CHAPTER THREE

FORCES IN

CENTRIFUGAL PUMPS

Forces in Centrifugal Pumps

As we have seen in Chapter 3, a centrifugal pump can deliver a wide range of flows and heads of liquid column. The operating point on the curve; the internal pressures generated in the pump result in radial and axial forces on the rotor of the pump.

In this chapter, we shall discuss these forces in qualitative and quantitative terms and their impact on the design of the pumps.

4.1 Axial Thrust

The axial forces of thrust generated in a centrifugal pump results from the internal pressures acting on the exposed areas of the rotating element.

It may appear as simple as a product of the net of discharge and suction pressure and the exposed area of the impeller. Though this is truly the basis, there are many uncertainties that are not covered by this simple approach. The other variables that may affect the evaluation of axial loads are:

- Location of the impeller relative to the casing wall
- Impeller shroud symmetry
- Surface roughness of the walls
- Wearing ring clearance
- Geometry of the balancing holes.

Many of these variables are indeterminate and hence the axial forces calculated are at best approximate values.

In critical applications and during the testing of prototypes or to trouble-shoot repeated thrust bearing failures, it maybe essential to arrive at accurate thrust values. In such cases, thrust-measuring devices are installed. The internal pressures may also be recorded to get an idea of the hydraulic forces.

If the results are found unacceptable then design modifications maybe necessary.

The approximate method of axial thrust calculations is explained below. The following assumptions are made:

- Impeller profile is symmetrical.
- Impeller centerline coincides reasonably with the volute centerline (within 0.8 mm).
- Pressure on the front cover and back shroud is equal and constant from the impeller outside diameter to the wearing ring diameter.
- The pressure reduction, which is parabolic in nature, is ignored and average pressure acting on the shrouds is considered to be 3/4th of the pump discharge pressure.

4.1.1 Axial Thrust in Single-Stage Overhung Impeller Pumps

Axial thrust calculation in these simple pumps is made complex due to many variants in the design of the impellers and mating components. Axial thrust in some of the typical construction is discussed below.

4.1.1.1 Overhung Impeller Pumps: Closed Impeller with Front Wearing Ring

Consider perfect impeller symmetry (Figure 4.1), the axial forces in the region above the diameter D_1 is balanced. The pressure on either side is assumed to be the same.

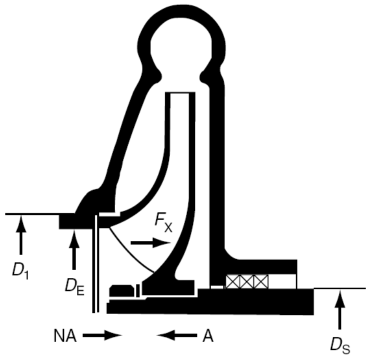

Figure 4.1 – Closed Impeller with Only Front Wearing Ring

The net axial thrust is due to the pressure difference acting on the area between the front wearing ring and the shaft diameter.

$$NA\,Thrust = A_S \times P_S + F_X$$

$$A\,Thrust = \frac{3}{4} \times P_D \times [A_1 - A_S]$$

Net Axial Thrust is the difference of the two.

F_X – This is the axial force due to change in momentum. It is computed using the equation,

$$\frac{[Density \times Q^2 (Flowrate)]}{A_E}$$

Q is m□/s, AE is in m□ and Density in kg/m□, and F_X is in newton.

Areas, A_1, A_S are obtained by the standard formula using their respective diameters.

$$Area = \frac{\pi}{4} \times D^2$$

4.1.1.2 Overhung Impeller Pumps: Closed Impeller with Wearing Rings on both Sides and Provided with Balancing Holes

Consider a closed impeller with wearing rings on both sides as shown in Figure 4.2:

D_1 and D_2	-	Wearing diameters on front cover and back shroud of impeller respectively
DE	-	Impeller eye diameter
DS	-	Pump shaft diameter

