CHAPTER SIX

CENTRIFUGAL

PUMP

MAINTENANCE

Centrifugal Pump Maintenance

7.1 Introduction

The basis of centrifugal pump maintenance is a direct function of its criticality to its application. For example, an ordinary garden water pump would not require the same kind of attention as a boiler feed water pump in a major power plant or a firewater pump in a refinery.

The criticality of any pump equipment is based on the following criteria:

- Failure can affect plant safety
- Essential for plant operation and where a shutdown will curtail the process throughput
- No standby or installed spares
- Large horsepower pumps
- High capital cost and expensive to repair or longer repair lead time
- Perennial 'bad actors' or pumps that wreck on the slightest provocation of an off-duty operation
- Finally, pump trains, where better operation could save energy or improve yields are also likely candidates.

Once the criticality of a pump can be ascertained based on the factors mentioned, the pumps can be classified as:

- Critical
- Essential
- General purpose

After this categorization, the type of maintenance philosophy can be assigned. The pumps, which fall in the category of critical machines, are usually maintained with the predictive and proactive techniques.

The essential category pumps are assigned with preventive maintenance whereas maintenance for the general-purpose pumps maybe less stringent.

In actual operations, a mix and match of techniques is applied with a prime intention of maximizing runtime lengths and reducing downtime and costs.

The present day focus on continuous process plant pumps is to adopt a mix of Predictive and Preventative Maintenance (PPM).

There are four areas that should be incorporated in a PPM program. Individually, each one will provide information that gives an indication of the condition of the pump; collectively, they will provide a complete picture as to the actual condition of the pump.

These include:

- Performance monitoring
- Vibration monitoring
- Oil and particle analysis
- System analysis

7.1.1 Performance Monitoring

The following six parameters should be monitored to understand how a pump is performing:

1. Suction pressure (P_s)
2. Discharge pressure (P_d)
3. Flow (Q)
4. Pump speed (N)
5. Pump efficiency (η)
6. Power

When the driver is an electrical motor, the power is easiest to measure using portable instruments. These can be clipped on the electrical phase cables and used to measure the ampere, voltage, power factor, and eventually the power.

In conjunction with the process parameters such as suction and discharge pressures, flow rate and fluid characteristics, the efficiency of the pump can be computed and benchmarked against expected values provided by the OEM.

The suction and discharge pressures are measured using calibrated pressure gages screwed on to the provided fittings on pipelines with isolation valves.

The flow measurement is usually done with obtrusive instruments such as venturi tubes, orifice plates, and pitot tubes. When such devices are not installed, the non-obtrusive types like the doppler or transit time devices can be used. These can be portable but caution must be exercised because each device must be calibrated, and independent testing has shown these devices are sensitive to the liquid being pumped and are not precisely accurate.

A detailed method to compute the pump performance is given in Section 3.14.

Analysis of efficiency or inefficiency can help one to determine whether the losses are on account of:

- Hydraulic losses
- Internal recirculation
- Mechanical losses

Direct reading thermodynamic pump efficiency monitors (such as the Yates meter) are now available and capable of interpreting the pump's operating efficiency in a dynamic manner by measuring and computing the rise in temperature (albeit in mK) of the fluid as it moves from the suction to the discharge side of the pump. Highly sensitive and accurate temperature probes and pressure transducers are positioned on the suction and discharge pipe work to measure temperature and pressure, whilst a power meter provides information on the power absorbed by the electric motor.

Motor efficiency obtained from the manufacturer of the electric motor and drive losses are factored in to the software program, to then calculate the operating efficiency of the pump. This reading could be compared with the pump's commissioning data and drop in performance or efficiency could be determined.

The advantage of this set up is that:

- No dismantling of the pump is necessary.
- Offers cost savings and energy savings by increasing the pump availability and reliability factors.
- The time to maintain the pump set maybe predicted and planned more accurately and in a qualified manner in line with predictive and planned maintenance strategies.
- If a flow meter is installed to measure process liquid flow, then the pump monitor is able to verify the accuracy of the meter readings by calculating 'Q' from the empirical formula for power 'P'.

Though the capital cost for such meters maybe high, the return on investment may soon be realized for large capacity pumping systems involving numerous pumps. Alternatively, the service could be purchased at an affordable cost, or services provides could be contracted to periodically measure and report on the performance of the pumps as part of the PPM program.

7.1.2 Vibration Monitoring

Vibration monitoring on pumps can be online or offline based on the criticality of its pump. Online vibration measurements would use proximity probes to measure shaft vibrations. The most common type of vibration monitoring is offline using portable data collects and measuring casing vibrations.

The vibrations are measured for both the pump and its driver. Vibration measurements are collected at bearing locations of these machines. While reporting, it is necessary to follow a convention for labeling the various bearings of a machine train from where measurements are made. The general convention followed is to start labeling from where the power comes in.

So, for a simple machine train consisting of a motor and pump will be labeled in the following way (Figure 7.1):

- Motor non-drive end bearing – A
- Motor drive end bearing – B
- Pump outboard bearing (next to the coupling) – C
- Pump inboard bearing (away from coupling) – D

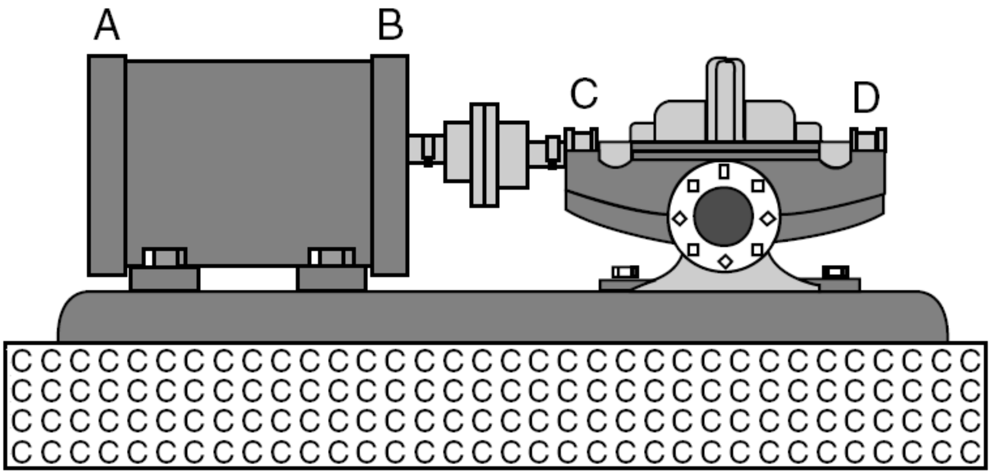


Figure 7.1 – Bearing Naming Nomenclature

Once having labeled the bearings, it is important that vibrations are taken in three Cartesian directions. In vibration nomenclature, these are vertical, horizontal and axial directions. This is because of the construction of machines; their defects can show up in any of the three planes and hence should be measured.

Vibration reporting is generally done in the manner shown below.

Velocity (mm/s-pk) 20/8/02	**Vertical**	**Horizontal**	**Axial**
A	2.4	1.7	1.0
B	2.1	1.9	1.2
C	4.3	5.6	2.7
D	3.7	4.1	2.1

Table 7.1 –

Hand held measurements are subject to a number of errors. It is thus important that personnel carrying out this task is aware of the possible errors that can occur while taking measurements.

The errors can occur due to:

- Position on machinery
- Probe angle
- Probe type
- Pressure

A base reading of vibrations is taken at various operating points. The vibration levels are minimum at BEP and rise when the operating point is away from the BEP. A comparison with the original base readings can indicate the deterioration in the mechanical health of the pump.

In absence of the base reading, the limits provided in API 610 can be used as a reference.

Frequency analysis of vibrations helps to determine the nature of the pump fault.

The three most common types of problems associated with pumps are:

1. Unbalance
2. Misalignment
3. Bearing defects

These can be easily detected using vibration analysis. Some of the infrequently occurring problems of the hydraulic nature like cavitation, improper impeller to volute tongue gaps, can be detected using vibration analysis.

Infrared thermography measurements may also be utilized in conjunction with vibration analysis to pinpoint the source of a problem due to the pick up of an elevated temperature at the trouble spot.

7.1.3 Oil and Particle Monitoring

The testing of the oil and possible particles in it can be a good indicator of the condition of the pump bearings and the oil program of the company.

There are several tests that can be performed on the lubricant to determine the condition of the bearing or determine why a bearing failed so appropriate corrective action can be taken.

These tests include:

- Spectrographic analysis
- Viscosity analysis
- Infrared analysis
- Total acid number
- Wear particle analysis and wear particle count
- Moisture content

Most of these tests have to be performed under laboratory conditions. Portable instruments are now available that enable the user to perform the test on site.

7.1.4 System analysis

A system analysis is a method of determining the system resistance curve of the pump as depicted in Section 3.6. This can be further used to evaluate the NPSH-a for the pump as in the installed condition.

A typical system analysis will include the following information, NPSH-a, NPSH-r, static head, friction loss through the system, and a complete review of the piping configuration and valves and fittings.

System analysis is often overlooked because it is assumed the system was constructed and operation of the pumps is in accordance with the design specifications.

This is often not the case. As has been stated previously, it is imperative to know where the pumps are being operated to perform a correct analysis and this is dependent on the system.

In addition, the process must be understood because it ultimately dictates how the pumps are being operated. All indicators may show the pump in distress when the real problem could be that it is run at low or high flows due to excessive hydraulic forces inside the pump.

By now, we see that pump maintenance is not all about dismantling and assembling pumps. Pump maintenance is an activity to prolong the efficient runtimes of the equipment to improve its overall effectiveness.

The techniques mentioned above are used to monitor the health of the equipment and on indication of pump distress, the cause and severity of this distress is also indicated.

If a pump is observed to be in distress, it should be taken up for corrective actions immediately for to prolong maintenance can lead from primary failures to secondary failures. This increases the downtime and the maintenance costs.

In case of non-essential pumps, it is possible that a pump is left unmonitored or unanalyzed until failure is encountered.

It is important to define the term 'failure' now. The word 'failure' does not necessarily imply a pump seizure or a heavy seal blowout forcing a shutdown of the pump.

The pump is installed to perform a function under specified conditions. When it is no longer capable of performing a function under the specified conditions it is defined as a failure.

For example, if the pump has to deliver a certain flow rate, consuming a certain horsepower and if this horsepower consumption exceeds a specified limit, the pump is stated to have failed even though the operation of pump may appear to be very smooth.

7.2 Pump Breakdown and Removal

In real applications, breakdown of pumps is a common event. The typical failure causes are:

- Mechanical seal failure
- Excessive vibrations
- Pump rubbing or seizure
- Inadequate performance (flow rate, head developed, power consumption)
- Leaking casing

Before a pump is removed for a repair or an overhaul, it is essential to know the failure type before its dismantling in the workshop.

