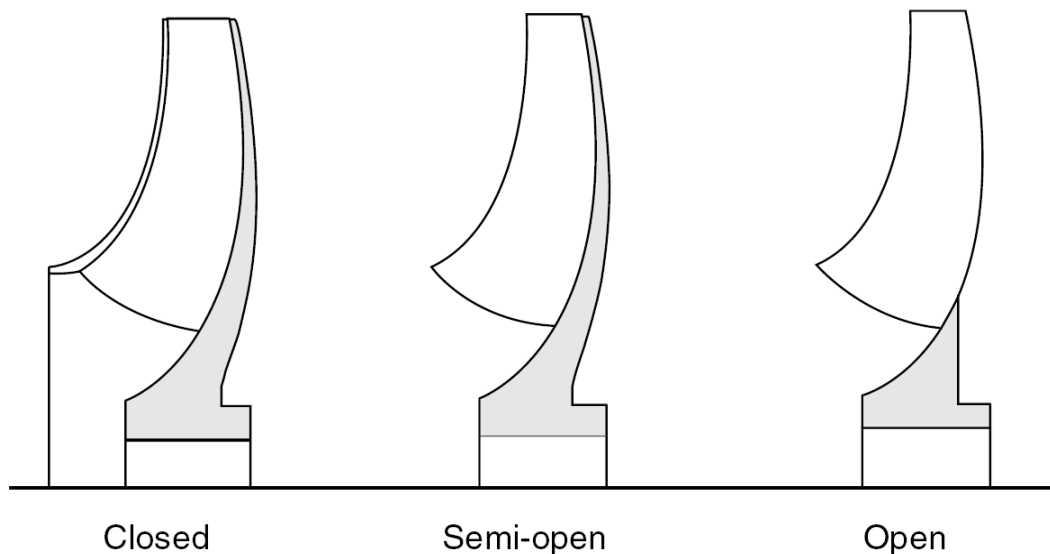


Figure 2.1 – Types of Impellers

2.1.1.1 Closed Impellers

The closed impeller consists of radial vanes (typically 3–7 in number), which are enclosed from both sides by two discs termed 'shrouds'. These have a wear ring on the suction eye and may or may not have one on the back shroud. Impellers that do not have a wear ring at the back typically have back vanes. Pumps with closed type impellers and wear rings on both sides have a higher efficiency.

2.1.1.2 Semi-Open Impellers

The semi-open type impellers are more efficient due to the elimination of disk friction from the front shroud and are preferred when the liquid used may contain suspended particles or fibers. The axial thrust generated in semi-open impellers is usually higher than closed impellers.

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2.1.1.3 Open Impellers

There are three types of back shroud configurations. The first one is a fully scalloped open impeller as shown in Figure 2.2.



Figure 2.2 – Fully Scalloped Open Impeller

The back shroud is almost taken out and thus the axial thrust caused by the hydraulic pressure is almost eliminated.

The second type is known as the partially scalloped open type of impeller as shown in Figure 2.3. It experiences a greater axial thrust than the fully scalloped open impeller.

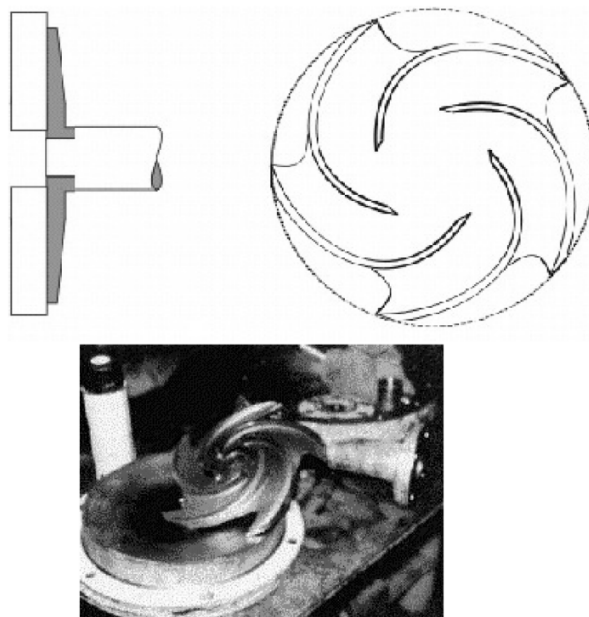


Figure 2.3 – Partially Scalloped Open Impeller

However, this has higher efficiency and head characteristics.

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The third type is known as the fully back shroud open impeller (Figure 2.4) where there is an open impeller with a full back shroud. It normally has almost 5% higher efficiency than a fully scalloped impeller, though it has diminished head generation capabilities.

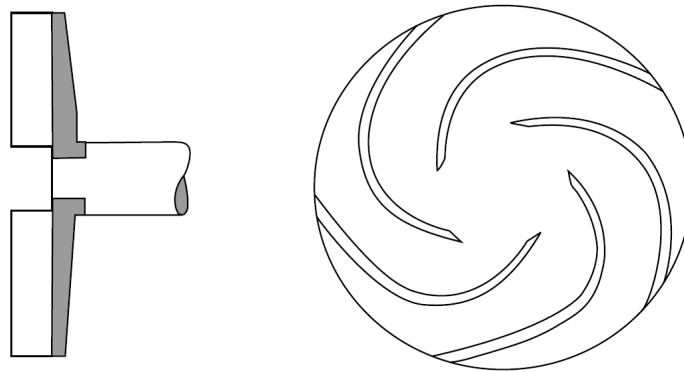


Figure 2.4 – Fully Back Shroud Open Impeller

The fully shrouded open impellers experience the maximum axial thrust among the open impeller types. To reduce this effect, back vanes are provided to relieve the hydraulic pressure that generates the axial thrust.

The vortex or non-clog impellers (Figure 2.5) are the fully shrouded open type of impellers. These are used in applications where the suspended solid's size maybe large or the solid's maybe of crystals and fibers type. The vortex impeller does not impart energy directly to the liquid. Instead it creates a whirlpool, best described as a vortex. The vortex in turn imparts energy to the liquid or pumpage. The location of the impeller is usually above the volute, so it experiences hardly any radial forces. This allows extended operation of the pump even at closed discharge conditions.

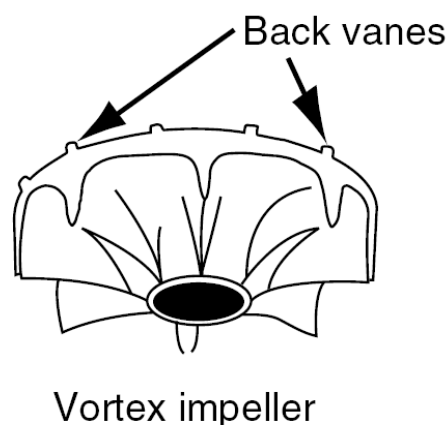


Figure 2.5 – Vortex Impeller

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Some of the other non-clogging designs of impellers in the closed and semi-open types are shown in Figures 2.6 and 2.7.

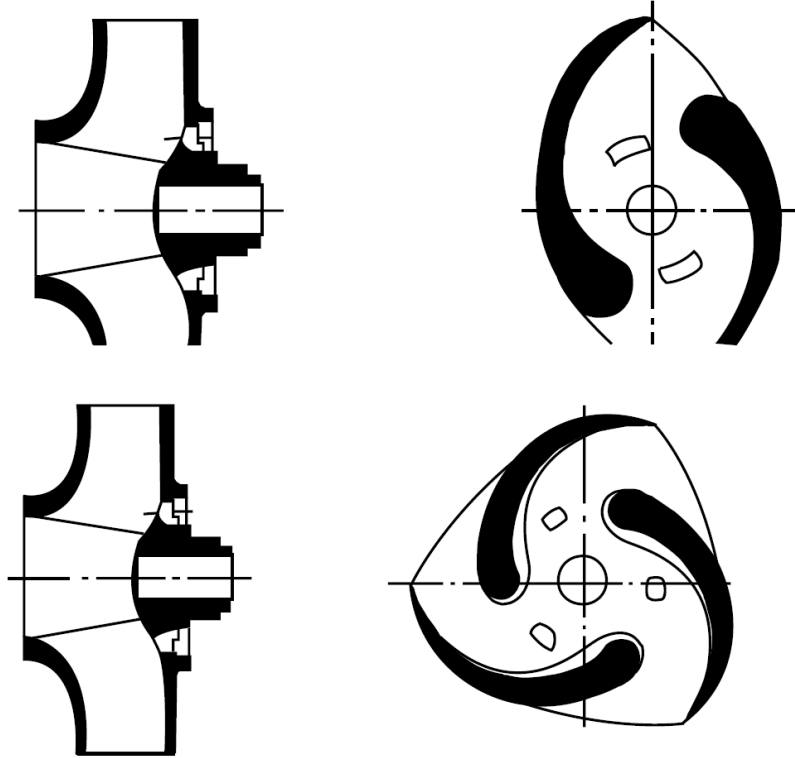


Figure 2.6 – 2 & 3 Passage Closed Non-Clog Impellers

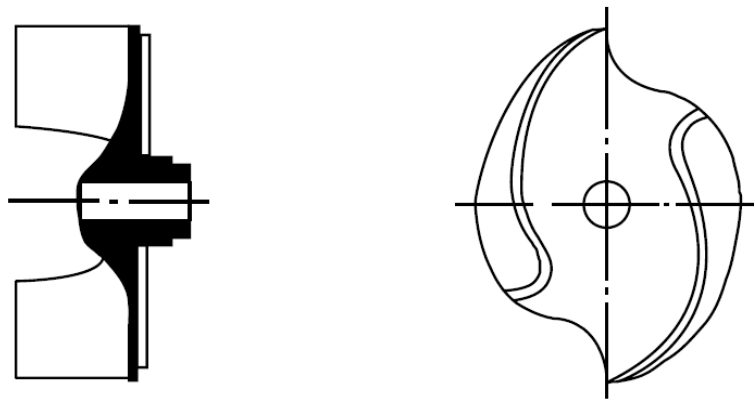


Figure 2.7 – Semi-Open 2-Passages Non-Clog S-Shaped Impeller

In general, most of the open impellers are of the partially scalloped and fully shroud types. Fully open impellers are rarely used because of its lower efficiency and the bending load on the vanes.

2.1.2 Impeller Suction

In general, an impeller has one eye or a single opening through which liquid suction occurs. Such impellers are called as single-suction impellers. Pumps with a single suction impeller (impeller having suction cavity on one side only) are of a simple design but the impeller is subjected to higher axial thrust imbalance due to the flow on one side of the impeller only.

In certain pumps, the flow rate is quite high. This can be managed by having one impeller with two suction eyes. Pumps with double-suction impeller (impeller having suction cavities on both sides) has lower NPSH-r than single-suction impeller. Such a pump is considered hydraulically balanced but is susceptible to an uneven flow on both sides if the suction piping is improper.

Generally, flows that are more than 550 m³/h (or 153 l/s) may necessitate a double suction impeller (Figure 2.8).