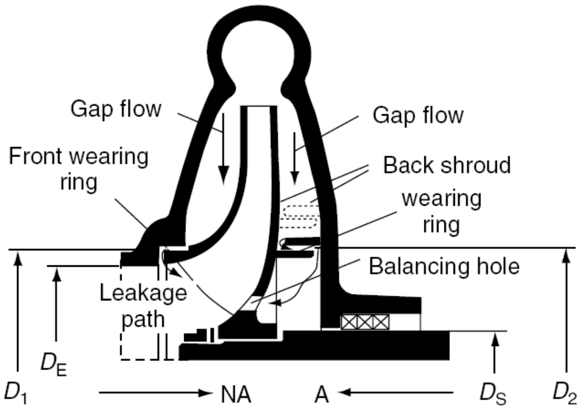

Figure 4.2 – Closed Impeller with Wearing Rings on both Sides and Provided with Balancing Holes

In this construction, the balancing holes drilled on the back shroud of the impeller play an important role in reducing the axial thrust. Due to these holes, the pressure in the region under the back wearing ring is reduced to suction pressure. However, some studies have indicated that the pressure in this region is 2–10% of the differential pressure depending on the diameter of the balancing holes.

The following equations are used to compute the axial thrust; the total area of balancing holes is considered to be eight times the area between the wearing rings.

When the wearing ring diameters, D_1 and D_2 are equal

$$Axial Thrust = A_S \times P_S - [0.03 \times p_D \times (A_1 - A_S)] + F_X$$

Where F_X is the thrust due to momentum change.

When the wearing ring diameters, D_1 and D_2 are not equal

$$NA Thrust = A_S \times P_S - [0.03 \times p_D \times (A_1 - A_S)] + F_X$$

$$A Thrust = \frac{3}{4} \times P_D \times (A_1 - A_2)$$

Net Axial Thrust is the difference between the two.

4.1.1.3 Overhung Impeller Pumps – Closed Impeller with Back Vanes

$$NA\,Thrust = A_S \times P_S + \left[0.00253 \times (A_R - A_S) \times (U_R^2 - U_S^2) \times \rho\right] + F_X$$

$$A\,Thrust = \frac{3}{4} \times P_D \times (A_1 - A_s)$$

Net Axial Thrust is the difference of the two.

U_R and U_S are the peripheral velocity and can be calculated by R ($\pi \times D_R \times N/60$) and ($\pi \times D_S \times N/60$) respectively.

$$\text{All areas - A} = \left[\frac{\pi}{4} \times D^2\right]$$

In many cases, it is not preferable to install a wearing ring on the back shroud. In such cases, back vanes are integrally cast on the back shroud of the impeller.

The clearance between the back vane (in Figure 4.3: 'a – h') and the casing is very important and is usually kept between 0.4 and 1 mm.

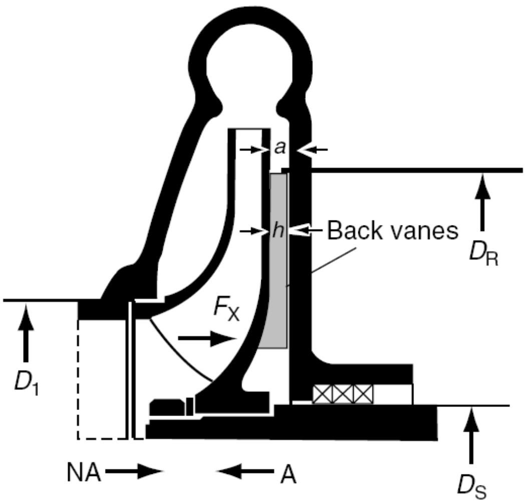

Figure 4.3 – Closed Impeller with Back Vanes and no Back Wearing Rings

As the back vanes rotate with the impeller, they themselves act like impellers expelling liquid from the inner to the outer diameter. This helps in reducing the pressure at the back of the impeller and the axial force.

However, this method is unreliable, as it is dependent on the clearance (a – h). If this gap increases, the pressure acting on the back shroud increases. This results in a higher axial thrust.

This design is less efficient than described earlier with back wearing rings. It is estimated that the back vanes can result in as much as 3% drop in efficiency of the pump.

4.1.1.4 Overhung Impeller Pumps – Open Impeller with front Wearing Ring

Consider perfect impeller symmetry, the axial forces in the region above the diameter D_1 is balanced. The pressure on either side is assumed to be the same.