Thus prior to the removal of the pump the following exercise must be conducted:

- Check with the operating cause on the perceived cause of failure
- If it is safe to operate the pump, rerun it to diagnose failure by
  – Sight, smell, noise, and touch
  – Perform a vibration analysis
  – Measure bearing temperatures
  – Carry out a performance check.

The above steps will positively indicate the probable cause of failure and ascertain that it is specific to the pump and not to the process or pumping system, which includes the suction and discharge system associated with the pump.

If the above analysis indicates the problem is due to the pump, the following sequences of field checks are recommended:

- Check flush lines and quench lines for leak, corrosion, or plugging
- Check the balance line if included in the pump design
- Check the suction strainer for blockage and insure all valves are open
- Visually check the oil condition and the oil level
- Check coupling for wear or lack of grease
- Check the condition of the gages
- Observe the condition of the base plate and pump supports
- Isolate and danger tag the pump valves and motor (pad lock motor switch)
- While dismantling notice the pipe strain on the pump
- Once the pump is decoupled, measure the radial and end clearance of both thepump and the motor

- Run the motor and confirm satisfactory operation. If unsatisfactory, the motor should also be removed for repairs.

7.3 Single-Stage Pump Dismantling and Repair

Once the pump is taken out for an overhaul, the observation process does not end but becomes even more focused.

A recommended procedure is to match and mark all parts prior to the dismantling of the pump and make the following checks:

- Open the pump casing bolts and separate the pump casing or the volute.

- Inspect the pump casing for corrosion and erosion after it is removed. The gasket face should be particularly inspected to insure that there is sufficient land area and the surface is smooth. In case damage is observed, these should be repaired in appropriate manner.

- Similarly, inspect the impeller and the nut for wear, corrosion, and erosion. Impeller vanes should be inspected for pitting or cavities in the areas mentioned in Section 3.12.2 to look for signs of cavitation, suction, or discharge recirculation.

- Inspect the impeller and casing wearing rings for any signs of rub and/or detachment. This can be an indication of excessive shaft deflection, improper assembly, or piping strain.

- Open the seal flange nuts and check for the seal tension.

- Remove the impeller from the shaft after opening the nut. In case it has balancing holes, these should be unplugged. Wearing rings on the backside should be inspected carefully.

- The pump shaft sleeve is then pulled out along with the rotary head of the seal. All these should be carefully dismantled and checked for weak signs on the primary seal face, as well as, the secondary seals like the O-rings and gaskets if any.

- The bearings are checked for roughness. The endplay is measured as shown in Figure 7.2. Shaft is also checked for any signs of corrosion of erosion.

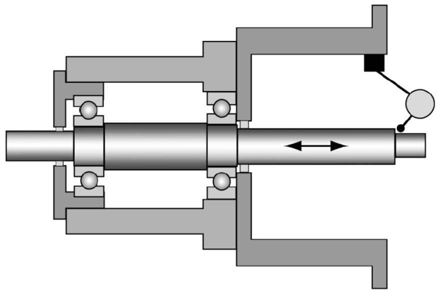


Figure 7.2 – Method to Check Endplay or Axial Float of a Pump Shaft

The endplay or axial float of a pump shaft is measured by placing the pointer of a dial gage against any step on the shaft. The magnetic base maybe fixed to seal housing or any other convenient location.

The shaft end is then lightly pumped over to one end and the dial reading is set to zero. The shaft is then moved over to the other end and a reading is recorded again. The endplay should be in the region of 0.001–0.003 in. (0.02–0.07 mm).

An axial float greater than this leads to pitting and fretting in the points of contact in the shaft packing or mechanical seal areas.
On finding a higher float, one should ascertain whether this is due to an improper assembly of bearing or bearings in the housing or due to defective bearings.

- Shaft runout is a measure of shaft roundness combined with a permanent bow or bend. The effect of shaft runout is greatest on a mechanical seal due to the orbital motion of the shaft. A shaft with a 3mils runout (0.075 mm) will move the seal faces 3mil per side or a total of 6 mils. In addition, a bent shaft tends to cause vibrations that have a large impact on the life of mechanical seals and bearings. This check can be carried out after the shaft is bare and the bearings have been removed. The shaft is placed on V-blocks that are resting on a flat machinist's table.

Dial gage pointer is placed at different locations and readings are taken while the shaft is rotated by hand.

Alternately, the shaft can be clamped at lathe centers and reading. The bearings journals are set at zero and runout at other locations is measured by keeping the dial gage on the tool post.

The next method, as shown in Figure 7.3, is recommended only when the pump bearings are in good condition.

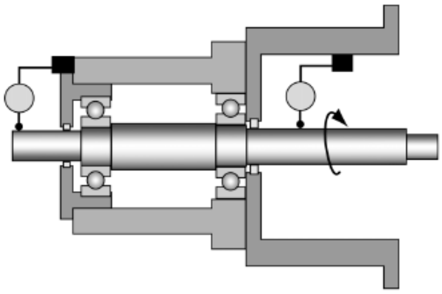


Figure 7.3 – Method to Check Bent and Runout of a Pump Shaft

In this method, the dial gage is fixed on any part of the housing. The pointer is placed on the OD of the shaft at various locations. The shaft is rotated by hand.

The runout should not be greater than 0.05 mm. When a pump sleeve is mounted on the shaft, then the runout should be measured on the sleeve OD.

The combined runouts of the shaft and the sleeve should be 0.05 mm. It is recommended to contain the maximum runout of the shaft and the sleeve to within 0.025 mm. Thus even in a worst case, the runout will not be more than 0.05 mm. In case the reading is higher, the shaft should be straightened.

Shaft radial movement check is next on the list and this can cause shaft whip, deflection, and ultimately, vibrations. This usually occurs when the bearing fits are loose due to corrosion, wear, or improper machining. Even defective bearings can cause this to happen.

The method to check this condition is shown in Figure 7.4. A dial gage is fixed to the housing as close as possible to the inboard bearing housing and the pointer is placed on the shaft outside diameter (OD). The shaft is then lightly lifted and pressed downward and the dial reading is recorded.

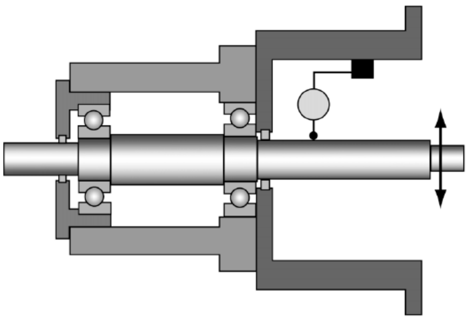


Figure 7.4 – Method for Checking for Shaft Whip or Deflection

Any reading above the value of 0.07 mm is unacceptable and calls for corrective actions.

- Seal housing squareness is an important check to insure that the seal faces are parallel to each other when they are installed. A seal housing that does not sit square will cause an angularity between the seal faces and cause an unequal wear of the seal faces leading to shortened seal life (Figure 7.5).

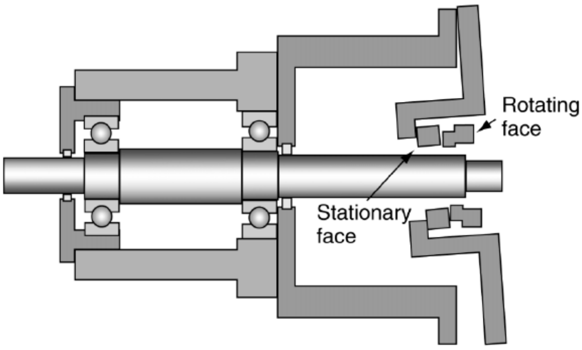


Figure 7.5 – Notice the Effect of Angularity of
Seal Housing on Seal Faces in this Exaggerated View

This check is slightly different from the checks mentioned above. In this case, the magnetic base of the dial gage is placed on the shaft and the pointer is placed on the face of the seal housing as shown in Figure 7.6.

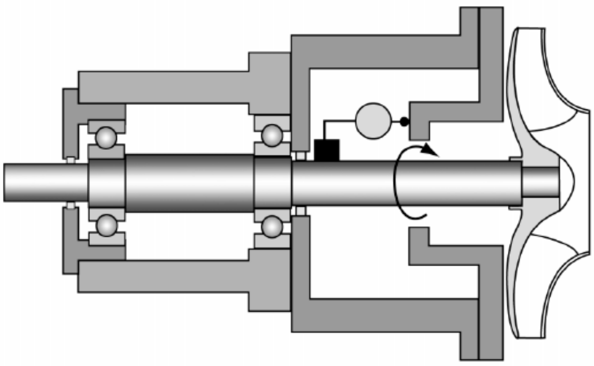


Figure 7.6 – Method for Checking for Stuffing Box Squareness

The shaft is then rotated and facial reading is recorded. The allowable value is 0.05 mm.

In case a higher reading is obtained, the seal housing should be put on a lathe and face should be squared with the mating face of the seal housing with the bearing housing.

The seal housing face should be free of any nicks, burrs, or any other surface defect.

- One type of seal assembly problem occurs due to the angularity of the seal housing as mentioned above. The next problem is the offset of the seal faces.

  This occurs when the seal housing is not concentric with the shaft axis (Figure 7.7)

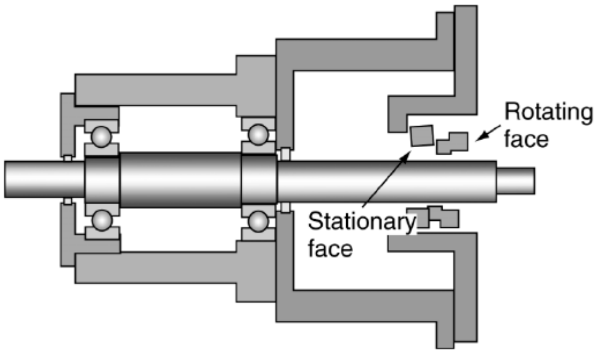


Figure 7.7 – Notice the Effect of Non-Concentricity of Seal Housing on Seal Faces – Exaggerated View

The way to record the seal housing concentricity is shown in Figure 7.8: A dial gage is fixed to the OD of the shaft and the pointer sweeps the inside bore of the seal housing. The inside of the bore may have a rough surface because of corrosion or wear. In such a case, it should be cleaned with sandpaper, washed, and dried with a solvent.

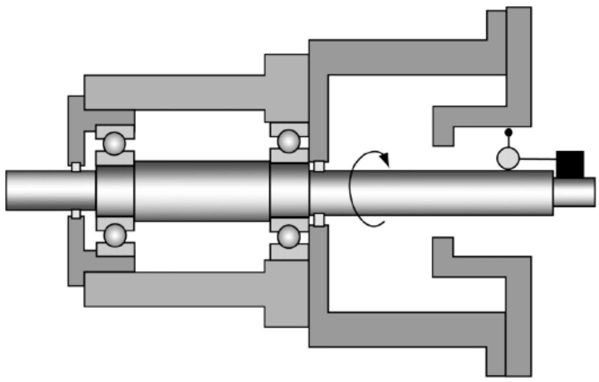


Figure 7.8 – Method for checking for stuffing box concentricity

The allowable limit for concentricity is 0.05 mm. In case the reading is in excess of the allowable limit it should positioned and doweled until concentricity is achieved.