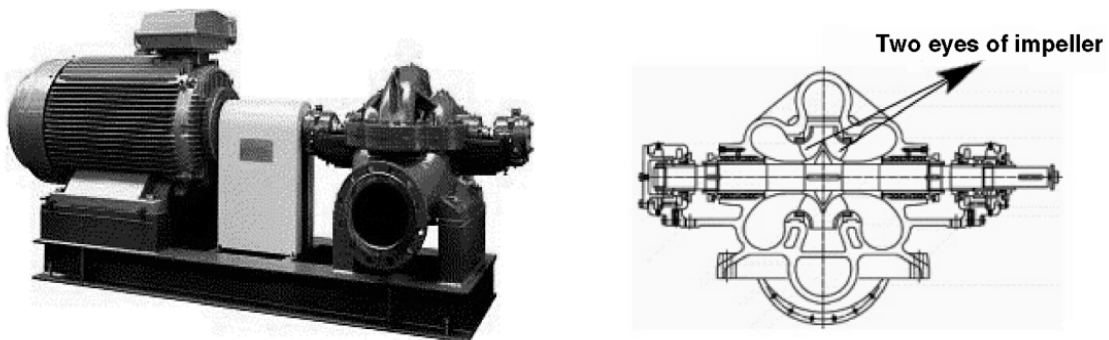


Figure 2.8 – Pump with Double Suction Impeller

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2.1.3 Flow Outlet from Impeller

The flow direction of the liquid at the outlet of the impeller can be:

- Radial (perpendicular to inlet flow direction)
- Mixed
- Axial (parallel to inlet flow direction)

The flow outlet is determined by an important parameter called as the specific speed of the pump. As the specific speed of a pump design increases, it becomes necessary to change the construction of the impeller from a radial type to an axial type (Figure 2.9, and Figure 2.10 for mixed flow type).

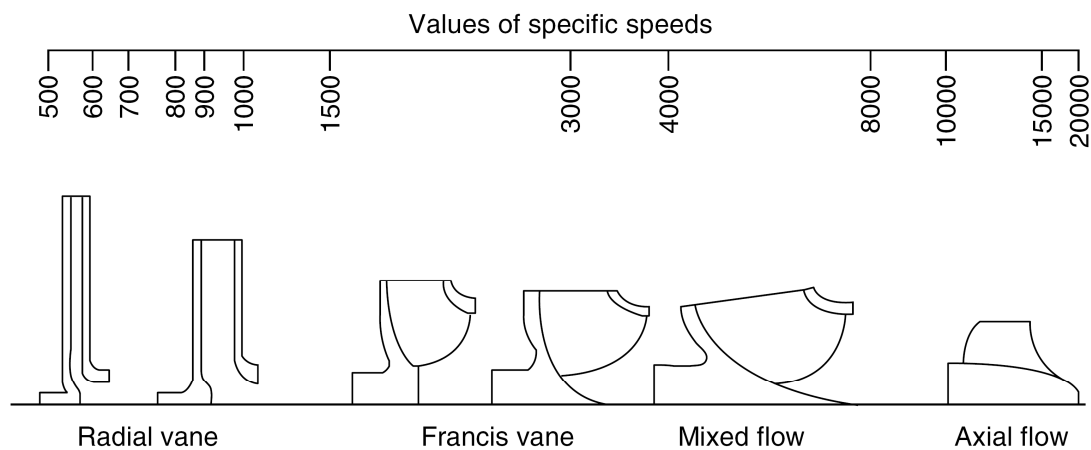


Figure 2.9 – Shapes of Impellers According to Their Specific Speeds

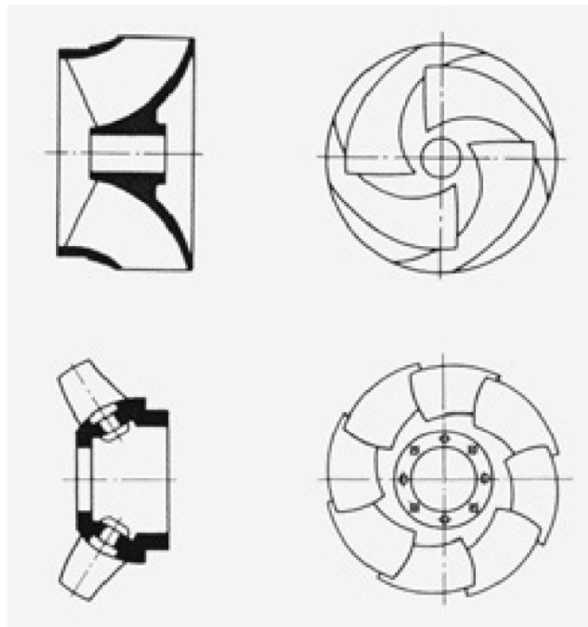


Figure 2.10 – Mixed Flow Impeller and Propeller Vaned Mixed Flow Type Impeller

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Generally, it can be said that for low specific speeds (low flows and high heads) radial impellers are used whereas for high specific speeds (high flows and low heads) axial (propeller) impellers are used (refer Figure 2.11).

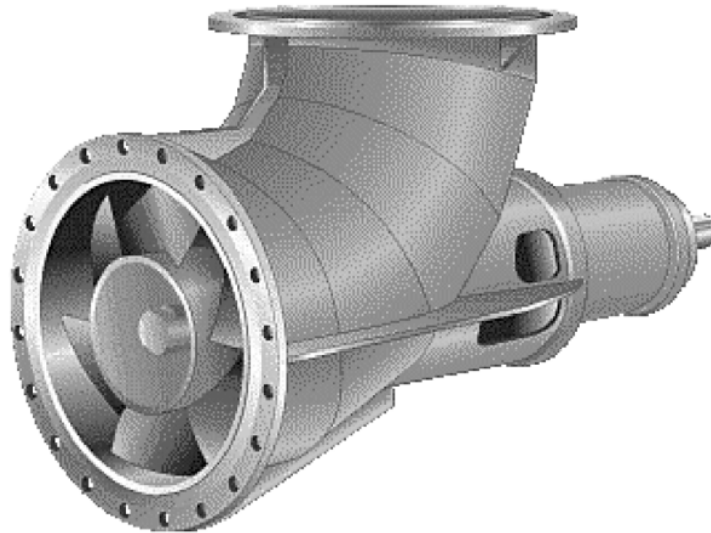


Figure 2.11 – Pump with Axial Flow Impeller

2.2 Pump Casings

At the impeller outlet, the velocity of the liquid can be as high as 30-40m/s. This velocity has to be reduced within a range of 3-7m/s in the discharge pipe.

Velocity reduction is carried out in the pump casing by recuperators. The kinetic energy in the liquid at the outlet is converted to pressure energy by the recuperators.

Here, energy conversion has to be undertaken with a minimal loss to have an insignificant effect on pump efficiency.

Some of the recuperators are:

- Concentric casing
- Volute casing
- Diffuser ring vanes
- Diagonal diffuser vanes
- Axial diffuser vanes.

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2.2.1 Concentric Casing

Concentric casings are usually found in single-stage centrifugal pumps and in the last stage of multistage pumps (Figure 2.12).

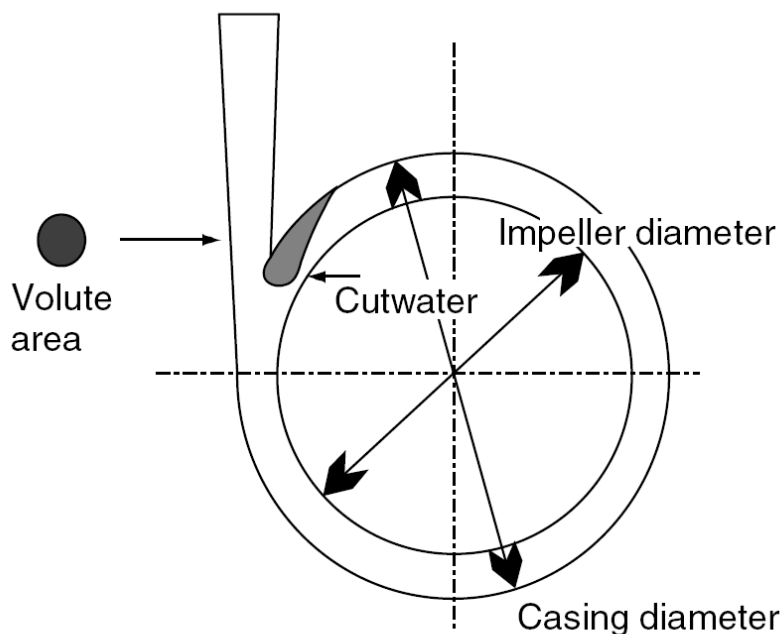


Figure 2.12 – Concentric Casing Pump

In some of the earlier designs of a single-stage centrifugal pump for larger heads, an annular delivery passage is used in conjunction with a diffuser ring. The liquid outlet is through a conical diffuser.

The ratio of the impeller diameter to the diameter of the casing is not less than 1.15 and not more than a ratio of 1.2.

The volute width is designed to accommodate the maximum width of the impeller. The capacity at the most efficient point of operation is controlled by the volute diameter (d).

To minimize the recirculation in the volute, a cutwater tongue is used. In addition, this helps in significantly reducing the radial loads on the shaft.

In pumps with a specific speed of less than 600 (US-gpm, feet, rpm), the concentric casing provides higher efficiency than a conventional volute casing. Above the specific speed, N_s of 600, the efficiency progressively drops.

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The concentric casings are used:

- For less flow and higher head; low specific speeds N_s is in the range of 500–600
- Where the pump casing has to accommodate several impeller sizes
- Where pump has to use a fabricated casing
- Where volute passage has to be machined from a casting
- Where foundry limitations result in higher impeller width

2.2.2 Volute Casing

Volute casings when manufactured with smooth surfaces offer insignificant hydraulic losses. In pumps with volute casings, it is possible to trim down impeller vanes and shrouds with minimal effect on efficiency.

In volute casings, the kinetic energy is converted into pressure only in the diffusion chamber immediately after the volute throat. The divergence angle is between 7° and 13° .

The volutes encountered can be of various cross-sections and these are shown in Figure 2.13.

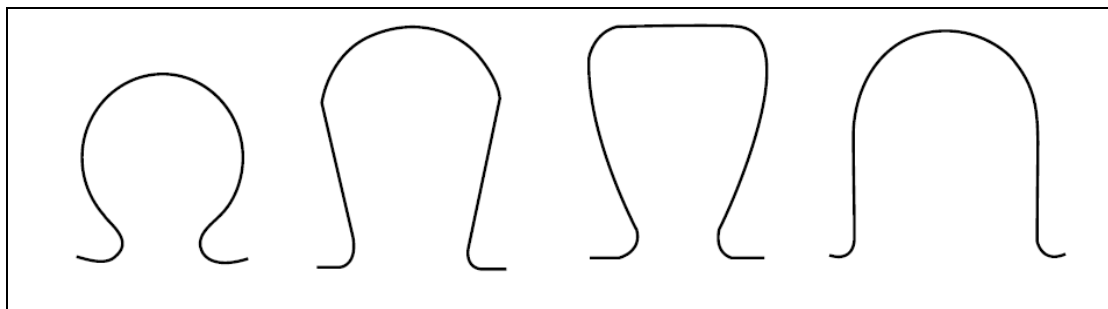


Figure 2.13 – Different Volute Cross-Section Shapes

The first two profiles are of circular cross-section; the third is called as the trapezoidal cross-section, which is typically found in single-stage pumps. The last profile is the rectangular cross-section.