The net axial thrust is due to the pressure difference acting on the area between the front wearing ring and the shaft diameter (Figure 4.4).

$$NA\,Thrust = A_S \times P_S + \left[(A_3 - A_E) \times \left(\frac{P_D}{2}\right)\right] + F_X$$

$$A\,Thrust = \frac{3}{4} \times P_D \times (A_3 - A_S)$$

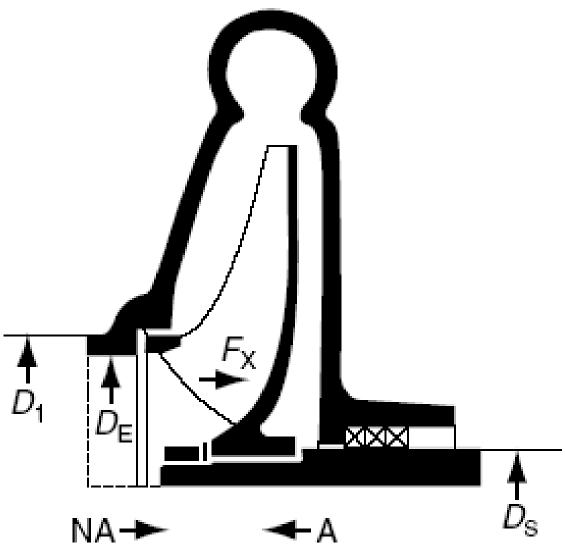

Figure 4.4 – Semi-Open Impeller with Front Wearing Ring

Net Axial Thrust is the difference of the two.

A semi-open impeller maybe designed with or without the back vanes based on the level of axial thrust generated in a pump.

Due to the absence of the front cover, the clearance between the impeller vanes and the pump casing is important. When back vanes are present, even the back clearance is important (Figure 4.5). This compounds the assembly problems.

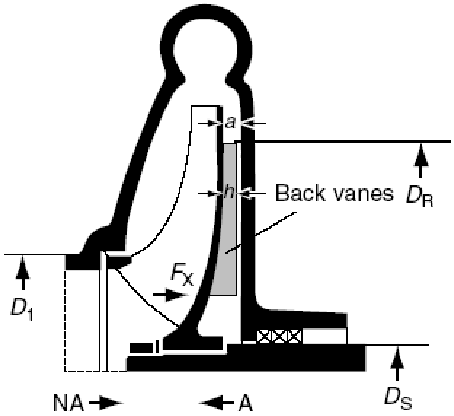

Figure 4.5 – Semi-Open Impeller with Back Vanes

$$NA\,Thrust = A_S \times P_S + \left[0.00253 \times (A_R - A_S) \times (U_R^2 - U_S^2) \times \rho\right] + \left[\frac{P_D}{2} \times (A_3 - A_E)\right] + F_X$$

$$A\,Thrust = \frac{3}{4} \times P_D \times (A_3 - A_S)$$

Net Axial Thrust is the difference of the two, where A3 is the area of the impeller

4.1.2 Double Suction Impeller

The double suction pumps equipped with double suction impellers, theoretically, develop little axial thrust due to the symmetry of their design.

However, the construction of such pumps can never be in perfect symmetry and therefore there is always some axial thrust generated which needs to be absorbed by a simple thrust bearing.

The symmetry of the design makes it necessary for the alignment of the impeller outlet and the volute center to be maintained. When this is violated, substantial thrust can be generated. This thrust is proportional to the axial displacement.

4.1.3 Multistage pumps – stacked impellers

In this design of pumps, all impellers are stacked in one direction. As a result, the axial thrust pushing the rotor toward the pump suction can be quite high (Figure 4.6).