In in-between bearing pumps which have two seal housing located at the two ends, this check is even more critical.

- In case the bearings are found to feel rough and/or there is excessive radial and axial float, the bearing housing should be dismantled to replace the bearings. Usually, the outer race of the bearing has a transition fit with the housing and the inner races of the bearings have an interference fit on the shaft OD.

  Thus, the pump shaft with the bearings can be drawn out of the bearing housing with slight tap of the hammer.

  The bearings can be removed from the shaft usually by using a hydraulic press. Once the bearings have been removed, the dimensions of the bore and the shaft OD need to taken and the bearing fits should be worked out.

  The shaft at this stage can be checked for straightness as mentioned earlier.

  After this is checked, it is polished and kept ready for reassembly. If the bearing fits are found to be excessive, suitable repair procedures have to be worked out depending on the material of construction and size of the shaft.

  The housings can be re-sleeved with rings of the same material of construction.

  The shaft diameter too can be subjected to weld build-up and re-machining.

  However, this has to be done considering the material of construction and thickness of the section.

  Some materials may need a heat treatment and a smaller diameter shaft could warp if the weld procedure results in excessive heat build up.

7.4 Preparation for Reassembly

7.4.1 Impellers

The pump impeller is a rotating part, subjected to high velocity; and the adverse effects of poor hydraulic design can be a cause of wear and erosion. As mentioned earlier the vanes need to be examined for cavitation and recirculation phenomenon. This information needs to be communicated to the operations.

Any corrosion of impeller should be viewed seriously and a review of the material of construction should be made if the corrosion rates are excessive.

A badly worn/corroded impeller should be replaced.

When the impeller is to be reused, the fit of its bore with the shaft should be checked and that of its keyway.

The diameter of the wearing rings should be measured and compared with its pump casing, its mating ring. The clearances should be in line with the API 610 (7th Edition) recommendations. These are given in Section 2.3.

This clearance is probably the one factor that has a major impact on the efficiency of the pump. A higher clearance leads to higher recirculation losses.

Figure 7.9 is a graph for double suction pumps showing the relationship between the percentage power loss and the specific speed. (Adapted from – When To Maintain Centrifugal Pumps – I.J.Karassik, HP-April 1993)

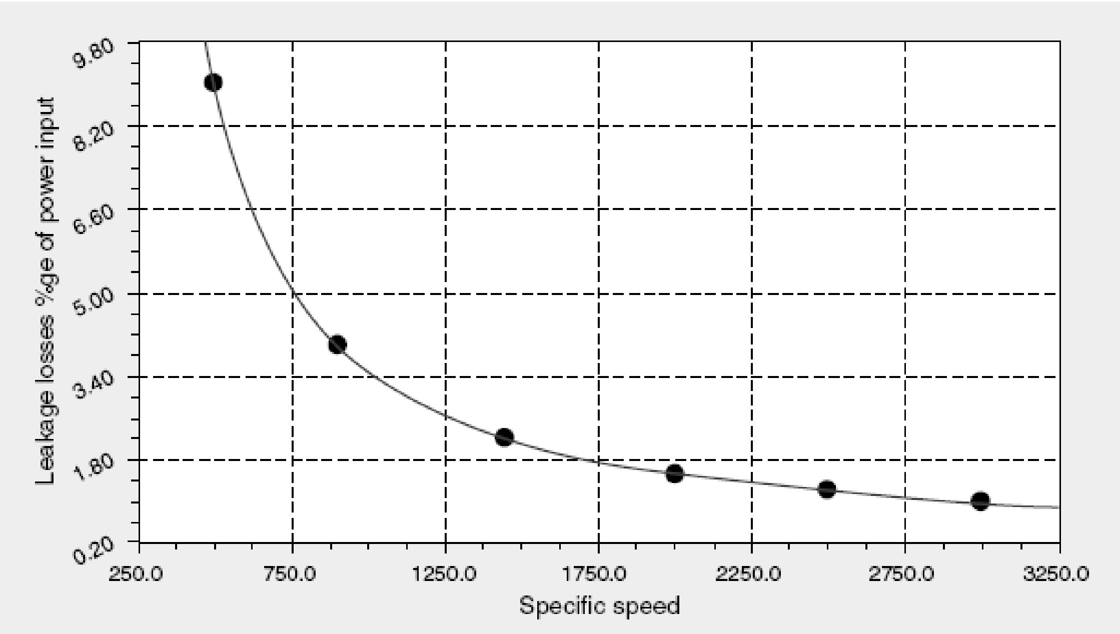


Figure 7.9 – Results from an Experiment of
Wearing Ring Losses in Double Suction Pumps

From the graph, it can be seen that the percentage power losses due to wearing clearances are much higher in lower specific speed pumps.

Restoring back the clearances for such pumps gives higher returns in terms of leakage loss reduction. Thus, such pumps need more attention with respect to wearing ring clearances.

The wearing rings are press fitted to the impeller, casing, and held in place by the following ways:

- Tack welding and then grinding off the excess bead
- Threaded wearing rings (against the direction of rotation)
- Drill, tap, and fasten with grub screws in the radial direction. The diameter of the hole is not more than one-third the width of the ring
- Drill and tap at the ring and impeller/casing interface and fasten with grub screws in the axial direction.

API 610 generally does not recommend the first two methods.

After the wearing rings have been renewed and machined to obtain the requisite clearance, it is recommended to balance the impellers with allowable limits as specified by the API. The ISO 1940 standard that recommends a grade of G 6.3 for centrifugal pumps is far too liberal.

However, in case of pumps with long shafts and big impellers and heavier coupling halves, it is recommend carrying out the balancing after assembling the complete rotor. Two-plane rotor balancing is recommended over single-plane balancing of the pump impeller.

7.4.2 Pump Casing

All fits of pump casing and mating parts should be measured and recorded. All plugs and fittings must be removed, cleaned, inspected, and fitted back. The mounting pads should be flat and parallel to the pump centerline.

The casings should be cleaned. The gasket area should be polished with emery paper to ensure that no pieces of the old gaskets adhere to the surface.

In case there are signs of corrosion or wear, the possibility of coating the internals with resistant epoxy or its equivalent should be considered. A repair work is shown in Figure 7.10 for the internals of casing affected corrosion and pitting in salt-water application.

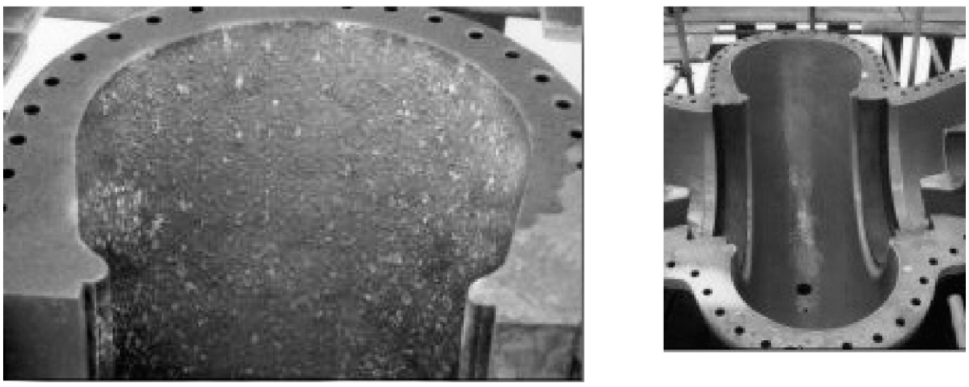


Figure 7.10 – Surface of a Pump Casing before and after a Finished Coat

The procedure involves descaling of the internals by ash, sand, or grit blasting. In the case of salt-water applications, steam cleaning is applied to insure all salt contamination is removed. This is again followed by grit blasting.

The final coating is done using epoxy compounds to provide the finish as shown in Figure 7.10.

Such coatings smoothen the surface roughness of the liquid path and thus reduce the hydraulic losses within the pump. This has positive impact on the efficiency of the pump; in some cases, this can be as high as 10–15%!

The casing bolts are cleaned and the right number are arranged and coated with anti-seize compounds.

7.4.3 Seal and Bearing Housing

The cleanliness of the seal and bearing housings is of utmost importance. The cleanliness required in the housing has a large impact on the life of the bearings and seals.

If the seal housing has cooling water jackets, it is recommended to get them chemically cleaned periodically.

The bearing housing mating faces with the seal housing should be inspected and checked to insure a proper fit.

The faces of the bearing housing covers are checked for smooth surface finish. The oil seals or any other shaft seal is replaced.

7.4.4 Shaft

The shaft is checked for straightness, condition of threads at the impeller nut location, keyway fit at impeller and coupling locations and fits for the anti-friction bearings.

In case the shoulders or steps are rounded, these are dressed up/repaired.

The location of lip seal area is inspected. In case these are damaged it has to be repaired or compensated by a sleeve of appropriate size.

7.4.5 Mechanical Seals

Mechanical seals are the prime causes of pump failure and many efforts to improve the reliability have led to a building of standard for seals called as API 682. This standard recommends the use of cartridge seals. Cartridge seals eliminate the skilled workmanship required to assemble a seal at the users end.

7.4.6 Pump Shaft Sleeve

The preparation for seal assembly begins with the check of the pump sleeve condition. The sleeve areas under secondary seals such as O-rings/PTFE wedges are mostly prone to fretting as shown in Figure 7.11.

Figure 7.11 – Pump Shaft Sleeve

To eliminate this problem, this area of the pump sleeve can be plasma sprayed to provide a hard ceramic coating like chromium oxide, tungsten carbide, or aluminum oxide. The hardness of nearly 70 Rc is obtained. After spraying, this area is ground to the exact dimension.

7.4.7 Seal Faces

The seal faces should be renewed. It is advisable to check the faces for the flatness using an interferometer and an optical flat. Here a monochromatic light source (helium or sodium) is used and an optical flat is placed between the seal face and the light source. The patters observed on the optical flat indicate the flatness of the seal faces (Figure 7.12).

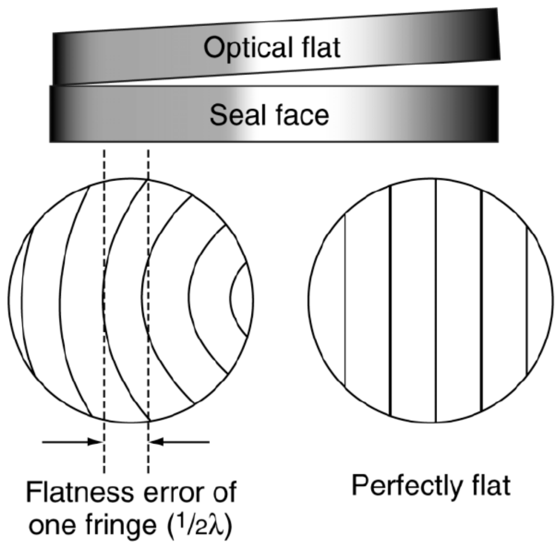


Figure 7.12 – Principle of confirming flatness of seal faces

The optical flat is placed on the piece to be measured. The monochromatic light is aimed at the piece and this light reflects off the piece back through the optical flat causing interference light bands (Figure 7.13).