The rectangular section is used in small single-stage pumps and in multistage pumps. It is economical to manufacture due to its low pattern cost and production time. The hydraulic losses are minimal in the specific speed range of less than 1100.

Volute casings are manufactured in various designs and these are:

- Single volute casing
- Double volute casing

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2.2.2.1 Single Volute Casing

Single volute designs are the most commonly found designs and those designed on the basis of constant velocity are the most efficient among all types. They are easy to cast and less expensive to manufacture.

In a single volute casing, the pressure distribution is balanced only at the Best Efficiency Point (BEP) of the pump. At other operating points, this leads to a residual radial load on the shaft, which is maximum at shut-off conditions and almost zero at the BEP.

At low flow rates, the pressure distribution is such that the surfaces of the impeller closest to the discharge are acted upon by high pressures. Those on the other side of the cutwater are acted upon by comparatively low pressures (Figure 2.14).

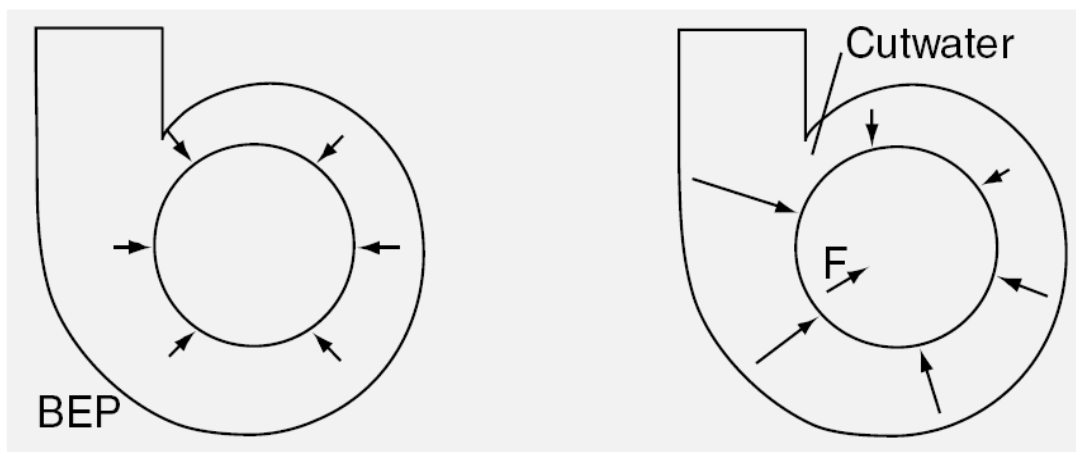


Figure 2.14 – Forces as Generated in a Single Volute

The resulting unbalanced forces can be assumed to be acting at a point 240° from the cutwater and acting in a direction which points to the center of the impeller.

Theoretically, these casings can be used over the entire range of specific speed pumps; however, these are used mainly on low capacity, low specific speed pumps. They can also be used in pumps handling slurries and solids.

2.2.2.2 Double Volute Casing

A double volute casing design is actually two single volute designs combined in an opposed arrangement (Figure 2.15). The total throat area of the two volutes is identical to that which would be used on a comparable single volute design.

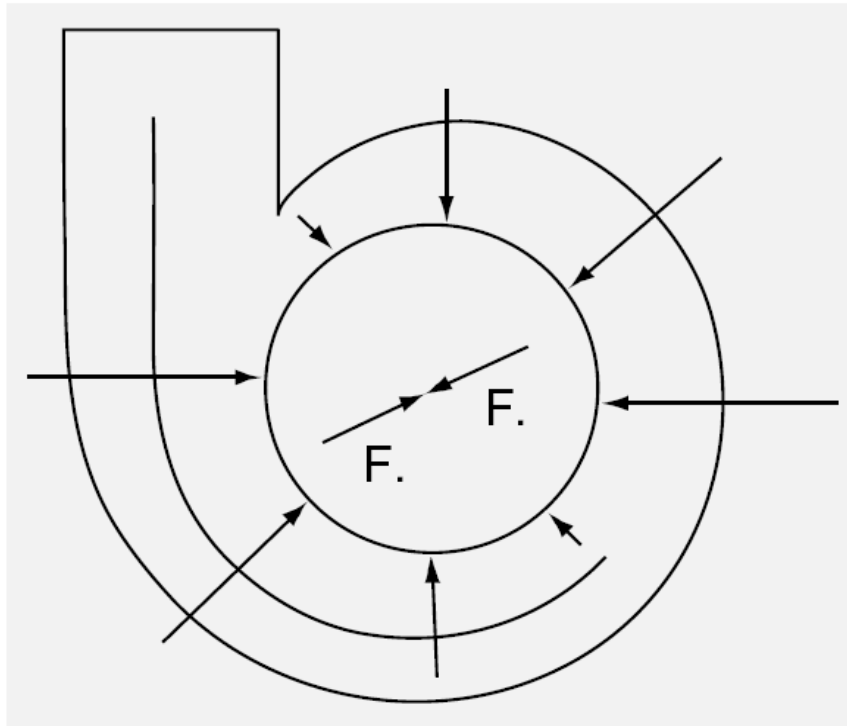


Figure 2.15 – Balance of Forces in Double Volute

Single volute designs inherently generate a radial load on the shaft. The double volute designs limit this radial force to a greater extent.

In this design, the volute is symmetrical about its centerline; however, the two passages carrying the liquid to the discharge flange are not symmetrical. As a result, the pressure forces around the impeller periphery do not cancel and this leads to some radial force.

The hydraulic performance of the double volute is on a par with the single volute design. At the BEP, the efficiency is marginally lower but is higher at operating points; lower and higher than BEP. Thus, for flows over the entire range, the double volute design is preferred.

Therefore, flow rate is the basic criterion that determines the selection of one design over another. For flows under 125m³/h, double volute designs are not used since it becomes difficult to manufacture and clean them in smaller casing. In larger pumps, double volutes are invariably used.

2.2.3 Vaned Diffuser Ring

The Vaned diffuser ring has a series of symmetrically placed vanes forming gradually widening passages (Figure 2.16). This ring comprises of a series of vanes set around the impeller. The flow from the vaned diffuser is collected in a volute or circular casing and is discharged through the discharge pipe.

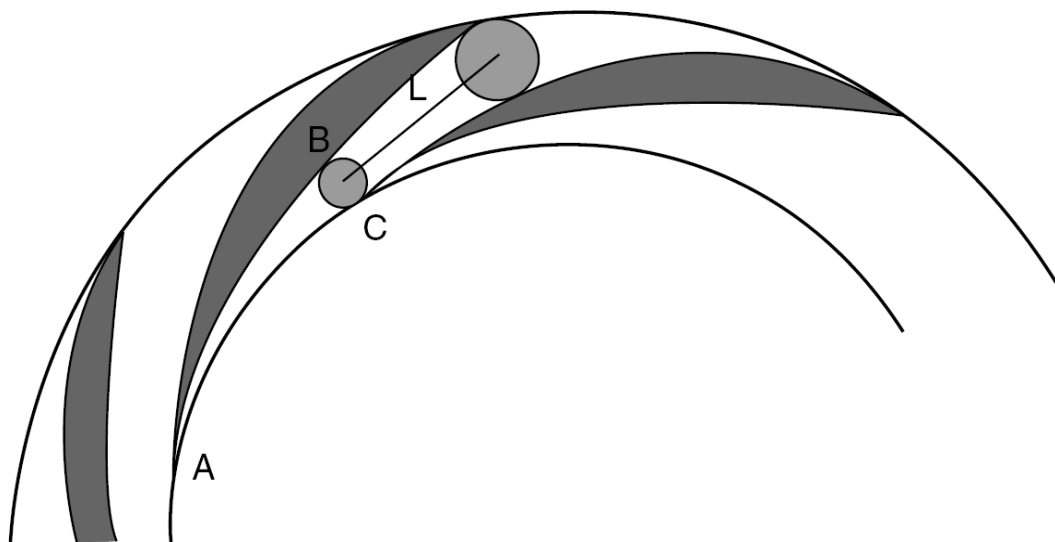


Figure 2.16 – Vaned Diffuser

In these passages, the velocity head is converted to pressure energy. The distance BC shown in Figure 2.16 is called as the throat.

The design of the vaned diffuser is similar to the volute except that there are many throats in a vaned diffuser compared to just one continuous expanding section in the volute.

From the throat onward, the area of the vane channel increases progressively so that, further, a slight increase in pressure takes place. The centerline of the vane channel after the throat may be straight or curved. The straight diffusing channel is slightly more efficient but results in a larger casing.

The vane surface from the vane inlet to the outlet can be shaped like a volute but even a circular arc works fine.

The number of diffuser vanes is usually one more than the impeller vanes, as it is found that the number of diffuser vanes should not be much larger than the number of impeller vanes.

With just one vane more than the impeller, it insures that one impeller passage does not extend over several diffuser passages.

2.2.4 Diagonal Diffuser Vanes

Diagonal diffuser vanes are recuperators for the mixed-flow impeller pumps. The functions of the diagonal diffuser vanes are:

- To change the direction of flow of the liquid leaving the impeller and direct it along the axis of the pump
- To reduce the velocity of liquid and convert it to pressure.

The vanes are disposed in the axial direction forming channels with no sudden changes in cross-section. They make it possible to use impellers of different diameters and breadths so as to extend the range of application of the given model of diffuser.

As the specific speed increases, the profiles of impellers and diffusers change and approximate to the shapes of impellers and diffusers of propeller pumps.

2.2.5 Axial Diffuser Vanes

Axial diffuser vanes are vanes placed behind the impeller of an axial flow pump (Figure 2.17). The functions of these vanes are similar to those of a mixed-flow pump.

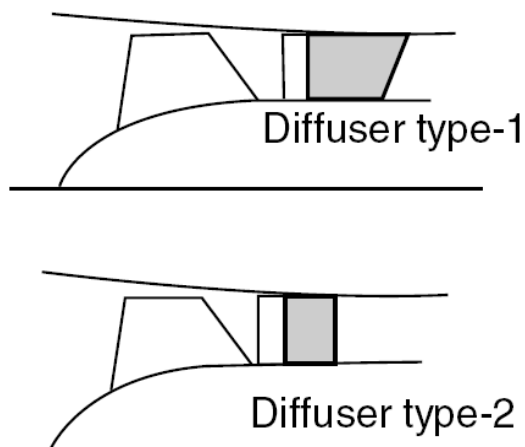


Figure 2.17 – Axial Flow Pumps with Diffuser Behind the Propeller

The vanes usually number 5–8. The lower number is found in pumps with a lower specific speed (diffuser type-1).

The efficiency is influenced to a certain extent by the shape of the diffuser passage.

This depends on the number of vanes and their axial length and the distance between the impeller blades and the diffuser vanes.

Shorter and higher number of vanes (diffuser type-2) for the same flow and head give better efficiency.

2.3 Wearing Rings

The impeller is a rotating component and it is housed within the pump casing. To prevent frictional contact, a gap between these two parts is essential.

So there exists a gap between the periphery of an impeller intake and the pump casing.

In addition, there is a pressure difference between them, which results in the recirculation of the pumped liquid. This leakage reduces the efficiency of the pump.

The other advantages of lower clearance is that reduced leakage prevents erosion due to suction recirculation and also provides a much better rotor dynamic stability to the pump.