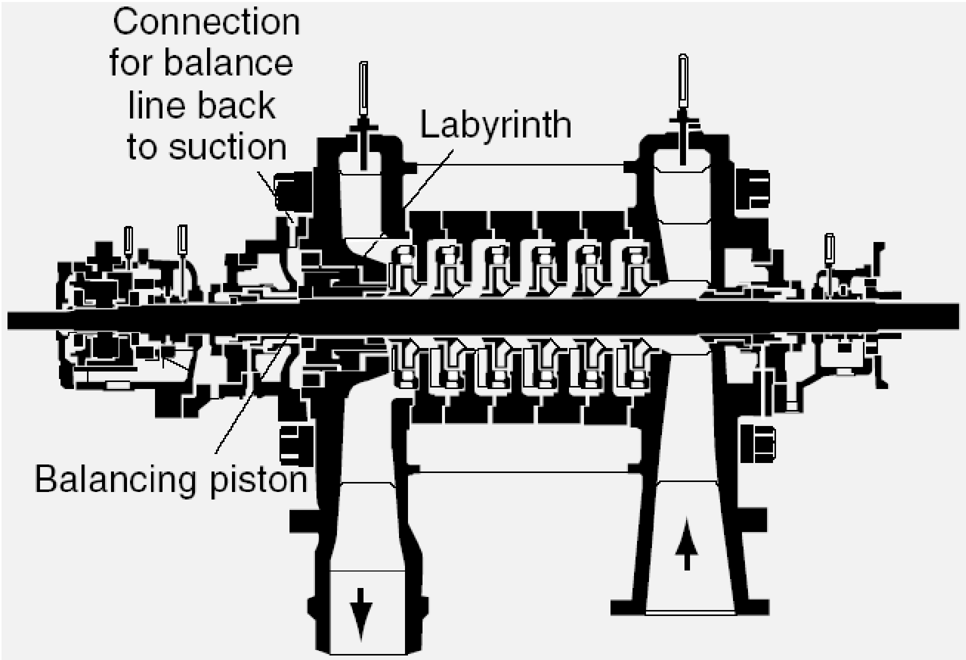

Figure 4.6 – Axial Thrust Balancing in a Multistage Stacked Impeller Design

In these designs, a balancing drum or piston is provided. This is a rotating circular disk, which runs against a stationary corresponding disk or ring with a close clearance. The pressure of the liquid is dropped across this labyrinth passage. This leak is connected back to the suction of the pump.

The pressure difference across the drum is same as the net of the discharge and suction pressures. The force generated due to the drum is in a direction opposite to that acting on the rotor. In this way, the axial thrust is reduced.

This arrangement also helps to reduce the pressure acting on the stuffing box.

However, there is a loss of efficiency due to recirculation of the liquid through the balancing line.

4.1.4 Multistage Pumps – Back-to-Back Impellers

In another arrangement in multistage pumps, impellers are arranged back-to-back. This simplifies the axial balance for an odd or even number of stages. In this design, there are two locations on the rotor that break down half total pump head. One is the center bushing between the centers back-to-back impellers and the other is the throat bush in the high-pressure stuffing box. Sleeve diameters at these locations can be sized to provide axial balance for any number of stages.

It is preferable to have some residual axial thrust to keep the bearing lightly loaded and thus keep the rotor in one place rather than to allow it to float.

An increase in wearing ring or bush clearances does not affect axial thrust. It is recommended that all impeller rings should be of the same diameter and on even stage pumps, the center sleeve diameter should be the same as the sleeve diameters in each stuffing box.

To balance an odd number of stages, the simplest solution is only to change the diameter of the center sleeve.

4.2 Radial Loads

In an end-suction centrifugal pump with an overhung impeller, the hydraulic radial load is due to the non-uniform velocity of the liquid within the pump casing or a volute. This unequal liquid velocity leads to a non-uniform pressure distribution of the pressure acting on the circumference of the pump impeller.

The radial load is most influenced by the design of the volute. The volute is designed to channel the liquid from the impeller to the discharge piping. The design of the volute is based on the flow at BEP. At this flow rate, the distribution of velocity and pressure of the liquid in the pump casing is uniform and there is negligible radial thrust on the impeller.

The minimal radial load at BEP is due to the cutwater of the volute. The cutwater causes a non-uniform pressure distribution on the circumference of the impeller that result in a net radial load on the impeller. This radial force is directed toward the cutwater.

At flows greater and less than the flow at BEP, the magnitude of the radial force increases. The direction of the radial force also changes as shown in Figure 4.7.