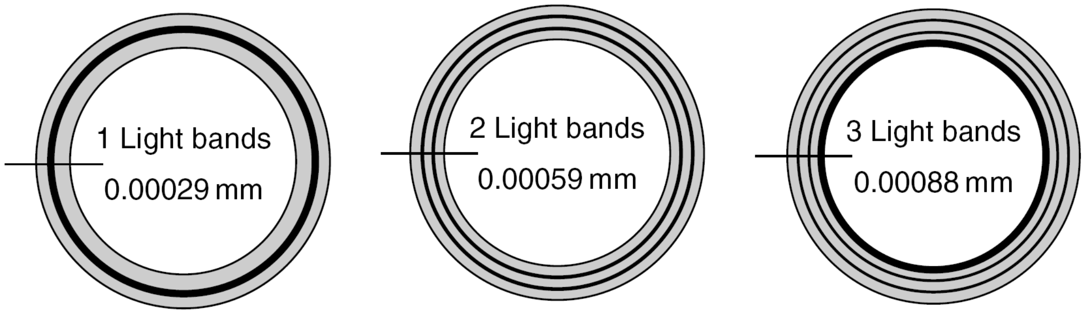


Figure 7.13 – Reading the Flatness of the Seal Faces Under the Monochromatic light Source

If the distance between the optical flat and the piece we measure is one-half the wavelength of helium, or an even multiple of the number, the band will show black. This is referred to as a helium light band and because it is one-half the wavelength of helium it measures 0.3 microns or 0.0000116 in.

The concentric rings indicate that the surface is not concave or convex. When these are present, the concentric rings give way to arcs. However, the number arcs or a ring cut by a straight line across the face is the number of light bands.

The allowable light band for a helium source check is 0.9 microns or 3 light bands. This degree of flatness will allow a mechanical seal to seal vacuum down to a measurement of one Torr (one millimeter of mercury). With this limit, the mechanical seal can easily pass fugitive emission specifications of less than one hundred parts per million.

The carbon graphite faces are known to relax after lapping. Although lapped to less than one light band by the seal manufacturer, we may see the readings as high as three light bands during the checks. These faces should return to flat once they are placed against a hard face that is flat.

It is preferred that the carbon/graphite seal faces should not be relapped because the relapping procedure drives the trapped solids further into these faces. Lapping powder or paste should not be used to lap carbon/graphite faces. They should be lapped dry on ceramic stones of varying grit or finish.

In addition to seal faces, the rotary-head that houses one of the seal faces comprises of a stainless steel retainer, which may house a single or multiple springs. The retainer may have anti-rotation provision.

All these should be checked to insure that they are capable of performing their function. It is advisable to replace the springs.

The grub screws used to fix the retainer on the pump sleeve should be unscrewed, cleaned, and reused.

When bellow seals are used, these should be inspected carefully to look for any cracks at the weld joints.

7.4.8 Secondary Seals

Once the faces have been checked, they are wrapped and stored carefully in an airconditioned and dust-free room until it is time for reassembly.

The secondary seals include all O-rings or gaskets to be used with the stationary seat, rotating head, and the pump sleeve.

During inspection if any of the O-rings were found to have swelled, shrunk, corroded, nicked, cut, extruded, or blistered the cause should be investigated.

For no reason the secondary seals should be reused. The savings will not justify even a pump leaking on the test bench.

Therefore, a new set with confirmed dimensions should be procured and kept along with the seal faces.

7.4.9 Seal Plate

The seal plate that houses the stationary seat should be free of corrosion. Sometimes these are fitted with a stop pin to prevent the rotation of the seat. The condition of the pin should be checked and if necessary, it should be replaced.

The gasket/O-ring surface should be cleaned and polished.

7.4.10 Seal Housing

The seal housing needs to be clean and dry. The internals maybe coated with an anti-rust compound.

The throat bush is fitted to the seal housing. Its clearance with the pump sleeve OD should be confirmed. If it is higher, it should be replaced.

In case it has integral cooling water jackets, these should flush and periodically chemically cleaned.

7.4.11 Bearings and Bearing Housing

New bearings should left pack until they are required for assembly. The housing fits with the bearing should be confirmed when there are repeated bearing failures or when the pump has high bearing temperatures.

Another important check is the concentricity of the bearing housing bores. To carry out this check the bearing housing is placed on a horizontal boring machine. The dial gage is fixed to the boring bar and the readings are measured in the vertical and the horizontal plane.

Alternately, dummy shaft having zero clearance with the housing bore can be used to confirm non-concentricity of the bores.

Bearing housing is typically made of cast iron and in areas where humidity is high and heavy rainfall is common, the inner surface is prone to corrosion. The corrosion dust or particles can be washed by the splashing oil and act as abrasives for the bearings.

Thus, it is necessary to paint the internals of the housing with an oil resistant paint. Prior to assemble, the paint condition should be inspected.

The bearing housings have to be sealed to prevent moisture ingress. The bearing covers that are bolted to the bearing hosing should be installed with proper bearing isolators.

The oil level gage opening should be cleaned and new one should be fitted to it.

The bearing housing vent should also be removed, cleaned and screwed back.

7.4.12 Coupling

The coupling should be inspected for wear and its fit with the shaft. If these are normal, they should be greased if applicable.

7.4.13 Pump Clearance/Overhaul Chart

Preparation for assembly is a detailed exercise and lays the basis for a quick assembly and reliable repair. The preparations require acceptable clearance and accurate data to insure that assembly will not be hampered because of incorrect dimensions and unclear expectations of any sub-assembly procedure.

A standard clearance chart is given at the end of this topic.

7.5 Pump Assembly

The pump assembly described below (Figure 7.14) is for a single-stage overhung impeller centrifugal pump.

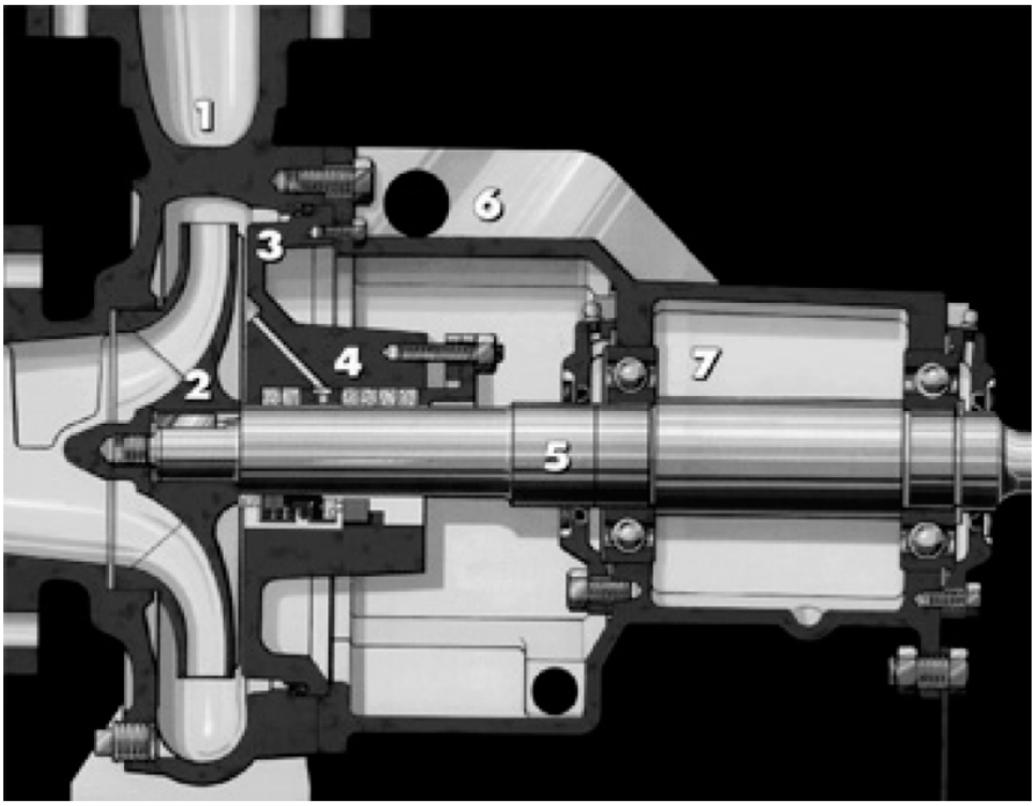


Figure 7.14 – Cross-Section of a Single-Stage Overhung Centrifugal Pump

The reassembly process begins at the pump shaft, which has undergone checks for runouts, condition of steps/shoulders, keyways, and fits at the journals.

7.5.1 Bearing Assembly

- Apply oil on the bearing seat of the shaft lightly.
- Shielding, if any, must face in proper direction. Angular contact bearings, on pumps where they are used, must also face in the proper direction. Duplex bearings must be mounted with the proper faces together. Mounting arrangements vary from model to model (refer to the OFM pump manual).
- The bearing should be pressed on squarely. Do not cock it on to the shaft. Be sure that the sleeve used to press the bearing on is clean, cut square and contacts the inner race only.
- The bearing should be pressed firmly against shaft shoulder. The shoulder helps to support and square the bearing.
- In case the fits are tight, the bearings maybe heated using an induction heater with a de-magnetizing cycle. The temperature should not increase beyond 110ºC.

- The snap rings are then properly installed with the flat side against the bearing, and that lock nuts are tightened.
- The bearings should be lubricated.
- This assembly should then be wrapped in a plastic cover while the bearing housing is made ready.
- Prior to placing the rotor in the bearing housing, it is insured that it is spotlessly clean.
- In pumps, the inboard bearing (closer to the impeller) is near the locating bearing (fixed bearing). The bearing cover that provides the step to the outer race of the bearing should be fixed. This is done after a sealing compound has been applied to the mating faces.
- Oil is smeared on the bearing housing bores.
- Shaft with the mounted bearings is tapped in the bearing housing till the outer race of the inboard bearing rests against the step.
- Apply sealing compound on the outboard mating face for the bearing cover and tighten the bolts.
- Another type of bearing assembly is depicted in Figure 7.15. The outboard bearing is the locating bearing and is placed in a housing that can be screw jacked axially after loosen the clamping bolts. The advantage of this arrangement is the rotor can be moved axially without dismantling the pump.
- This is particularly useful in setting the clearance between the back vane impeller and the casing wall.