As a result, the vibration of the pump operates with lesser vibrations.

Thus, it is essential to keep this gap or clearance at an optimum value. When this clearance is kept at a lower value, the efficiency improves but there is always a risk of contact of the impeller with the casing.

Such a frictional contact may render the impeller or the casing useless which would be a loss since these are expensive parts. Therefore, in the areas of the impeller intake, metallic rings are fitted on the impeller eye as well as on the pump casing.

Accordingly, the wearing ring on the impeller is called as impeller wearing rings and the one fitted on the casing is called as the case wearing ring (Figure 2.18).

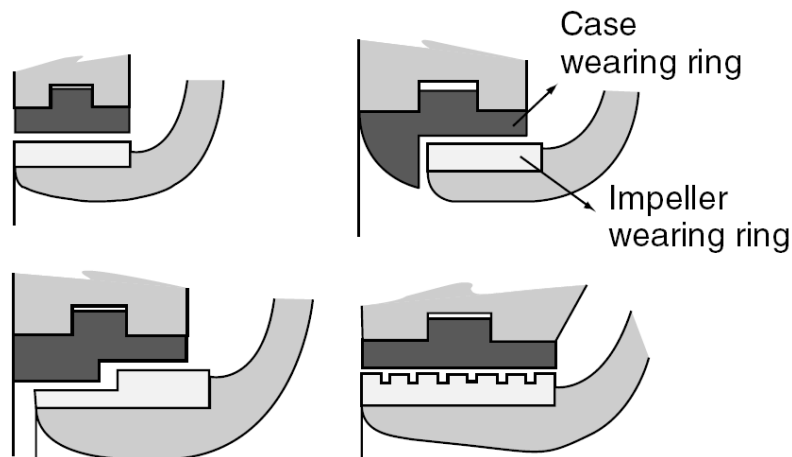


Figure 2.18 – Wearing Rings of Different Types

The cross-section of wearing rings shown in Figure 2.19 is fitted with an impeller eye and is called as the front wearing ring. However, in some cases, wearing rings are installed even at the back shroud of the impeller.

Usually, these are required when impellers are provided with balancing holes in order to minimize the axial thrust coming onto the pump impeller and consequently onto the bearings. The arrangement of the wearing rings on the back of the impeller is shown in Figure 2.19.

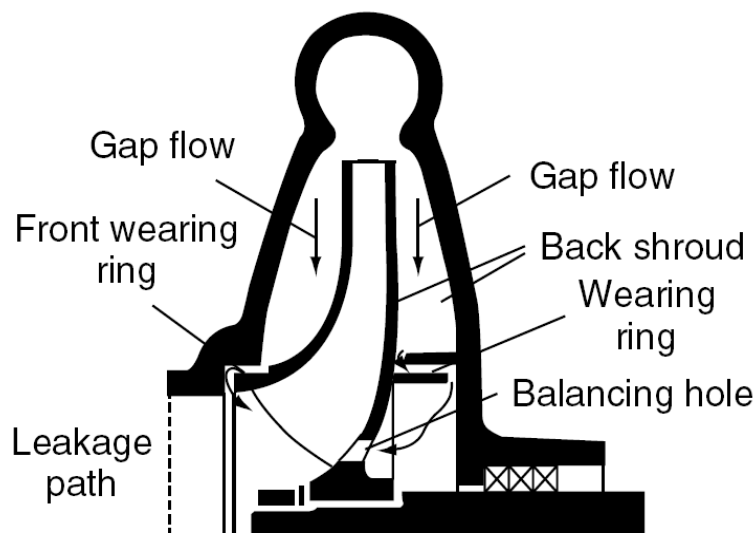


Figure 2.19 – Locations of Wearing Rings

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The material of the wearing rings is selected to prevent seizure on frictional contact. As a result, materials like SS-316 which have galling tendencies are not considered for this application.

The other materials considered favorably are:

- Austenitic Gray Iron Castings – ASTM A-436, Type-1
- Austenitic Ductile Iron Castings – ASTM A-439, Type-D2
- 12% Chrome Steels – AISI 420 (hardenable)
- 18 Cr – 8 Ni Steel Castings – AISI 304
- Copper Alloy Sand Castings (Bronze) – B-584, Alloy C 90 500
- Aluminum Bronze Sand Castings – B-148, Alloy C 95 800
- Monel – K 500
- Nickel 200

The hardness range of the case wearing ring is in the region of 225–275 BHN, whereas the corresponding impeller wearing ring is kept harder by about 50–100 BHN. The range of hardness varies from 325 to 375 BHN.

API 610 standard for centrifugal pumps provides guidelines on the minimum recommended wearing ring clearances for metallic wearing rings. However, these clearances have to be in line with the pumping temperatures, thermal expansion, and galling tendencies of the ring material and the efficiency of the pump.

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For materials that have galling tendencies and pumps operating at temperatures above 260°C diameters are provided with an additional clearance of 5 mils (0.127 mm) over and above those recommended in Table 2.1.

Diameter of Rotating Member at Clearance Inches		Minimum Diametrical Clearance	
From	To	Inches	mm
<2		0.01	0.254
2.000	2.499	0.011	0.28
2.500	2.999	0.012	0.30
3.000	3.499	0.014	0.36
3.500	3.999	0.016	0.41
4.000	4.499	0.016	0.41
4.500	4.999	0.016	0.41
5.000	5.999	0.017	0.43
6.000	6.999	0.018	0.46
7.000	7.999	0.019	0.48
8.000	8.999	0.02	0.51
9.000	9.999	0.021	0.53
10.000	10.999	0.022	0.56
11.000	11.999	0.023	0.58
12.000	12.999	0.024	0.61
13.000	13.999	0.025	0.64
14.000	14.999	0.026	0.66
15.000	15.999	0.027	0.69
16.000	16.999	0.028	0.71
17.000	17.999	0.029	0.74
18.000	18.999	0.03	0.76
19.000	19.999	0.031	0.79
20.000	20.999	0.032	0.81
21.000	21.999	0.033	0.84
22.000	22.999	0.034	0.86
23.000	23.999	0.035	0.89
24.000	24.999	0.036	0.91
25.000	25.999	0.037	0.94

Table 2.1 – Minimum Diametrical Clearance

For 26 in. and above, the diametrical clearance is recommended to be 0.037 in. plus 0.001 in. clearance for every additional inch of impeller diameter.

For example, a 30 in. impeller wearing ring diameter will have a minimum recommended clearance of 0.037 in. + 0.004 in. = 0.041 in.

API is also quite particular in the way the rings need to be fitted to the impellers.

API 610 does not recommend tack welding of rings to impellers. They should be pressed with locking pins or threaded dowels, in the radial or axial direction.

Thermoplastic composite materials are also now being considered as ideal wearing ring materials. They can be applied to the stationary wear part or with the mating component remaining in steel. The use of thermoplastic composite material provides for greater hardness differential between wear parts, the thermoplastic serving as a sacrificial component.

Thermoplastics too have their limitations, however in some cases, they provide the best alternative.

Thermoplastic composite materials are non-galling and have a lower coefficient of friction. They demonstrate excellent wear resistance in clean liquids. Some of these plastics contain reinforced carbon fibers, which greatly enhance the mechanical properties of these plastics. As a result, they can be a direct replacement of the metal wearing rings.

Due to the reduced friction and low galling tendencies, it is possible to almost have half of the clearances that would be considered as optimum with metal wearing rings.

This possibility allows improving pump efficiency especially in low specific speed pumps.

However, the limitations of such materials are that:

- Maximum life is obtained in clean fluids.
- They do not have a wide compatibility with various chemicals.

2.4 Shaft

The pump rotor assembly comprises of the shaft, impeller, sleeves, seals (rotating element), bearings or bearing surfaces, and coupling halves. The shaft, however, is the key element of the rotor.

The prime mover drives the impeller and displaces the fluid in the impeller and pump casing through the shaft.

The pump shaft is a stressed member for during operation it can be in tension, compression, bending, and torsion. As these loads are cyclic in nature, the shaft failure is likely due to fatigue.

The shaft design depends on the evaluations of either the torsion shear stress at the smallest diameter of the shaft or a comprehensive fatigue evaluation taking into consideration the combined loads, the number of cycles, and the stress concentration factors. The design at all times involves sophisticated finite-element computer evaluations.

The shaft design is limited not only to the stress evaluation but is also dependent on other factors such as:

- Shaft deflection
- Key stresses
- Mounted components
- Critical speeds (rotordynamics)

The most common pump shaft material is plain carbon steel, typically BS-970-En 8.

Higher grades include BS-970-En19 or AISI 4140, ASTM-A322, Grade-4140 (quenched and tempered).

Austenitic steel shafts may also be used of grade ASTM A-276, Type 316 and AISI 304.

Some applications like sour water with pH less than 7, drain water or slightly acidic non-aerated liquids, and hydrocarbons containing corrosive aqueous phase may demand shafts made from aluminum bronze material. The recommended grade is B-150-Alloy C 63 200.

Special applications may call for Monel or even Hastalloy C shafts.

The mechanical seals or gland packing, in contact with the shaft, can cause excessive wear due to frictional contact or fretting corrosion. As a sacrificial component, shaft sleeves are used. These are fitted closely onto the shaft; and seals and gland packing are exposed to the sleeve rather than the shaft. It is far less expensive to replace a sleeve than the complete shaft.

The material of construction of the pump sleeves is similar to that of the shaft but the standardization favors the use of SS-316. The portion of the sleeve that is exposed to the secondary seal of the mechanical seals such as an O-Ring or a Teflon wedge is hard coated. The plasma sprayed, hard coating can be of Chrome-oxide, tungsten carbide, or alumina. This offers hardness around 70–72 Rc. The surface is then provided with a ground finish.

2.4.1 Fixable and Rigid Shaft Design

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 hand, 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 satisfactory 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.

PRACTICAL CENTRIFUGAL PUMPS

2.5 Shaft Sleeve

Pump shafts are usually protected from erosion, corrosion, and wear at the seal chambers, leakage joints, internal bearings, and in the waterways by renewable sleeves. Unless otherwise specified, a shaft sleeve of wear, corrosion, and erosion resistant material shall be provided to protect the shaft. The sleeve shall be sealed at one end. The shaft sleeve assembly shall extend beyond the outer face of the seal gland plate. (Leakage between the shaft and the sleeve should not be confused with leakage through the mechanical seal).

2.6 Stuffing Box and Seal Housing

2.6.1 Stuffing Boxes

The stuffing box is a chamber or a housing that serves to seal the shaft where it passes through the pump casing (Figure 2.20).