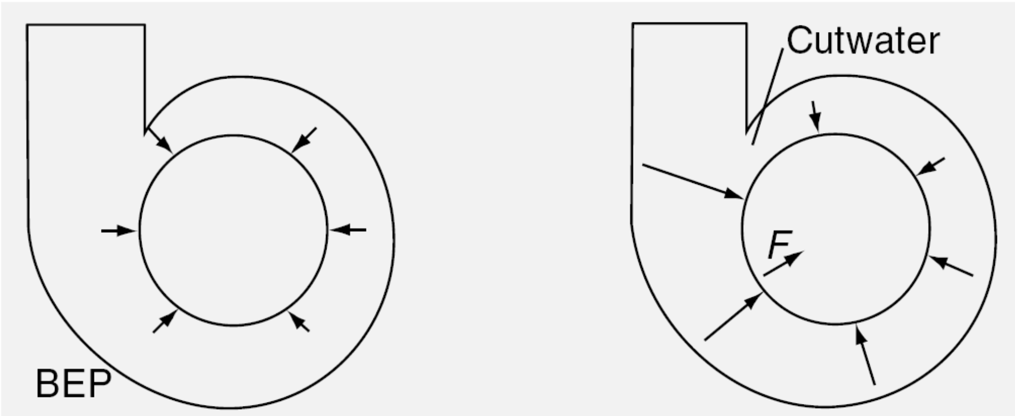

Figure 4.7 – Radial Forces in a Single Volute for Flows at BEP and Greater than BEP

Double volutes are observed in large pumps. An additional rib is cast in the volute and this construction provides for two cutwaters. This can be seen in Figure 4.8.

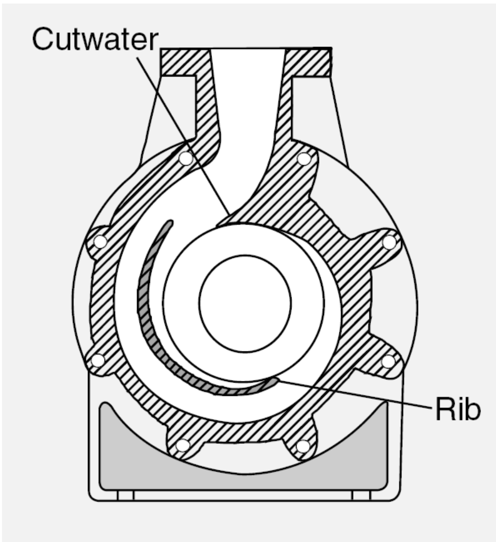

Figure 4.8 – Double volute pump

The symmetry of the volute around the impeller due to rib results in an equal distribution of the forces and pressures acting on the impeller and thus the radial loads in this construction are reduced to a large extent.

The radial forces in such a construction over the range of operation is shown in Figure 4.9.

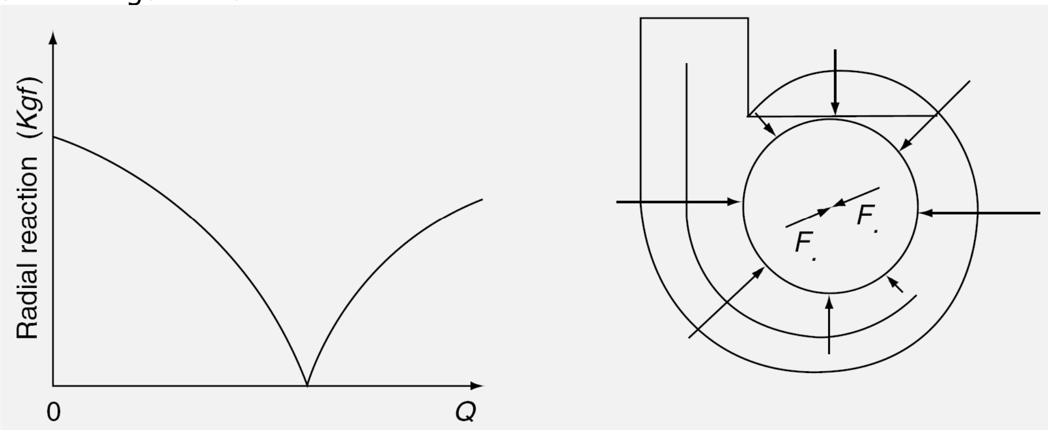

Figure 4.9 – Radial Forces Over the
Flow Range of the Pump with a Double Volute

In concentric casing pump (covered under Section 2.2), uniform velocity and pressure distribution occur only at zero flow. The uniformity is progressively lost as the flow rate increases. At BEP, the non-uniformity is much higher than at no-flow situation.

This is quite the opposite of what happens in a conventional volute casing.