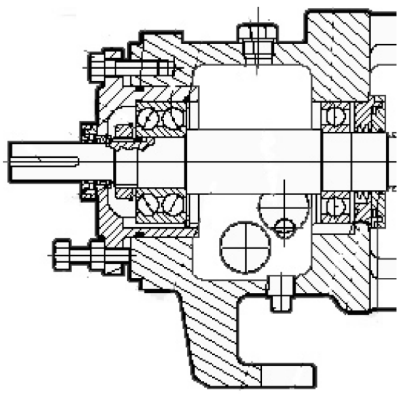


Figure 7.15 – A bearing Assembly – Outboard Bearing is the Locating Bearing

- In this assembly, the bearings are fitted on the shaft and the lock nut is tightened against the outboard bearing. The housing is then slid over the outer race and snap ring is slipped in the slot.
- This rotor assembly is then placed in the bearing housing and the bolts are tightened.

7.5.2 Seal Assembly

- The stationary seat is firmly clamped with the seals in the seal end plate.

- This is then bolted to the seal housing.
- The seal housing is then carefully slid over the shaft so that there is no impact between the shaft surface and the stationary seat.
- It is then bolted to the bearing housing taking care of match marks made prior to dismantling.
- The above is necessary for if even a single bolthole is out of phase it may not be possible to make the sealant connections in the field.
- Once this is complete, the next step is to adjust the seal tension.
- The springs of a seal have a certain effect on the face pressure and a certain seal tension has to be maintained. In a seal design comprising of multisprings, the seal tension is generally of the order of 3–4 mm depending on the size of the seal. Single spring seals have a much higher seal tension.
- To arrive at the correct seal tension, the following measurements need to be taken. The figures below help to explain this procedure. The rotary head of the seal is assembled and total length is measured. This is the free length from the edge of the sleeve to the seal face without compression. Let us call this length as F (Figure 7.16).

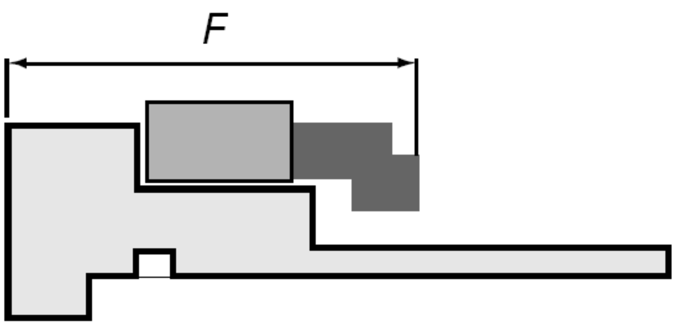


Figure 7.16 – Sleeve installation

Before we install the sleeve with the rotary head on the shaft, we need to measure the distance between the shaft step and the face of the stationary seat. This can be measured directly or by using a frame of reference that is based on the construction of the pumps. This is shown in Figure 7.17. Let us call this distance as A. The next step is to know the length S of the sleeve. This is a differential between easily measurable lengths 'L' and 'M' (Figure 7.18). Once F, S, and A are known, then F – (S + A) = 3 mm. This seal tension will be achieved once the seal is assembled.

- Some sleeve designs have an O-ring to prevent leakage through the gap between shaft OD and the sleeve ID while some designs may require a gasket as shown in Figure 7.19.

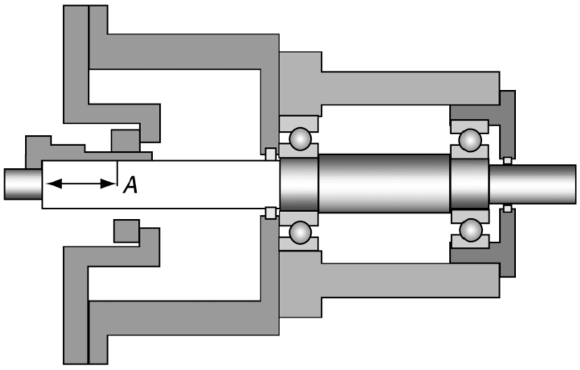


Figure 7.17 – Sleeve Installation Step 1

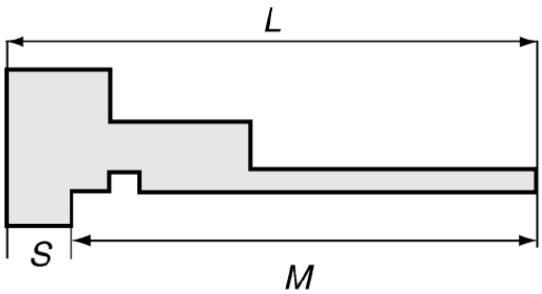


Figure 7.18 – Sleeve Installation Step 2

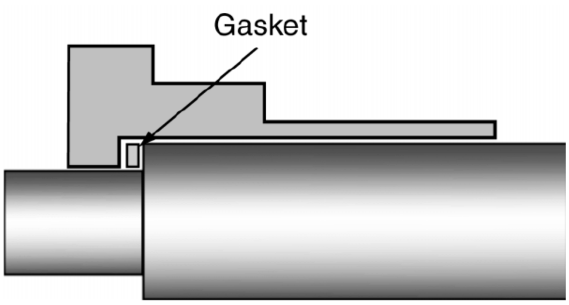


Figure 7.19 – Typical Sleeve Design

In this case, the thickness of the gasket has to be taken in account for computing the seal tension. If G is the gasket thickness, the new formula is Seal Tension = F – (S + A + G) In case the seal tension is improper, cuts on the sleeve steps can be taken. Caution: Care has to be taken that the step of balance carbon and sleeve step does not foul up once the seal is under compression.

- Squirt oil from an can to lubricate the faces of the seals
- Insert the sleeve with the rotary head
- The seal tension adjustment can be eliminated by the use of cartridge seals (Figure 7.20). These come with pre-adjusted seal tension.

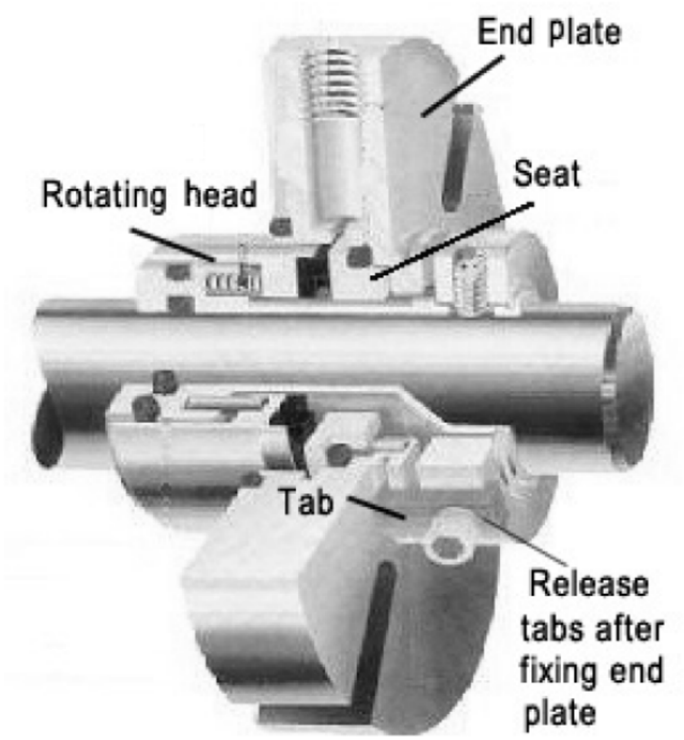


Figure 7.20 – Cartridge Seal – Seal Tension is Achieved after Releasing Tabs

7.5.3 Impeller and Casing Assembly

- Once the sleeve with rotary head is placed, the compression is achieved after installing the impeller and locking it to the shaft with the help of the nut.
- After the impeller is fixed, it is a good idea to measure the runout of the impeller wearing ring in this installed condition. This should be within 0.05 mm.
- The casing gasket is placed in the pump casing.
- The casing is then bolted to the seal-housing flange, once again taking care of the match marks.
- The bolts are tightened to the specified torque value.
  Note: The correct torque for bolting is based on the size, thread, and material of the bolt. Another very important factor is whether the bolt has been lubricated as say by application of any anti-seize compound.
  If such a compound has been applied, then the tightening torque values should be set to approximately 2/3rd of the torque as specified for dry bolts.
- This completes the main assembly.

7.5.4 Seal Hydrotest

- If the workshop has a provision to test the seals, it should be done so.
- To carry out the above test, blind flanges are bolted to the inlet and outlet nozzles of the pump.
- The blind flanges should be drilled and tapped to screw in fittings for filling the pump with water and venting air from the top most point.
- Water is then filled in the pump and all air is vented out.
- Pump is then pressurized to a minimum of 1–2 kg/cm□ above the shut-off discharge pressure of the pump.
- Once pressurized, compressed air should be used to dry the pump externally, especially near the seal and shaft interface.
- The shaft is then rotated slowly by hand and pump should be observed carefully for any leaks from the seal.
- The casing joint should also be examined.
- Any drop in the hydrotest pressure should be investigated.
- If the hydrotest is satisfactory, the pressure should be released and the water should be drained.

7.5.5 Installation of Coupling, Lines, and Fittings

- The pump coupling half can now be mounted onto the shaft.
- Shaft end should preferably be flush with the coupling half. However, if it was not so originally, this should have been recorded and kept along similar lines.
- The sealant lines, oil level gages, cooling water lines, or any other should be fitted.
- After the blind flanges are removed, the pump nozzles should be completely covered by a tape.
- The pump is ready for installation at site.

7.6 Vertical Pump Repair

The construction of the vertical pumps basically comprise of:

- Head/driver assembly
- An electrical motor and a cast or fabricated base from which the column and the bowl assembly is suspended
- Column and shaft assembly
- The column pipe is the link between the head assembly and the pump bowl assembly as shown in Figure 7.21. Its main function is to conduct the liquid from the bowl to the discharge. Within the column pipe is the pump shaft that transmit the power from the driver to the pump impellers. The pumped liquid generally lubricates the line shaft bearings in the column
- Pump bowl assembly
- Each bowl comprises of impeller, suction, and discharge casing

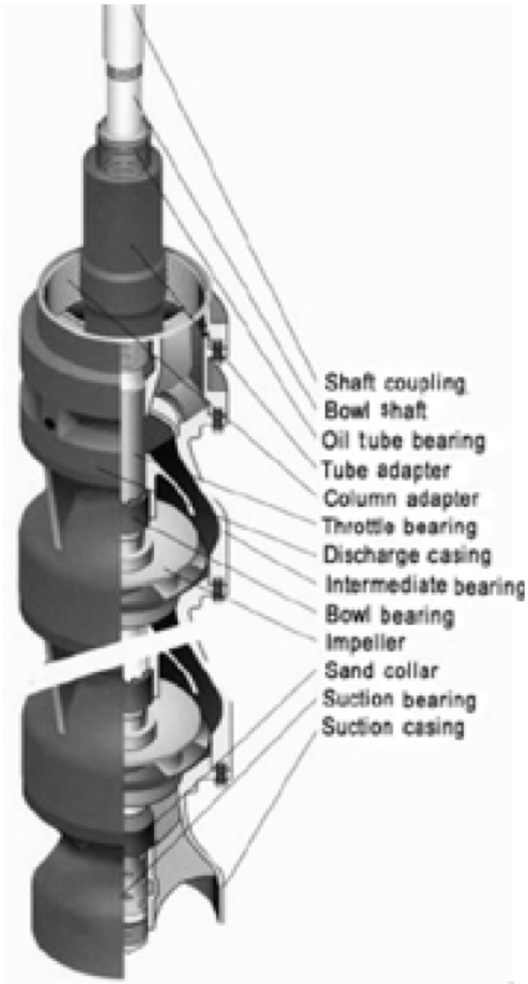


Figure 7.21 – Details of Vertical Turbine Pump
(American Turbine Pumps)

There are three main reasons to overhaul or repair a vertical turbine pump:

1. One of them is the complete pump breakdown leading to unexpected disruption of a critical system or plant operations.
2. Reduced pumping efficiency due to increased clearances between the bowls and enclosed impeller skirts, increased clearances between the bowl shaft and bearings, and breakdown of the fluid passages. Over time, running a badly worn pump can be just as costly as some system shutdowns.
3. When operational conditions demand altered requirements in pressure and capacity.