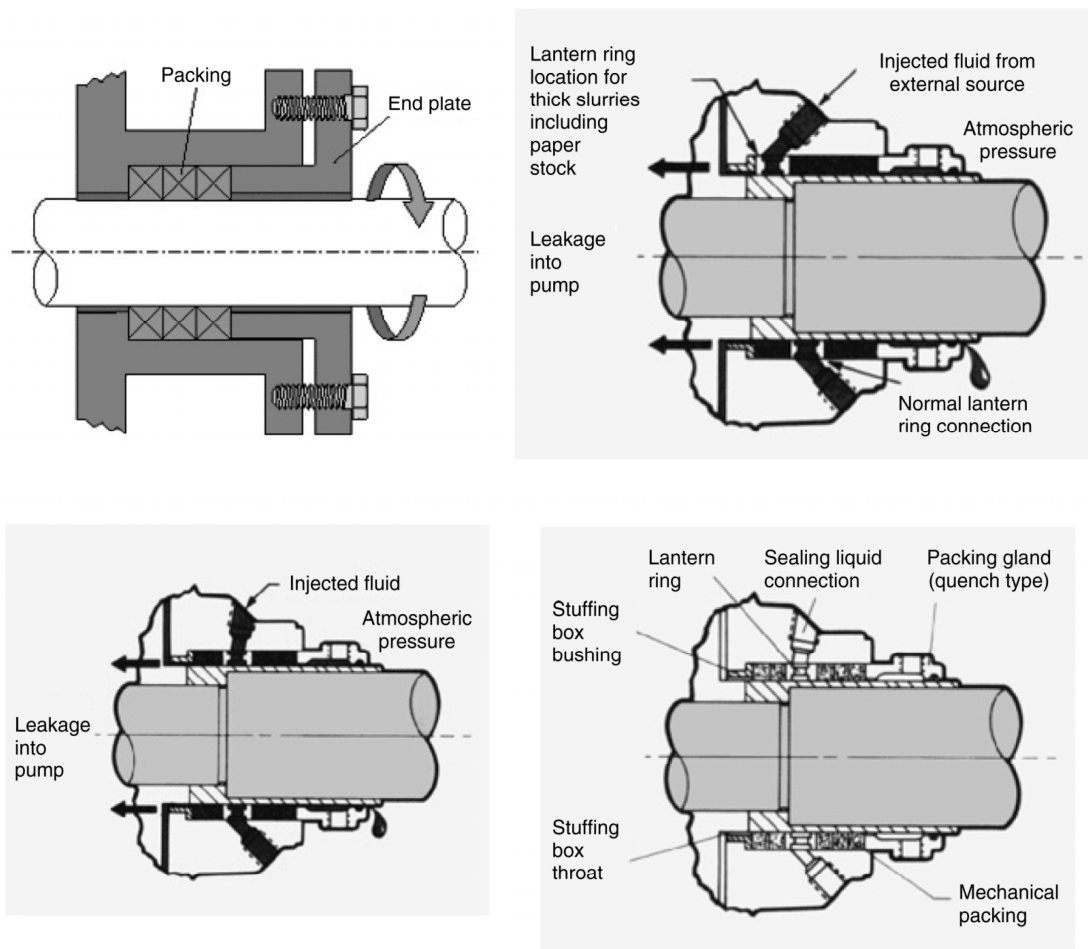


Figure 2.20 – Stuffing Box

PRACTICAL CENTRIFUGAL PUMPS

In a stuffing box, 4–6 suitable packing rings are placed and a gland (end plate) for squeezing and pressing them down the shaft.

The narrow passage, between the shaft and the packing housed in the stuffing box, provides a restrictive path to the liquid, which is at a high pressure within the pump casing.

The restrictive path causes a pressure drop, prevents leakage resulting in considerable friction between the shaft and the packing, and causes the former to heat up. It is thus good practice to tighten the gland just enough to allow for a minimal leak through the packing. This slight leakage of the liquid acts as a lubricant as well as a coolant. Obviously, this cannot be allowed for hazardous and toxic liquids, but then gland packings are also not used in such applications.

When pumps are handling dirty or high-pressure liquid, lantern rings are used. These are rings with holes drilled along its circumference.

A lantern ring substitutes one of the packing rings in the stuffing box and is situated at the pump end or midway between the packings.

In applications where the discharge pressure of the pump is higher, a restrictive bush is placed at the throat of the stuffing box.

When the liquid being handled is at a higher temperature (above 120°C), the stuffing box has an integrally cast water jacket housing. This allows for water circulation and keeps the packings at a lower temperature.

When toxic or corrosive liquids are handled, it is necessary to insure complete sealing of the stuffing box. Leakage of such liquids is a hazard to the plant personnel and can also be detrimental to the outer surface of the pump and foundation. It can also result in the loss of a valuable product.

PRACTICAL CENTRIFUGAL PUMPS

This is achieved by first reducing the pressure in front of the packings. The reduction in pressure is brought about by having radial blades at the back shroud of an auxiliary impeller (see Figure 2.21). This auxiliary impeller is also called as a repeller.

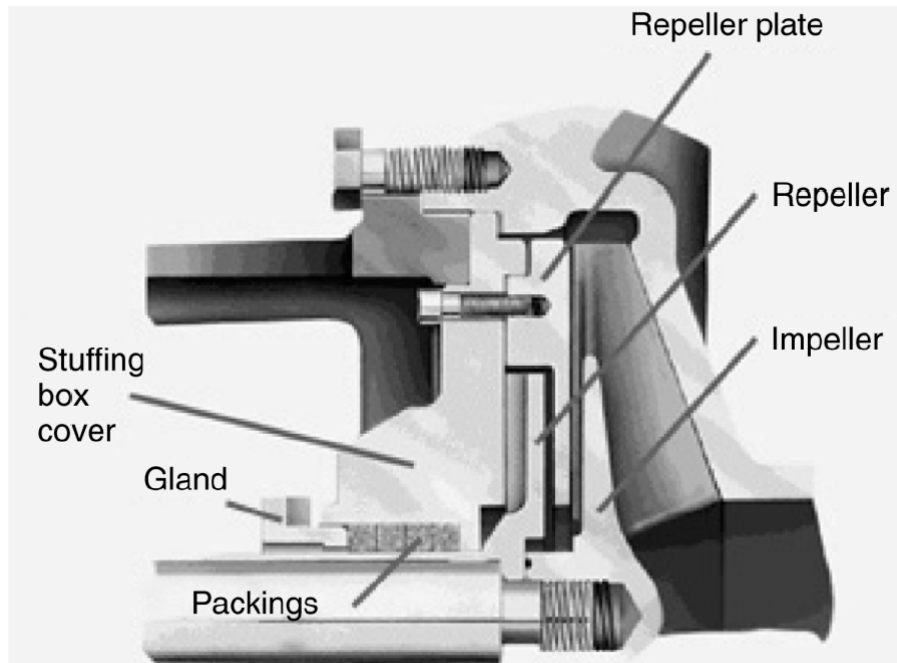


Figure 2.21 – Stuffing Box for a Corrosive Liquid

As the repeller rotates with a shaft, it throws the liquid outwards thus reducing the pressure in front of the packings. The pressure generated by the repeller is dependent on the length of the blades and its clearance with the casing.

Another common design is to cast back vanes of the main impeller itself. The back vanes help reduce the pressure acting on the packing.

The magnitude of the work done due to the friction between the shaft and the packings is influenced by:

- Kind of packing quality
- Length of the gland
- Diameter of the shaft
- Speed of rotation
- Pressure acting on gland
- Volume of liquid passing through the packing

In a properly operating stuffing box, the friction losses are usually of the order of 1% of the total pump power. This is independent of the size and kind of pump.

The present day packing used in pumps are predominantly made of PTFE (Teflon)/Graphite filaments. These are braided and formed into square shapes. They offer heat dissipation and low friction qualities. For example, see Figure 2.22 for a high-performance filament packing.



Figure 2.22 – A High-Performance Filament Packing
(EGK[®]** Filament Packing – Style 2070)

Packing properties can be enhanced by including other special materials during braiding of the packings, to handle contaminants, acids, alkalis, temperature, speed, and other factors. For example, a material called as Aramid can enhance the mechanical properties of the packing. This helps in prevention of extrusion of packing and withstands slurries and abrasives in liquids.

2.6.2 Mechanical Seals and Seal Housings

The stuffing boxes described above have many disadvantages and these include:

- A persistent leakage and loss of product if the shaft surface is not smooth.
- If the gland is too tightened, the shaft/sleeve gets hot and there can be rapid wear of the surface as shown in Figure 2.23.
- They require constant supervision.



Figure 2.23 – Wear on Shaft/Sleeve Due to Tight Packing

PRACTICAL CENTRIFUGAL PUMPS

As a result, the use of gland packing is being phased out but is still used in noncritical and low-power applications. In most of the applications, mechanical seals are used. Most of the disadvantages of packing are eliminated by the use of mechanical seals.

From its origins in 1930s, the technology of mechanical seals continues to evolve at a rapid pace. This is, especially, in regard to the enhancement of the reliability of seals.

Until 1950s, packing in the stuffing box was a standard method of shaft sealing. As operating conditions became more demanding and pumps were used on a greater variety of fluids, mechanical seals were designed to handle these changing conditions.

Mechanical seals comprise of two perfectly lapped mating faces. One face is stationary and the other is rotating. The leakage resistance in gland packing is along the axis of the shaft but in seals, it is orthogonal.

The seal faces cannot run mating with each other without any lubricant (Figure 2.24).

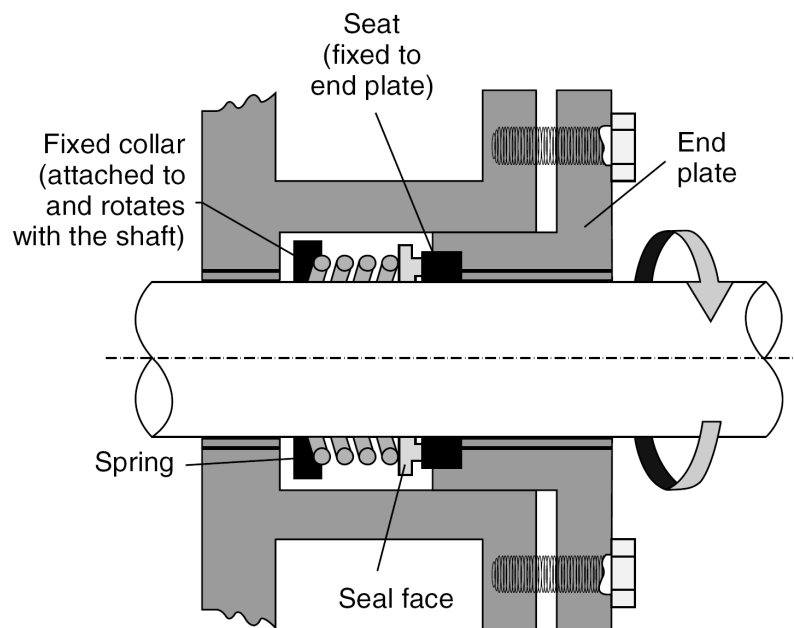


Figure 2.24 – Mechanical Seal

This can lead to an early wear and seal damage results in leakage. Usually, the sealant fluid is injected in the seal housing at a specified pressure, which lubricates and cools the faces.

PRACTICAL CENTRIFUGAL PUMPS

The fluid between the faces can escape into the atmosphere and this is called as fugitive emissions. In some applications, fugitive emissions are unacceptable and in such cases, multiple seal arrangements are used.

However, due to its precise design, mechanical seal demands careful attention to precision during pump assembly.