In a vaned diffuser pump, covered in Section 2.2, the velocity distribution around the impeller is more uniform and therefore has a much lower radial loads on the impeller.

For an end-suction single volute type of pump, the radial forces at shut-off conditions (the pump is running and the discharge valve is shut or almost shut) can be calculated using the following formula:

$$R_{SO} = K_{SO} \times \left[\frac{(H_{SO} \times \rho)}{2.31} \right] \times D_2 \times B_2$$

At other points of operation, the following formula can be used to obtain an approximate value of the radial load on the impeller.

$$R = \left(\frac{K}{K_{SO}}\right) \times \left(\frac{H}{H_{SO}}\right) \times R_{SO}$$

Where:

K = $K_{SO} \times [Q / Q_{BEP}]$
R_{so} = radial thrust (in pounds) at shut-off
R = radial thrust (in pounds) at operating conditions
K_{SO} = thrust factor at shut-off (obtained from Figure 4.10)
K = thrust factor at operating conditions (see above formula)
H_{SO} = total head at shut-off (in feet)
H = total head at operating conditions (in feet)
ρ = specific gravity of the liquid
D_2 = impeller diameter (in inches)
B_2 = impeller width at the discharge including shrouds (in inches)
Q = capacity at operating conditions (in gpm)
Q_{BEP} = capacity at the best efficiency point (in gpm).
X = An exponent varying between 0.7 and 3.3 established by testing. It is safe to assume a linear relationship between 0.7 at a specific speed of 500 and 3.3 at a specific speed of 3500.

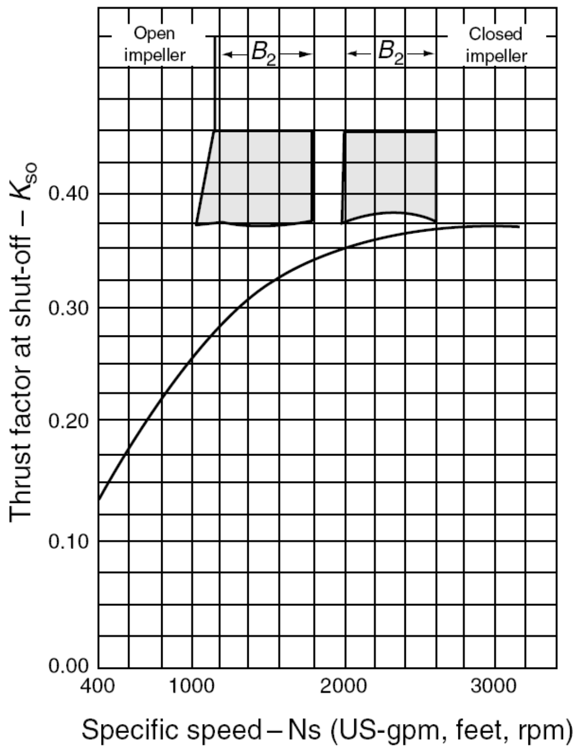

Figure 4.10 – K_{so} – Thrust Factor at Shut-Off – (First Formula)

Another approximate variation of the formula available for calculating the radial load is as given below:

$$R = K \times H \times D_2 \times \frac{B_2}{2.31}$$

Where:

K = $0.36 \times [1 - (Q/Q_{BEP})^2]$

R = radial thrust (in pounds) at operating conditions

D_2 = impeller diameter (in inches)

B_2 = impeller width at the discharge including shrouds (in inches)

H = total head at operating conditions (in feet)

Q = capacity at operating conditions (in gpm)

Q_{BEP} = capacity at the best efficiency point (in gpm).

In addition to the hydraulic radial loads generated mainly due to the pump construction and the operating point, there are also other factors that exert radial loads on the impeller.

A fluctuating radial load is sometimes due to the reduced clearance between the impeller vanes and the cutwater of the volute. The resulting vibration has a frequency, which is the product of the number of vanes on the impeller and the pump speed. This is called as the vane pass frequency.

The mechanical imbalance of the pump rotor also generates a radial force on the pump shaft. Uneven flow through the impeller can also cause imbalance of the impeller.

4.2.1 Shaft Deflection Due to Radial Loads

A radial load such as that from hydraulic origin or due to mechanical imbalance causes the pump shaft to bend downward when it is in one position. When the shaft is rotated by 180°, it still bends downward in a similar way. This bending of shaft due to a constant load in one direction is called as deflection (Figure 4.11).