The vertical pump comprises of a lot many more components than a simple horizontal pump and the maintenance problems are also different and can be substantially higher.

The rotor of the vertical pumps not stabilized by gravity is prone to gyroscopic effects that can cause damage to the rotor and the casing. Thus, the repair of such pumps demands care and attention.

7.6.1 Pump Dismantling and Repair

The repair and dismantling procedure of a vertical turbine pump is influenced by its construction. Pump repair is a lot more complex than a single-stage pump and it is advisable to prepare a datasheet for the pump.

This datasheet should comprise of all pump details, clearance and fits data, and any special remarks and procedures:

- The first step in the disassembly procedure should be to make the pump lie horizontally on the floor with the discharge nozzle facing the ground.
- Take a paintbrush and match mark all the joints. As the bowl joints can look similar, it is advisable to punch consecutive numbers on the joints taking care that these are not in addition to some other numbers of a previous overhaul. This would lead to a lot of confusion during assembly.
- Push the pump upper shaft toward the suction and measure the distance to the motor adapter face and adapter face configuration.
- Now pull the rotor outward and re-measure this distance to get the total axial float of the rotor.
- The pumps' bowls maybe individually screwed together, bolted together with a series of bolts around the bowl flange (most common design) or bolted together with tie bolts that extend from the bottom of the bottom to the top of the pumping section through the bowls. These maybe opened one section at a time.
- In the pump design, where a split collet bushing is used to lock the impeller on the shaft there the distance between the shaft end and the impeller must be measured and recorded.

- The float is measured and recorded at each stage (Figure 7.22).

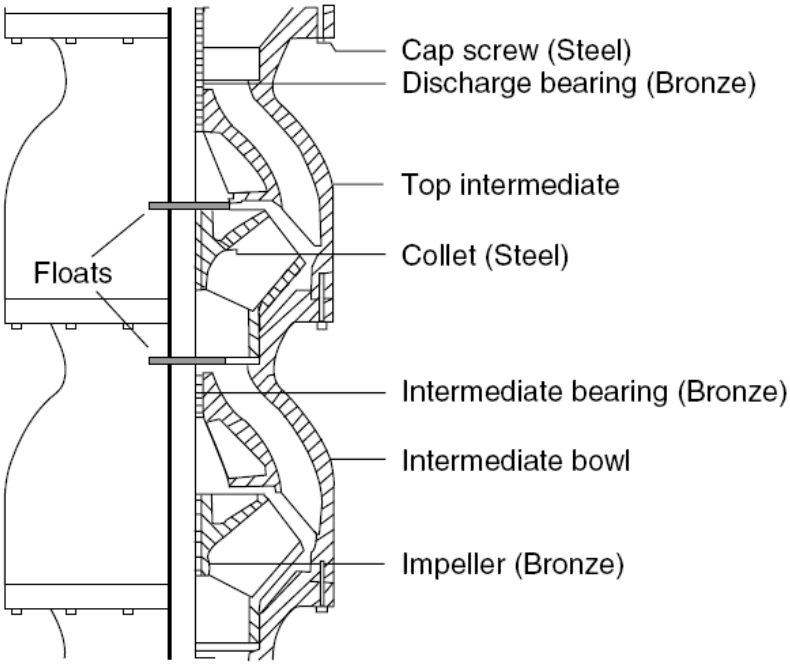


Figure 7.22 – Sketch Indicating Floats in Assembly

- As the disassembly progresses, it is essential to match mark the impeller, collets, keys, locking pins, split keeps, and keeper plates. These should be arranged and kept in an organized manner.
- Similarly, the line shaft couplings should be match marked along the mating shaft on each end.
- The line shaft couplings should be removed using a pipe or chain tongs on the mating shafts. Wrenches applied to couplings can cause the coupling to collapse and seize on the shaft. If normal methods fail, the coupling can be cut using thin grinding wheel. The main aim is to prevent damage to the shaft, as it is the most expensive item.
- The rigid coupling halves from the motor and pump should be checked for trueness and squareness.

7.6.2 Preparation for Assembly

Many checks and repairs have to be carried out in this stage:

- The fluid passages of the bowl and impeller need to be carefully examined for wear. A significant impact on efficiency can be observed if the above are damaged, pitted, have a particle build-up or have eroded coatings. Coatings are normally applied to the interior of the bowl assembly, mainly to enhance efficiency or for the protection of internals. However, the velocity of the fluids and suspended solids being pumped often damage the coating. Cleaning, recoating, or replacing the affected parts are options dependent on the severity of the problem.

- The next step is to check the condition of the shaft. The bearing areas are susceptible for wear. If the condition is found to be good, the next step is to measure the shaft runout. A typical layout for this procedure is shown in Figure 7.23.

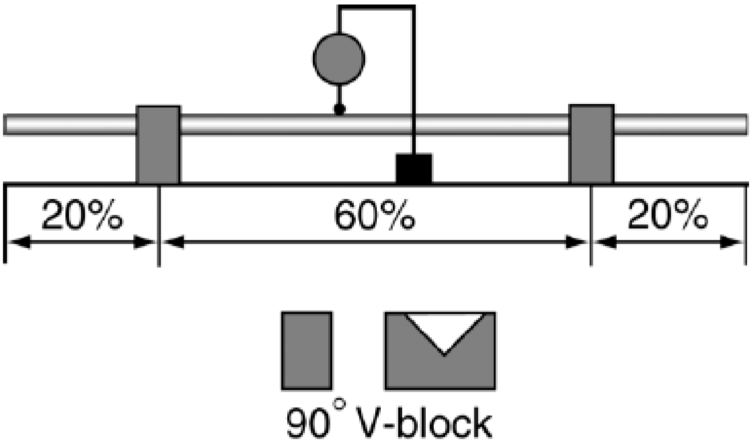


Figure 7.23 – Measuring Shaft Straightness

The shaft is placed on V-blocks or knife-edge rollers, if available at the recommended distances. The dial gage is placed at various locations and the shaft is rotated a full circle by hand. The dial readings are measured and recorded. The shaft runout should be within 3–4 mils (0.075–0.1 mm). If it is higher, efforts should be made to straighten it within the tolerances using a press.

Whenever possible, a single-piece shaft is recommended. If more than one piece is used and couplings are required, a radial weep hole should be provided in the center of the coupling. The shaft should be screwed together and runout is checked.

The bowl shaft is typically made of some grade of stainless steel (typically 12% Cr steel, ASTM – A-581, A-582, hot rolled bar, turned, ground and polished) rather than carbon steel because of the resistance of stainless to abrasion/corrosion.

If abrasive solids are pumped, shrunk-fitted sleeves can be fitted in the wear areas. Another solution can be the application of chrome or other hard coatings at these locations on the shaft.
The shaft could be in lengths of 20–24 in. with a straightness tolerance of 1.5 mils (0.4 mm) per foot (300 mm). The recommended surface finish is 16 µin. RMS

- Impeller to bowl wearing ring clearances are measured and recorded. The wearing ring clearances should be in accordance with the API 610 (7th Ed.) as tabulated in Section 2.3.

If the bowl does not have a wearing ring, then it should be checked to see if there is sufficient material available to bore the bowl and/or turn the impellers.

Nevertheless, wear rings can replace the worn material and bring the parts back into tolerance.

The fluid contains some particularly abrasive solids and the wear rings are made of materials to minimize the abrasive erosion. If wear rings are ordered with the original unit for easy replacement later, both bowl and impeller pieces are necessary to return the bowl assembly to its original tolerances.

- All impellers should be balanced individually as per API specifications.
- All bearing bores should be checked for parallelism with the shaft column.

- All bushings in the bearing retainers should be measured for their clearance with the shaft. If they are replaced because of excessive clearance, they should be properly secured to the retainer as per their design and material of construction. These bushings generally have a spiral groove for lubrication as the clearances are generally on the tighter side (refer Figure 7.24).

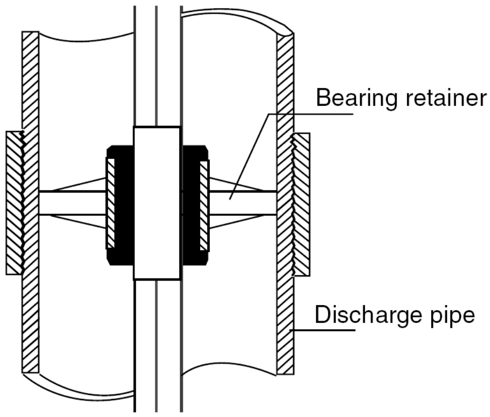


Figure 7.24 – Sketch Showing Line Shaft Bearings

Most often, the bushings are bronze, rubber, or a combination of the two. Other materials used include bronze-backed rubber bearings, Teflon, and carbon composites.

Bronze bushings are not used in corrosive, sandy, dirty water service. The allowable temperature range is from –30 to 90°C. Above a pumping temperature of 90ºC, cast iron bushings maybe used but these can withstand mild corrosive duty.

When corrosive or lube lubricity products like hydrocarbons are to be handled, bushings made from carbon are used. These are not recommended for temperatures below –30ºC.

The rubber bearings are typically made from Neoprene. These are mostly used in sandy or dirt sump application. These may not be used at higher temperatures.

When left with little choice coke-filled or glass-filled bushing could be used after proper consideration.
The typical shaft/bearing clearance is 0.010", except when rubber bearings are used. In that case, the clearance can run up to 0.030", which makes it a much more forgiving material in applications where abrasive solids are present.

- The pipe maybe pitted or encrusted with scale, or the coating maybe damaged.

Considering the degree of roughness it can be decided whether the pipe should be reused, recoated, cleaned, or replaced.

The faces of the tubes should be inspected for leakage or damage, and the tubes themselves should be replaced when necessary.