PRACTICAL CENTRIFUGAL PUMPS

There are three points of sealing as shown in Figure 2.25, common to all mechanical seal installations:

1. At the mating surfaces of the primary and mating rings
2. Between the rotating component and the shaft or sleeve
3. Between the stationary component and the gland plate

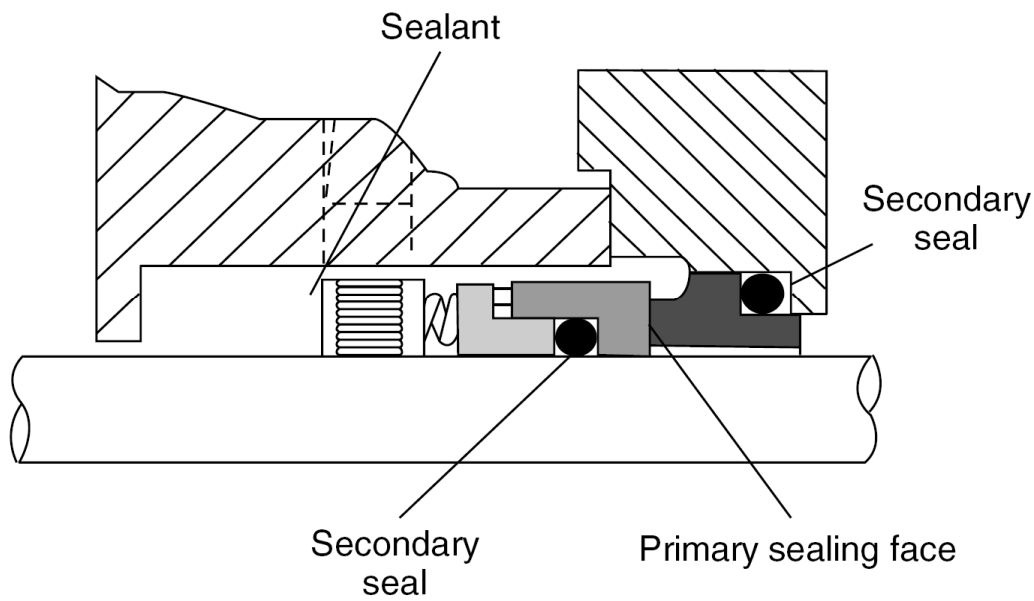


Figure 2.25 – Points of Sealing in a Mechanical Seal
(Pusher Type Unbalanced Seal)

When a seal is installed on a sleeve, there is an additional point of sealing between the shaft and sleeve. Certain mating ring designs may also require an additional seal between the gland plate and stuffing box.

Normally, the mating surfaces of the seal faces are made of dissimilar materials and held in contact with a spring. Preload from the spring pressure holds the primary and mating rings together during shutdown or when there is a lack of liquid pressure.

When a seal is installed on a sleeve, there is an additional point of sealing between the shaft and sleeve. Certain mating ring designs may also require an additional seal between the gland plate and stuffing box.

Normally, the mating surfaces of the seal faces are made of dissimilar materials and held in contact with a spring. Preload from the spring pressure holds the primary and mating rings together during shutdown or when there is a lack of liquid pressure.

PRACTICAL CENTRIFUGAL PUMPS

The secondary seal between the shaft and sleeve must be partially dynamic. As the seal faces wear, the primary ring must move slightly forward. Because of vibration from the machinery, shaft run out, and due to thermal expansion of the shaft to the pump casing, the secondary seal must move along the shaft. Flexibility in sealing is achieved from secondary seal forms such as an O-ring, wedge, V-ring, or bellows (see Figure 2.26 for bellows seal). Most seal designs fix the seal head to the sleeve or shaft and provide for a positive drive to the primary ring.

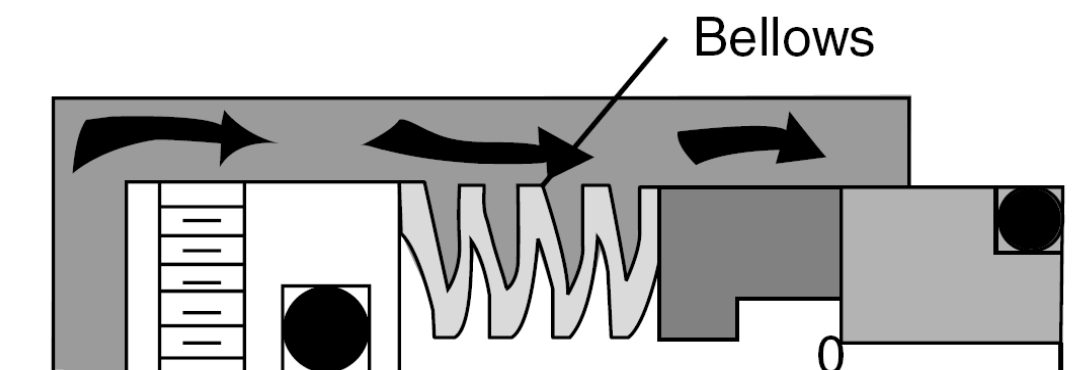


Figure 2.26 – Bellows Seal

Although mechanical seals may differ in various physical aspects, they are fundamentally the same in principle. The wide variation in design is the result of the many methods used to provide flexibility, ease of installation, and economy.

A seal arrangement is used to describe the design of a particular seal system and the number of seals used on a pump.

The most common sealing arrangements may be defined as:

- Single seal installations
- Internally mounted
- Externally mounted
- Dual seal installation
- Tandem seals
- Double seals
- Externally pressurized
- Internally pressurized

A single seal mounted inside the seal chamber represents at least 75% of all installations. It is the most economical sealing system available to the industry.

PRACTICAL CENTRIFUGAL PUMPS

Just as gland packings are housed in stuffing boxes, the mechanical seals are housed in seal housings. Research has indicated that providing an enlarged bore can provide distinctive advantages depending on the applications. Some of the seal housings suggested are given in Figure 2.27.

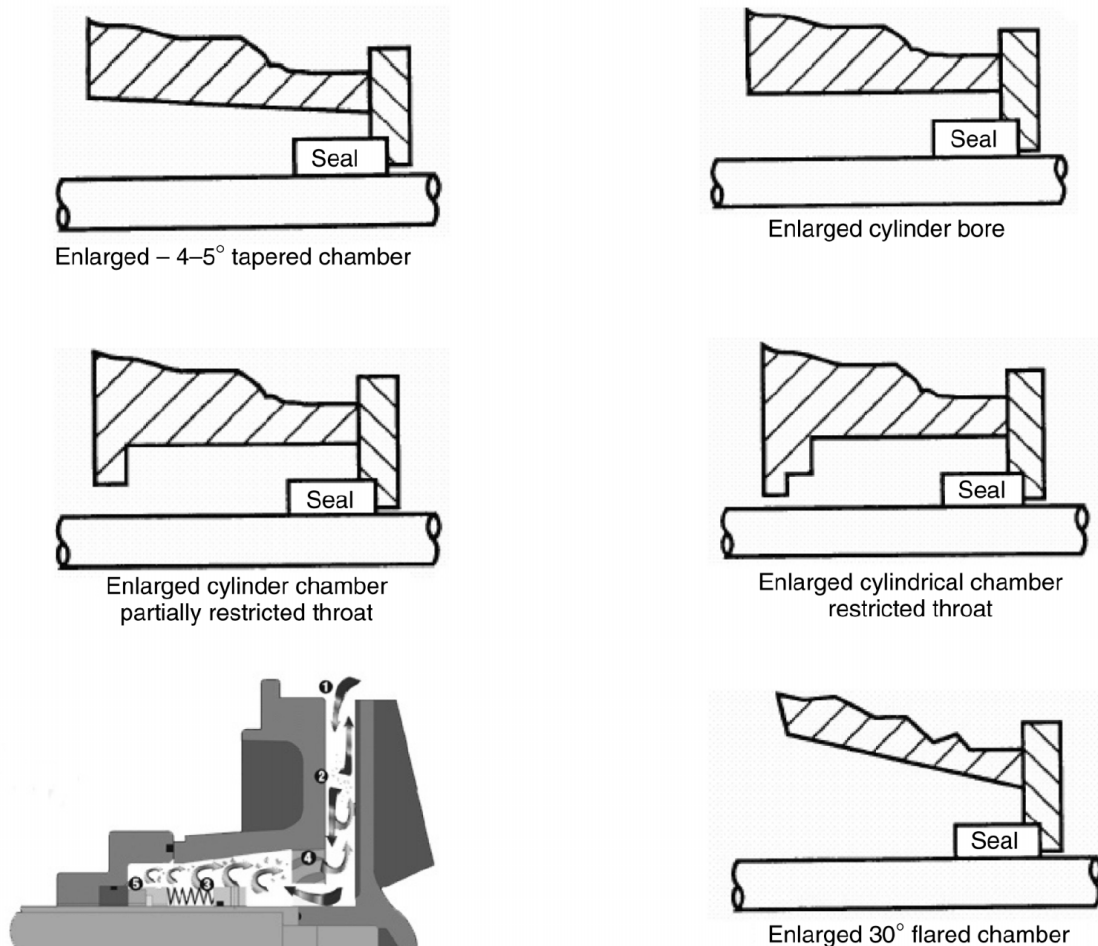


Figure 2.27 – Types of Seal Housings

Studies demonstrate that the fluid flow within the enlarged chamber is increased. This aids in the removal of seal generated heat.

Optimum selection of enlarged chambers can help deal with:

- Gases in seal housing under start stop conditions
- Light hydrocarbons with low boiling points
- Liquids with solid particles

These designs help in improving the reliability of the seals.

2.7 Bearing

2.7.1 Bearing Housing

The bearing housing encloses the bearings mounted on the shaft. The bearings keep the shaft or rotor in correct alignment with the stationary parts under the action of radial and transverse loads. The bearing house also includes an oil reservoir for lubrication, constant level oiler, jacket for cooling by circulating cooling water.

2.7.2 Types of Bearings

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.

PRACTICAL CENTRIFUGAL PUMPS

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.

2.7.2.1 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 2.28) is used frequently for large shaft 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.



Figure 2.28 – Self-Aligning Spherical Roller Bearing (SKF USA, Inc.)

2.7.2.2 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.

2.7.3 Bearing Arrangement

2.7.3.1 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.

PRACTICAL CENTRIFUGAL PUMPS

The most common ball bearings used in centrifugal pumps are;

1. Single-row, deepgroove,
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 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 2.29) 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 2.28) 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 2.30), 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.

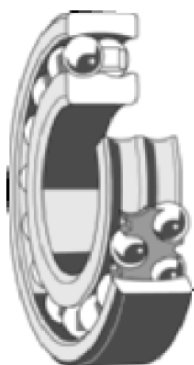


Figure 2.29 – Self-Aligning Ball Bearing (SKF USA, Inc.)



Figure 2.30 – Single-Row, Deep-Groove Ball Bearing (SKF USA, Inc.)

PRACTICAL CENTRIFUGAL PUMPS

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.

PRACTICAL CENTRIFUGAL PUMPS

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 2.31), typically with a 40° contact angle, and also as a double-row bearing (see Figure 2.32), most commonly with a 30° contact angle.



Figure 2.31 – Single-Row, Angular-Contact Bearing (SKF USA, Inc.)



Figure 2.32 – Double-Row, Angular-Contact Bearing (SKF USA, Inc.)

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 2.33). 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.