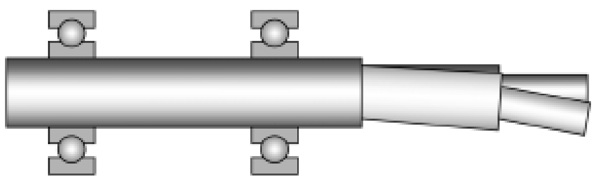

Figure 4.11 – Shaft Deflection

The shaft that is deflected rotates on its own centerline even though the centerline may not be straight.

This produces a reversal of stresses in the shaft in each revolution that could lead to fatigue cracking and an eventual breakage of shaft.

In addition to the above, there are several reasons why it is important to limit the distance that a shaft can deflect. The most important reason is that some of the rotating parts maybe exposed to the stationary parts. Stainless steel parts will gall as soon as rubbing contact is made that results in motor overload. Galling also tends to become progressively worse and may result in a complete seizure of the rotating elements.

Usually the surfaces which come into contact are the pump wearing rings or the throat bushing fitted to the seal housing and the shaft. The clearance between the wearing rings must be kept to a minimum in order to have a high-pressure drop across them. These surfaces are therefore the first to make contact. On pumps without the wearing rings, the first point of contact is usually of the shaft with the throat bush of the stuffing box.

The second effect of excessive deflection occurs in the stuffing box. Conventional pump packing when adjusted for the deflection under one operating condition will not readjust itself when the deflection is changed. For example, if the packing is set for operation at the high-capacity end of the curve and the discharge valve is throttled, the shaft would move from one extreme position to another, leaving a gap between the packing and the shaft. This gap coupled with the higher stuffing box pressure present under close to shut-off conditions would lead to excessive leakage.

Similarly, in pumps installed with mechanical seals, a shaft deflection can cause facial misalignment of the mechanical faces. This leads to an opening of the seal faces or an uneven mating of the seal faces, which then leads to an uneven and early wear of the seal faces leading to leakage.

There is often confusion between shaft deflection and another motion of the shaft termed 'shaft whip'. In a shaft whip, the shaft end rotates in a manner to generate a cone (Figure 4.12).

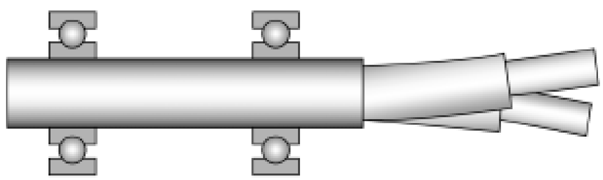

Figure 4.12 – Shaft whip

In this case, the shaft centerline changes 180° from every 180° turn of the shaft. Shaft whip occurs due to rotor dynamic problems and the radial loads have little contribution to this phenomenon.

4.2.2 Calculating shaft deflection

Shaft deflection is calculated by treating the shaft as a cantilever beam using an expanded version of the beam formula (Figure 4.13).

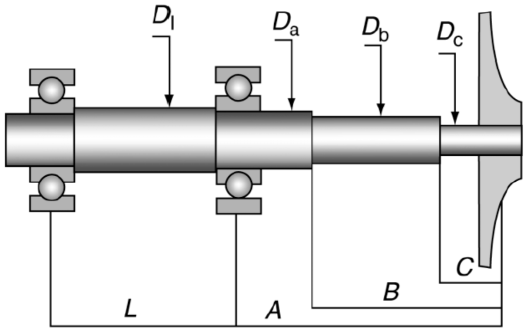

Figure 4.13 – Calculating shaft deflection

The formula for the deflection of a shaft with three major diameters in the overhung section is as given below.

$$\delta = \frac{P}{3E}\left[\frac{C^3}{I_C} + \frac{B^3 - C^3}{I_B} + \frac{A^3 - B^3}{I_A} + \frac{A^2 \times L}{I_L}\right]$$

Where:

δ	=	shaft deflection
P	=	radial resultant force
E	=	Young's modulus – modulus of elasticity of the shaft material
A,B,C,L	=	distances from impeller as shown in the figure above
I_A, I_B, I_C, I_L	=	Moments of Inertia at various diameters

Consistent units have to be used.