Enclosing tube bearings should have approximately the same tolerances as the bowl bearings (10 mils or 0.25 mm). Figure 7.25 show another drive shaft configuration with a separate oil lubricated column which offers extended bearing performance, but at a higher first cost, with pay back through the service life of the pump. Open-line shaft bearings should be replaced when they are visibly damaged. Usually, they are rubber and the condition of these bearings does not necessarily affect efficiency, but it is vital for supporting the shaft.

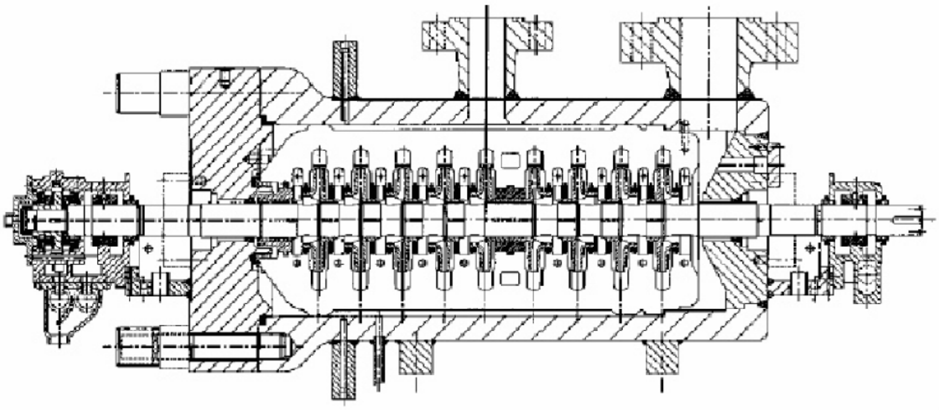


Figure 7.25 – Sketch of a Screw-Coupled Discharge Pipe with an Oil-Lubricated Column

- The discharge head of the pump generally requires the least maintenance. Most of the repairs are performed on the shaft seal, whether it is a stuffing box or a mechanical seal. Packing in the stuffing box should be replaced to control leakage. Stuffing box or seal-housing bearings need to be checked to make sure the shaft is properly supported.

  The repair of the mechanical seals is not very different from what is covered for the horizontal pumps.

  In vertical pumps, it makes more sense to have cartridge seals. In horizontal pumps of the overhung impeller types, the pump rotor has to be dismantled to access the seal, even though it maybe of the cartridge type.

  However, for in-between bearing pumps and vertical pumps, the impeller assembly need not be touched. Thus, seal replacement jobs can be done at site without removing the pump to a shop.

  Changing seals onsite can be tricky and prone to errors on account of site conditions, thus it makes a strong case for cartridge seals in these types of pumps.

- The last check is the condition of the driver. When a vertical pump is started, it has a momentary up-thrust hydraulic action. When the pump is operated at high flows on a continuous basis it is likely to have this continuous upward acting axial force. This can stress the motor thrust bearing besides causing the probability to buckle line shafts, rub impellers, and leak of mechanical seals.

  The thrust bearings should be checked to ensure that they are not excessively worn.

  Motors upgraded to premium efficiency or at least replaced or rewound can significantly increase overall pump performance.

7.6.3 Pump Assembly

- The assembly of the pump is carried in the vertical position. This prevents the risk of shaft or bowl assembly to develop sag. The bowl bushing clearance is typically 9 mils and in a horizontal position it is bound to touch after 3–4 bowl assemblies as the bowl have a spigot fits of 2–3 mils. In a vertical position, these get distributed.
- The jack-bolt in the suction piece must be used to position the end of the shaft to allow for accurate spacing of the impellers. If a load applied on a p reviously installed impellers can knock them off their collet while installing the next impeller. If the impellers become loose, there is this grave danger of a rub on starting.

- Impellers fitted with collets in comparison to those held by splittings; keys, snap rings, and others need a lot more attention. Excessive tightening of the collet can lead to cracking of the impeller in the hub area.
- After the pump bowls have been assembled, the lift should be checked and the rotor should to rotated by hand to check for any rubbing of internals.
- If the length of the pump is large, column sections have to be assembled in a horizontal position.
- The assembled positions should be rotated 180° with installation of every additional component. This aids in staggering the alignment clearances, fits through the length of the columns, and helps to keep the assembly along the shaft centerline.
- After the columns have been fitted, the discharge head is installed. At this stage, the shaft extension length can be compared with the ones taken before dismantling of the pump. Deviations of 1/16th to 1/8th are considered as normal and can be compensated with the use of gaskets.
- The shaft extension maybe supported and the shaft can be locked before dispatching it to the site. If the jack bolt at the suction bell is left in place, it should accompany a warning tag to remove it before installation.

7.7 Multistage Pump Repair

The multistage pump considered here are the multistage barrel type double case pumps. It comprises of two casings; an inner assembly that comprises of the complex shape of the stationary hydraulic passages. This casing sees the external differential pressure so any bolting required to hold it is minimal. The casing can be of the type that is split axially into similar halves and cast from the same pattern as shown in Figure 7.26. Else, this could be of the multiple diffusers casing type.

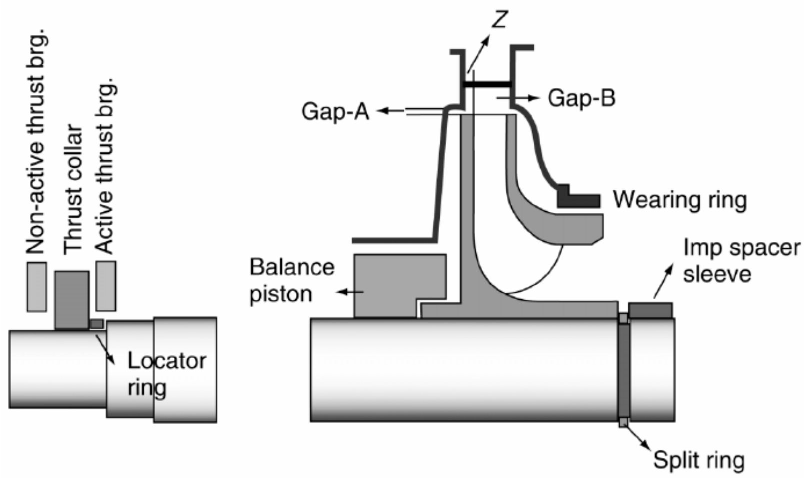


Figure 7.26 – Double Case Pump – Cross Section

The outer casing acts as a pressure boundary for the pumped liquid and is designed as a pressure vessel.

The rotor can be of opposed impeller configuration or of the inline type. The latter has balance disk to take care of axial thrust, which is much higher in this type of design.

These pumps are generally used as boiler feed water pumps, charge pumps handling hydrocarbons at high temperatures and pressures, water-flood pumps and pipeline pumps.

Each of these applications falls under the critical category and their downtimes can be quite expensive.

There are designs that have a fully separable inner case subassembly that includes the rotor. This is done by removing the outboard cover and without breaking the suction line joint.

As downtimes are expensive, a spare inner subassembly is usually made ready and kept as a spare. During a repair requirement, the inner assemblies are just replaced. For some designs that include the journal and thrust bearing, the downtime is further reduced as even these adjustments are made beforehand.

Once the inner subassembly has been dismantled, the shaft runout is obviously the prime check. The runout should be with 1 mil. In the bearing areas, it should be less than half a mil.

In some designs, the impellers are shrunk fitted on the shaft to prevent mechanical looseness and positioned axially by split rings on the suction side of each impeller hub.

The grooves for the split rings cause the shaft to develop the run outs and because of the shrink fits, the shaft fits should be measured and recorded.

The next step should be to balance the shaft with the half keys taped. The shaft is mounted on a balancing machine and spun to a speed of 300 rpm for about 10 min. The shaft runout is then measured at mid-span and the angular position is marked on it.

The balance should be achieved by machining the face of the steps of the shaft. Once a balance is achieved, the runout should be rechecked.
The next step is assembling the rotor for balancing. To do this the shaft is placed vertically. All the rotating components that include the impellers, thrust collar and balance, drum (if installed) are stacked sequentially.

Since most of the components are shrunk fit, these are heated to about 110–120°C depending on the bore and the shaft diameters.

The match marked split ring and the key for the first-stage impeller are first held in place. The impeller keyway is marked with a pen in case it is blind at one end.

The impeller is then heated to 120°C and carefully slipped over the key to its position until it makes contact with the split rings. A constant downward pressure is then applied until the impeller has firmly gripped the shaft. To aid this process, cooling air may then be passed from the OD of the impeller to the suction eye until the temperature of the impeller is ambient.

Install the other impellers in a similar manner.

Once all the impellers are firmly fixed, the runout of the rotor is measured. If the runout is found to be within limits, the impeller nuts are tightened and the runout is rechecked.

If this is also found acceptable, the rotor should be fitted with all the rotating components and sent for balancing.

In case excessive runout is detected, the shaft nuts are the probable causes and their faces should be rectified.

The next step is to balance the rotor. The reader is recommended to read a detailed procedure covered in the paper, 'Monitoring Repairs to Your Pumps' by W. Edward Nelson presented at the 12th International Pump Users Symposium in 1995.

Once the rotor has been balanced, the rotor should be unstacked.

Once again, the shaft is placed vertically and this time the impellers are stacked along with the diffuser elements in a manner similar to the one explained above.

After installation of every stage, the float should be checked. For this purpose, a jack maybe placed under the coupling end of the shaft. The required lift should be maintained throughout the assembly.

The diffuser covers have a small interference fit of 0.01 mm and may require some heat for installation. In case the cover does not fit, the preceding cover should be heated with a small torch around entire circumference for a minute. This cover can then be tapped lightly using a hammer.

After all the impellers and diffusers have been installed, the balance piston is installed.

Once it is cooled, the nut is tightened to the match marks made earlier.

The impeller diffuser lineup or location of the rotor axially to have a gap of 'Z' (shown in Figure 7.27) is adjusted by the thickness of the locator ring placed between the thrust collar and the shaft step (see Figure 7.27).