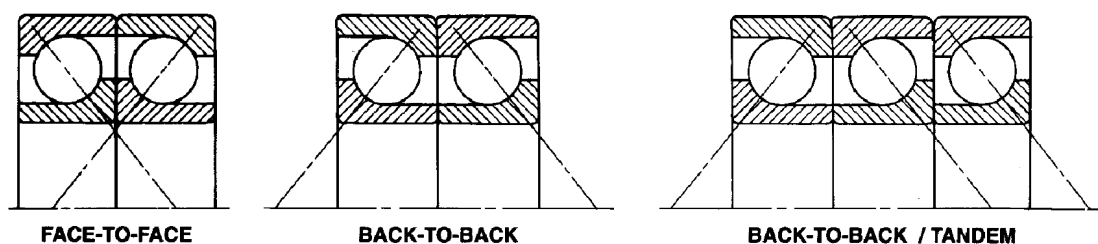


Figure 2.33 – Paired Bearing Arrangements (SKF USA, Inc.)

PRACTICAL CENTRIFUGAL PUMPS

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 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 2.34). 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.

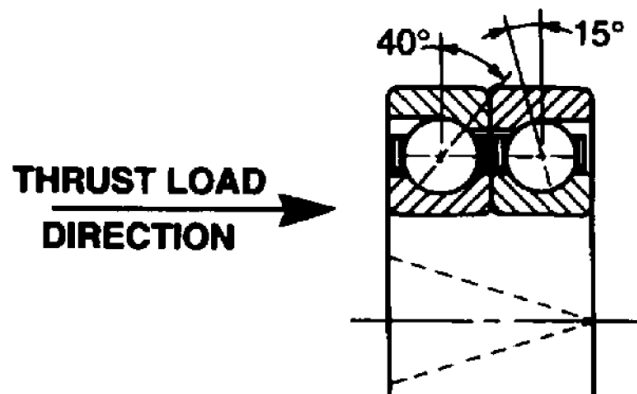


Figure 2.34 – Angular Contact Bearings with Different Contact Angles (SKA USA, Inc.)

PRACTICAL CENTRIFUGAL PUMPS

The double-row, angular contact ball bearing (see Figure 2.35) 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.

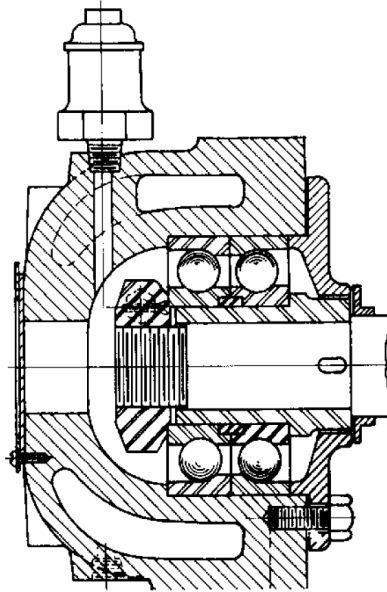


Figure 2.35 – A Double-Row, Angular-Contact Ball Thrust Bearing that is Grease-Lubricated and Water-Cooled

2.7.3.2 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.

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.

PRACTICAL CENTRIFUGAL PUMPS

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 it self very satisfactory. A vertically mounted thrust bearing arranged for grease lubrication **is shown in Figure 2.36.**

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 2.37).** In other designs, a constant-level oiler is used **(see Figure 2.38).**

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 2.39).**

Figure 2.36 – Figure 94

Figure 2.37 – Figure 95

Figure 2.38 – Figure 96

Figure 2.39 – Figure 97

2.7.4 Bearing Housing Protection Devices

There is a close relationship between the life of rolling element bearings and mechanical seals in pumps.

Liquid leakage from a mechanical seal may cause the bearings to fail, while a rolling element bearing in poor condition can reduce seal life. Only about 10% of rolling element bearings achieves their 3–5-year design life.

Rain, product leakage, debris, and wash-down water entering the bearing housing contaminate the bearing lubricant and have a catastrophic effect on bearing life.

A contamination level of only 0.002% water in the lubricating oil can reduce bearing life by as much as 48%. A level of 0.10% water will reduce bearing life by as much as 90%.

To improve the conditions inside a bearing housing, various types of end seals are used.

In almost every case, the normal operating life and quality of the end seal is not nearly as good as that of the rolling element bearings. Improving the quality of the end seals will increase the life of rolling element bearings.

PRACTICAL CENTRIFUGAL PUMPS

2.7.4.1 Felt and Lip Seals

One of the earliest, bearing housing isolators was the 'felt' (Figure 2.28). The bearing covers are provided with a groove in which a felt strip is cut and inserted. The felt acts as a barrier for oil and dust from the atmosphere.

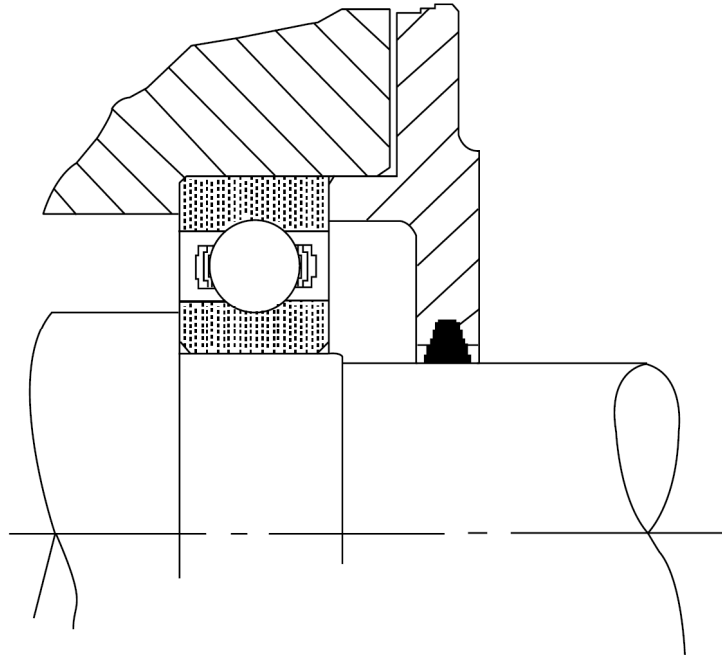


Figure 2.40 – Felt Seal

The lip or the oil seals (Figure 2.29) have low initial cost, availability, and are common.

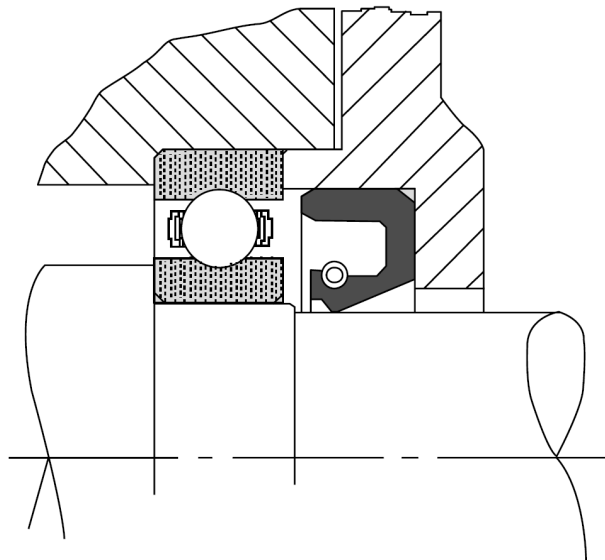


Figure 2.41 – Lip Seal

PRACTICAL CENTRIFUGAL PUMPS

New lip seals provide protection in both static and dynamic modes. Their major disadvantage is short protection life due to wear of the elastomer.

Life expectancy of a common single lip seal can be as low as 3000h, or 3–4 months. Thus, while a bearing is designed to last from 3 to 5 years of continuous operation, the lip seal will provide protection for only a few months.

The temperature limits of lip seals are -40 to 400°F (-42 to 203°C) for Viton.

2.7.4.2 Labyrinths

Labyrinths are devices that contain a tortuous path, making it difficult for contaminants to enter the bearing housing (Figure 2.30). Labyrinths are devices that contain a tortuous path, which in turn discourages and hence minimizes leakage of fluid without there being any physical contact between the Stationery and moving elements that make up the seal. Labyrinth seal design may vary hence selection must be based on its suitability for the application and purpose.

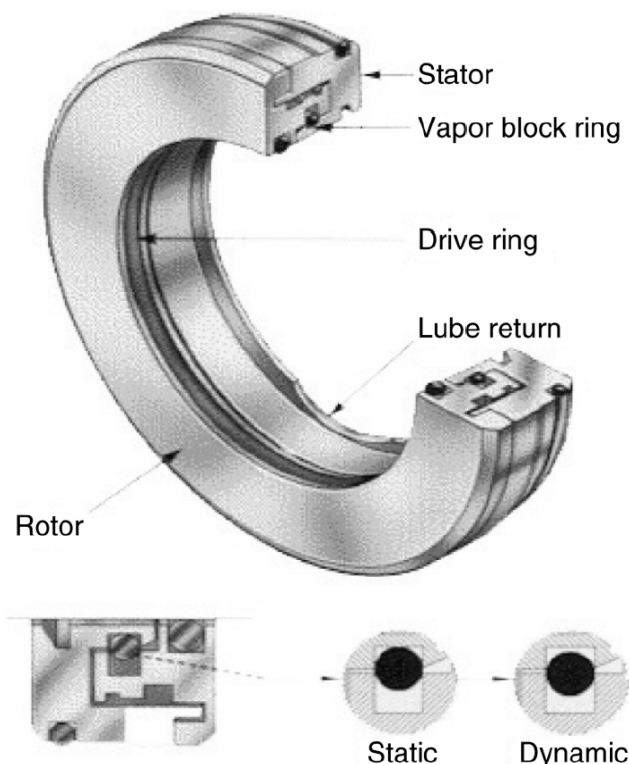


Figure 2.42 – A special Type of Labyrinth Seal (Inpro Seals)

PRACTICAL CENTRIFUGAL PUMPS

The advantages of labyrinths are their non-wearing and self-venting features. With no contacting parts to wear out, a labyrinth can be reused for a number of equipment rebuilds. Because the labyrinth provides an open, however difficult, path to the atmosphere, the bearing housing vent can be removed and the tapped hole can be plugged with a temperature gage.

The disadvantages of labyrinths include a higher initial cost than lip seals and the existence of an open path to the atmosphere, which can enable contamination of the lubricant by atmospheric condensate as the housing chamber 'breathes' during temperature fluctuations in humid environments. Also, they do not work as well in a static mode as in a dynamic, rotating mode.

The temperature limits of labyrinths are determined by the elastomer driving the rotor and holding the stator in place, the same as for the lip seal.

2.7.4.3 Magnetic Seals

Magnetic seals use a two-piece end face mechanical seal with optically flat seal faces held together by magnetic attraction (Figure 2.31).