The above formula can be reduced to state that shaft deflection δ is proportional to L□/D□, where L is the length of the shaft and D is the diameter of the shaft.

Pumps with lower L□/D□ would exhibit higher shaft deflections than pump shafts with higher L□/D□ ratio.

The shaft deflection at the mechanical seal faces should not exceed 0.05mm. The reliability of the seal is affected in case the deflections are higher.

However, it is recommended that the L□/D□ ratio should not be used as a yardstick to compare the reliability of the pump vis-à-vis another pump. Instead, the actual shaft deflection values as obtained by the OEM for the most severe flow conditions should be used as a basis for comparison.

Thus, we have seen that the hydraulic loads are dependent on the type and size of the impeller, casing and the operating point of the pump, the suction, and the discharge pressures. The magnitude and direction vary greatly with the change in any of the above mentioned factors.

The variation of radial forces as a function of the operating point is shown in Figure 4.14.

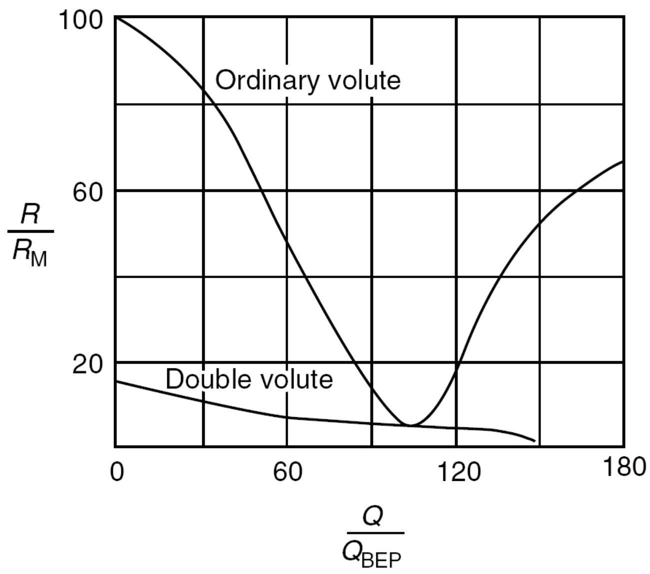

Figure 4.14 – Ordinary Volute vs Double Volute

The comparison is between the radial forces generated in an ordinary volute and in a double volute.

These curves are for particular specific speed pumps. Changes in specific speed can modify the shape of these curves.

The X-axis is the percentage ratio of the flow rate and flow rate at BEP. The Y-axis is the percentage ratio of the radial force to the maximum radial force at shut-off conditions.

In addition to the radial forces due to hydraulic factors, cavitation, misaligned belts, and couplings also contribute to the radial loads on the pump shaft.

A complete shaft analysis is done considering the axial as well as the radial forces. These are then used to compute the reaction forces at the bearing locations.

Once having known the forces that would be acting on the bearings, a proper selection of bearings can be made to obtain maximum availability from the pump (Figure 4.15).

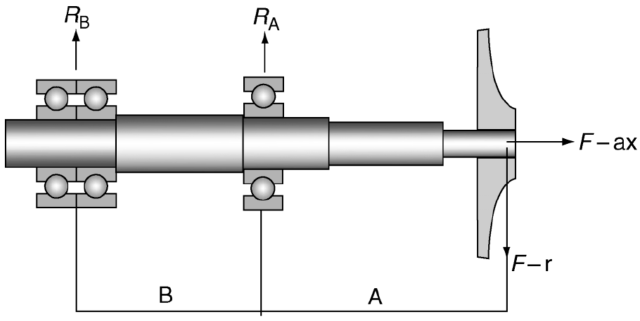

Figure 4.15 – The forces and reactions on a pump shaft

The method to evaluate the bearing reactions is as shown below:

$$R_A = \frac{F_r \times (A+B)}{B}$$

$$R_B = F_r + R_A$$

At location A, there is only a radial bearing, thus it will withstand the force:

$$P_A = R_A$$

At location B, there are double row angular contact bearings, which are meant to take the radial as well as the axial loads. Thus, they will take the combined load given by:

$$\mathrm{P_B = XR_B + YF_{\alpha x}}$$