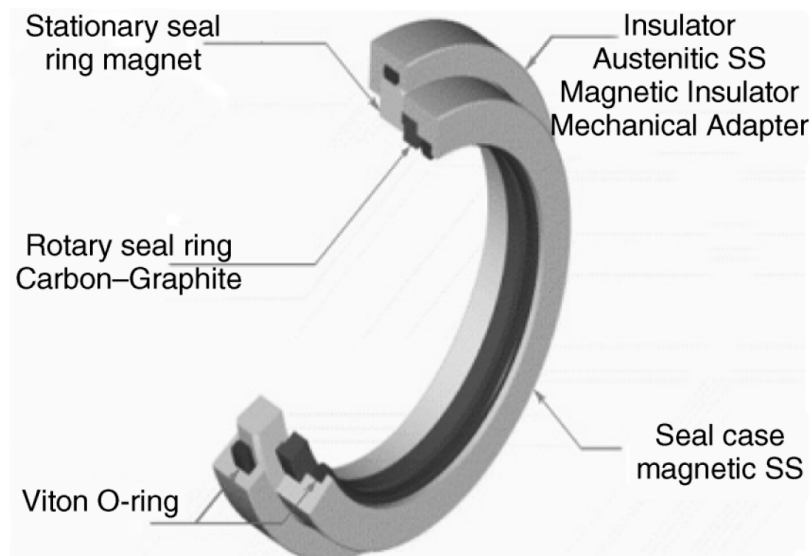


Figure 2.43 – Magnetic seal

They have a design life equivalent to mechanical seals and rolling element bearings and can be repaired.

The major advantage of magnetic seals is the hermetic seal they provide for the bearing housing. Because of the positive seal, other arrangements must be made to allow for the 'breathing' that results from expansion and contraction of the air pocket above the lubricant during normal temperature changes.

PRACTICAL CENTRIFUGAL PUMPS

Disadvantages of magnetic seals include higher initial cost and a shorter life than the almost infinite life of a labyrinth.

Magnetic seals are generally not recommended with dry sump, oil mist lubricated bearing housings, or grease-lubricated bearings. The upper operating temperature limit of magnetic seals is lower than that of labyrinth seals, in the range of 250°F (121°C).

2.8 Couplings

Couplings for pumps usually fall in the category of general-purpose couplings. Generalpurpose couplings are standardized and are less sophisticated in design. The cost of such coupling is also on the lower side. In addition, there are special purpose couplings that can be used on turbo machines and are covered by the API 671 specification.

In these couplings, the flexible element can be easily inspected and replaced. The alignment demands are not very stringent. The couplings fitted on pumps usually fall in any of the five types mentioned below.

These are:

1. Gear coupling
2. Grid coupling
3. Disk coupling
4. Elastomeric compression type
5. Elastomeric shear type

2.8.1 Gear Coupling

Gear couplings comprise of two hubs with external teeth that engage the internal teeth on a two or one piece flanged shroud or sleeve (Figure 2.32).

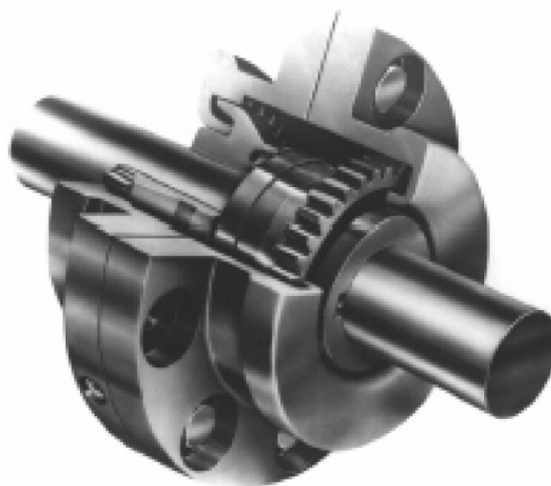


Figure 2.44 – Gear Coupling

One hub with shroud is mounted on the pump shaft and the other on the shaft of the prime mover. The flanges are bolted after packing grease between the meshing gears.

To contain the grease within an enclosed space, the flange faces and the shroud are provided with suitable static seals (like O-rings).

Some of the couplings may have a spacer between the flanged shrouds. In case of any misalignment between the shafts, sliding occurs between the external gears on the hub and its corresponding internal gears on the shroud.

Gear couplings are usually deployed on pumps above a rating of 75 kW.

2.8.2 Grid Coupling

The grid coupling in some ways is similar to the gear coupling. It also consists of two hubs mounted on the driver and driven shafts. The hubs are slotted and house a flexible grid member. Grease is applied to lubricate any sliding that may occur between the grid member and the slots of the hub.

A cover contains the lubrication. Grid couplings are usually not used in pumps with a power rating of 750 kW. A grid coupling is shown in Figure 2.33.

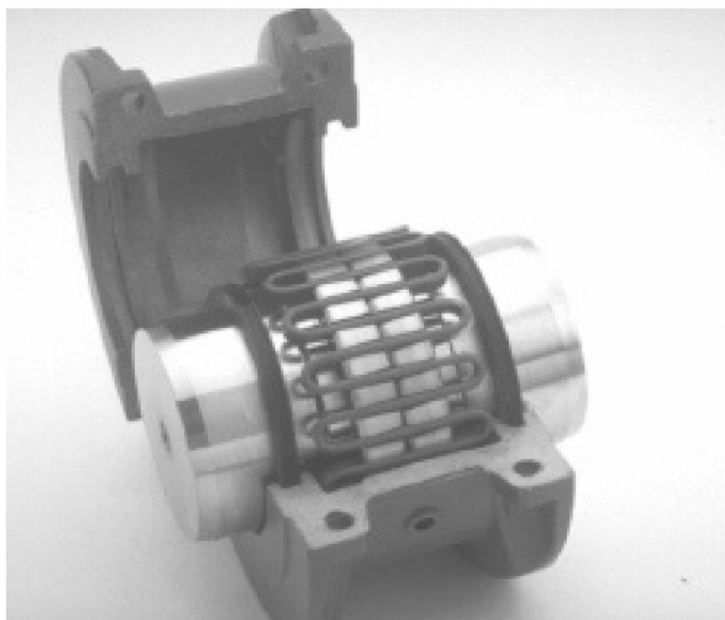


Figure 2.45 – Grid Coupling

2.8.3 Disk Coupling

A disk coupling shown in Figure 2.34 comes under the category of Metallic Element Coupling. Diaphragm couplings used on turbo machines also belong to this category.



Figure 2.46 – Metallic Disk Coupling

Metallic disk coupling comprises of two hubs mounted on the driver and driven shafts.

A set of flexible shims or metallic element is placed between the spacer and the hub.

The torque is transmitted by simple tensile force between alternate driving and driven bolts on a common bolt circle diameter.

Such couplings are used on pumps with a power rating of over 75 kW.

2.8.4 Elastomeric Shear Type Coupling

All elastomeric couplings are classified according to how their elastomeric elements transmit torque between driving and driven hubs. The elements could be either in compression or in shear.

In shear-type couplings, the driving and driven hubs operate in separate planes, while the driving hub pulls the driven hub through an elastomeric element suspended between them. Here, the element transmits and cushions the force between the hubs by being stretched between them (Figure 2.35).



Figure 2.47 – Elastomeric Shear Type Coupling

Among all couplings, this type can probably take the maximum amount of parallel misalignment.

These coupling are used in pumps below a rating of 75 kW.

2.8.5 Elastomeric Compression Type Coupling

In jaw couplings, the element (called as a spider) is loaded in compression between the jaws of mating hubs.

These jaws operate in the same plane, with the driving hub jaws pushing toward the driven hub jaws. Legs of the elastomeric spider transmit and cushion the force between the driving and driven jaws by being compressed between them.

Compression type couplings offer some advantages over the shear type of coupling (Figure 2.36). These include:

- Higher load capacity
- Greater torsional stiffness
- More safety
- Easier installation

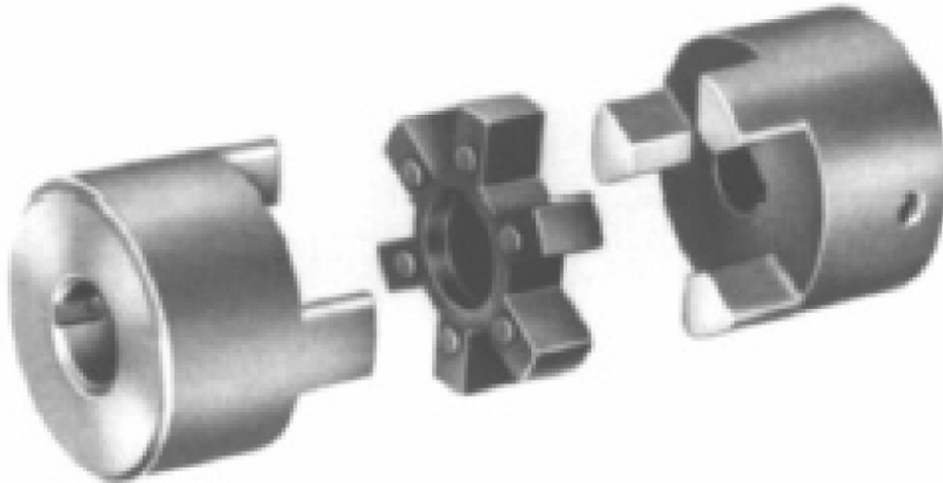


Figure 2.48 – Elastomeric Compression Type Coupling

The design of the coupling allows it transmit the torque even if the spider breaks. The driving jaws simply rotate until they contact the driven jaws directly, and the coupling continues to function, though it is accompanied by considerable noise and accelerated wear. In some cases, it can prevent an expensive downtime.

They typically accommodate angular shaft misalignment up to 1° and parallel misalignment up to 0.015 in.

The elastomeric material in the above two types of couplings is mostly NBR (Nitrile Butadiene Rubber), sometimes called Buna N. It is the most economical and widely used standard coupling element material. It resembles natural rubber in resilience and elasticity, and is resistant to oil, hydraulic fluid, and most chemicals.

The operating temperature ranges from -40 to $+100^{\circ}\text{C}$. With hardness of 80 Shore A, NBR provides the best damping capability among elastomeric elements.

Another material used is Urethane. It has 1.5 times the torque capacity of NBR with very good chemical and oil resistance. But has less damping capability (90 Shore A hardness) and narrower operating temperature range from -39 to $+71^{\circ}\text{C}$. Urethane spiders are good choices when the application calls for greater torque in a confined space, or for resistance to atmospheric effects such as ozone, sunlight, and hydrolysis in tropical conditions.

Hytrel (registered trademark of E.I. DuPont de Nemours & Co) is designed for high operating temperature range from -51 to $+121^{\circ}\text{C}$. It offers excellent resistance to oils and chemicals and can transmit 3 times the torque of standard NBR.

It also provides resistance to ozone, sunlight, and hydrolysis in tropical conditions.

With hardness of 55 Shore D, however, Hytrel cuts angular misalignment ratings in half, and damping capacity is low.

The spiders come in the following four types:

1. Standard solid center spider
2. Open center type
3. Snap wrap (with or without retainer ring): This flat-strip, open-end design connects the spider legs around the perimeter of the coupling rather than at the center. This allows for easy removal or installation without disturbing the alignment of either coupling hub. With no center connections, this design does not overlap into the bore, and therefore it allows shaft ends to extend at maximum bore diameter to a minimal distance between the shaft ends. This element is radially 'wrapped' around the jaws and needs to be held in place by either a ring or a collar. When retained by a ring, it has a maximum RPM limit of 1750. The collar configuration, on the other hand, achieves the same RPM rating as the standard coupling because the collar is attached to one hub.
4. Load cushions (separate blocks): As small, separate blocks, these cushions can be installed easily and removed radially, which can be very helpful for maintenance in heavy-duty applications. In certain models of coupling, load cushions must be held in place by a collar.