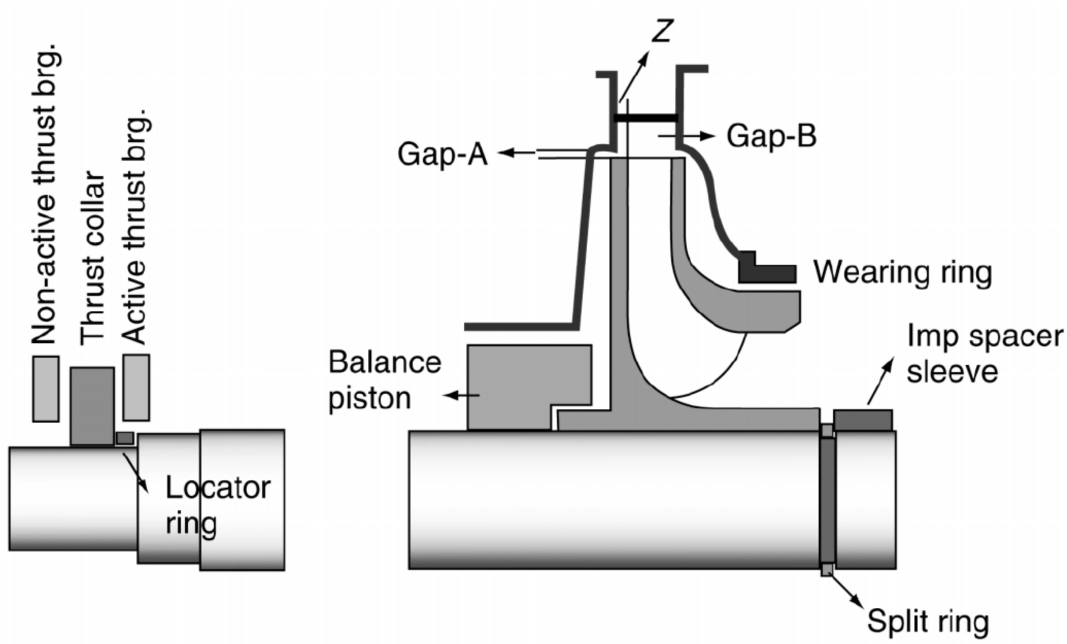


Figure 7.27 – Gap-A and Gap-B

The relation of shroud and hub positions to the diffuser/volute and the casing must be verified and determined during assembly. Axially, the impeller discharge should be entirely within the axial width of the diffuser. If this is not the case, the flow from the impeller will discharge against the diffuser or casing walls. The rotor sensitivity to impeller and diffuser/volute alignment is dependent on the ratio (B2/B3 – see Figure 7.28) of the impeller exit vane width and diffuser vane width and must be judged on a case-by-case basis. However, a typical overlap of not less than 1/2 of the impeller side-plate (shroud/cover) thickness is preferred.

During dynamic balancing, if grinding of the impeller sidewall extends to the impeller OD such that it affects the overlap, the pump performance can get affected.

In addition to this axial location, there are two more famous dimensions, which should measured, recorded/corrected during an overhaul. These have a pronounced impact on the hydraulic stability of any high-energy pump. These are called as Gap-A and Gap-B as shown in Figure 7.28.

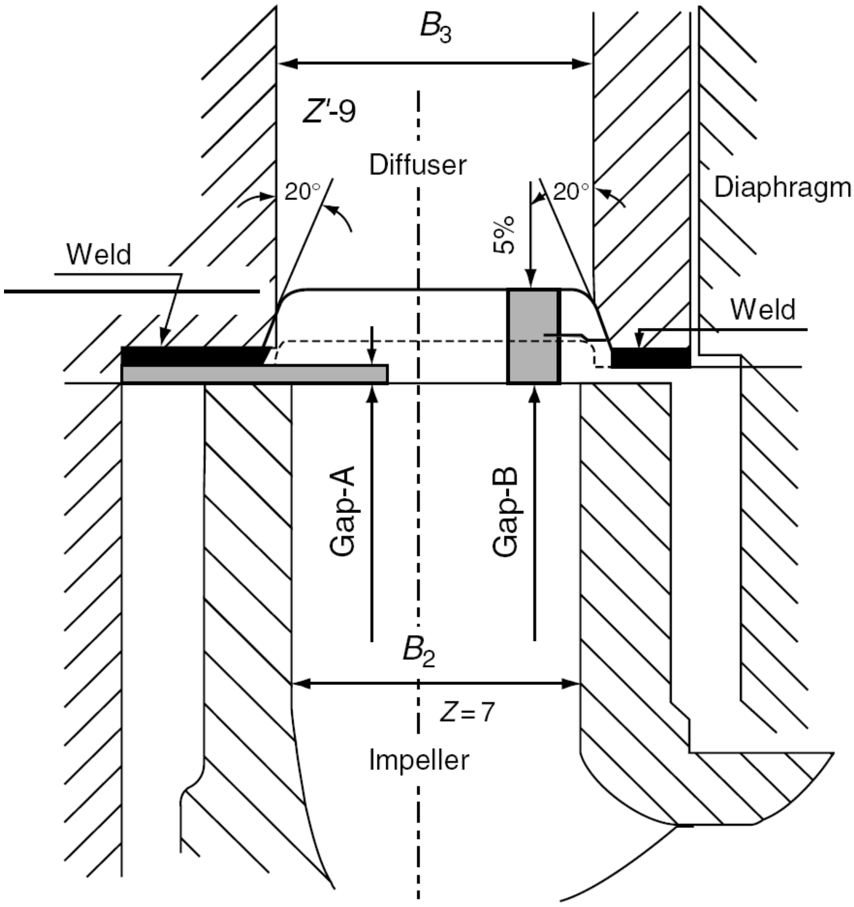


Figure 7.28 – Expansion Gaskets and Spacers

Gap-A is the radial clearance between the outer diameters of the impeller side plates; hub/cover, shroud, and the inner diameter of the diffuser channel side plates.

Gap-B is the radial clearance between the impeller vanes at its outer diameter and the diffuser or volute vane in the inner diameter. This is expressed as a percentage with respect to the impeller vane diameter.

The above gaps have to be maintained in accordance with the specific speed and construction of the pump. Correct gaps eliminate vane pass vibration frequency. Machining of the diffuser/volute vane to achieve the right Gap-B without affecting Gap-A causes no loss of efficiency. In fact, the high noise level, shock, and vibration caused by vane pass frequency are eliminated improving efficiency.

Gap-A has a typical value within the range of 35–85 mils. A preferred gap is of 50 mils, anything beyond 125 mils it losses its effectiveness.

In case of Gap-B, those pumps with a diffuser type construction the range is from 4 to 12% with a preferred gap of 6%. In a volute type of construction, this range varies from 6 to 12% with a preferred gap of 10%.

In case the number of impeller and diffuser vanes is even in number, the radial gap must be larger by 4%.

This assembles the inner subassembly comprising of the inner casing and the rotor.

These are then inserted in the outer casing using a guide fixture.

The expansion gaskets and spacers between the inner casing and outer head are installed as shown in Figure 7.29. Improper inner head gasket spacers can result in improper makeup of the intermediate diffuser covers, the suction spacer and the discharge diffuser spacer could lead to liquid recirculation resulting in drop in efficiency and erosion. The thickness of the spacer along with the gasket should achieve approximately 0.4–0.5 mm in compression on each gasket.

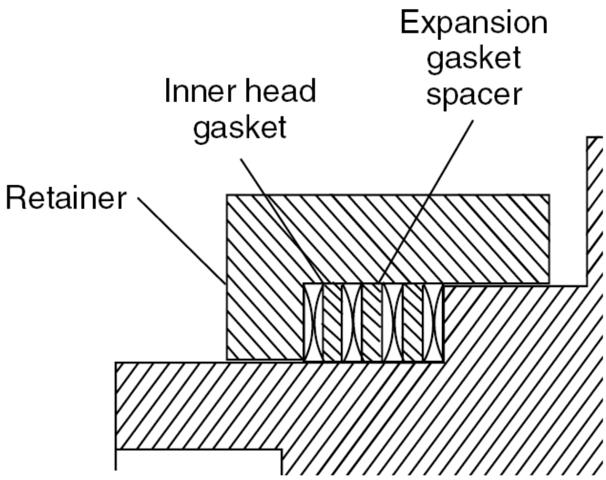


Figure 7.29 – Expansion Gasket Spacer

Next, the head is installed and tightened. In most designs, the head fit is all-metal and the gap at the head should be approximately 1–1.5 mm.

The bearing housings are installed with the lower half journal bearing in place. The coupling end housing is moved until suction impeller is centered in wearing rings within 0.025 mm. This measurement can be made using feeler gages.

The dial gage readings of the seal housing bores and faces are recorded.

The locating spacer and the thrust collar are inserted.

Before the bearings are placed in the housings, the seals are installed.

The bearing housing dowels are inserted and these are bolted.

Finally, the journal and thrust bearings are installed. The location of the thrust collar lines up the rotor axially to get the right gap between the impeller and diffuser.

Coupling halves are installed and made ready for alignment.

7.8 Optimum Time to Maintain Pumps

Over a period, the performance of the pumps deteriorates and it is always a debate whether to live with the problem or take the pump for an overhaul. In the first case, there is a continuous loss of money due to inefficient operation and for the latter; one has to incur the maintenance cost.

One has to work out an equation to determine the optimum time to take up the pump for an overhaul.

The algorithm presented here is the work of Ray Beebe presented in his paper 'Condition Monitoring of Pumps can save Money' at the APMA Seminar.

As per the theory put forward, the economic time to restore lost performance by overhaul will vary with the circumstances.

The basic assumption that is made is whether the:
- Deterioration is constant over a period of time
- Deterioration rate increases with time

In the first case, when the deterioration is constant over time, then a cash flow analysis can be done to insure that the investment in overhaul will give the required rate of return.

The same process is used in deciding on any investment in plant improvement.

When the deterioration rate increases with time, then the optimum time for overhaul will be when the accumulated cost of the increased power consumption equals the cost of the overhaul.

Deterioration in the pump performance can cause:

- Limitation in plant throughput: In such cases, the loss of production is quite high to justify an immediate cost of the pump overhaul and a matter of time to find an opportune window to carry out the overhaul.
- The pump operation is intermittent and does not affect the plant throughput.

  Any deterioration in pump performance only results in longer duration of operation; typically, in case of pumps used for loading products in tankers.

The additional time required for operation resulting in excess power consumption can be compared with the cost of the overhaul:

- In the initial stage, the pump performance deterioration does not affect the plant throughput and it is offset by opening a control valve or increasing the speed but over a period, a threshold is reached and the plant throughput begins to get affected. In some cases, the power requirement may exceed the motor rating leading to overload trips.

The performance calculation procedures described in the previous sections are used to determine the power that a pump consumes.

This can be benchmarked from the pump performance curves.

Using the two, one can determine excess power consumption of the pump and this can be used to trace lost expenses due to poor performance.

The pumps that use up excess power are primary candidates and should take a higher priority. The next thing factored is the cost of the overhaul. This now becomes a case of costs and benefits.

The following example is taken from Ray Beebe's paper for the second case where the pump runs intermittently and loss of performance does not affect the plant throughput:

- Duration of pump operation: 24 months – operation is 27% of this time
- Cost of the overhaul: $50 000
- Cost of power: 10 c/kWh
- Excess power consumption: 167 kW

If the excess power consumption rate increase is assumed to be linear then,

Rate of increasing cost of overhaul per month is computed as follows:

- Number of hours in one month: 30 × 24 = 720 h
- Number of hours of operation (27%): 720 × 0.27 = 194.4h
- Excess power units consumed in 1 month: 167 × 194.4 = 32464.8 kWh
- Cost of this excess power @ 10 c/kWh: 32464.8 × 0.1 = $3246.5

This is the cost of deterioration at the 24th month.

However, this is not the same for all of the 24 months. In fact, it must have been quite less in the initial stages. So, an assumption is made that the rate of deterioration is linear.

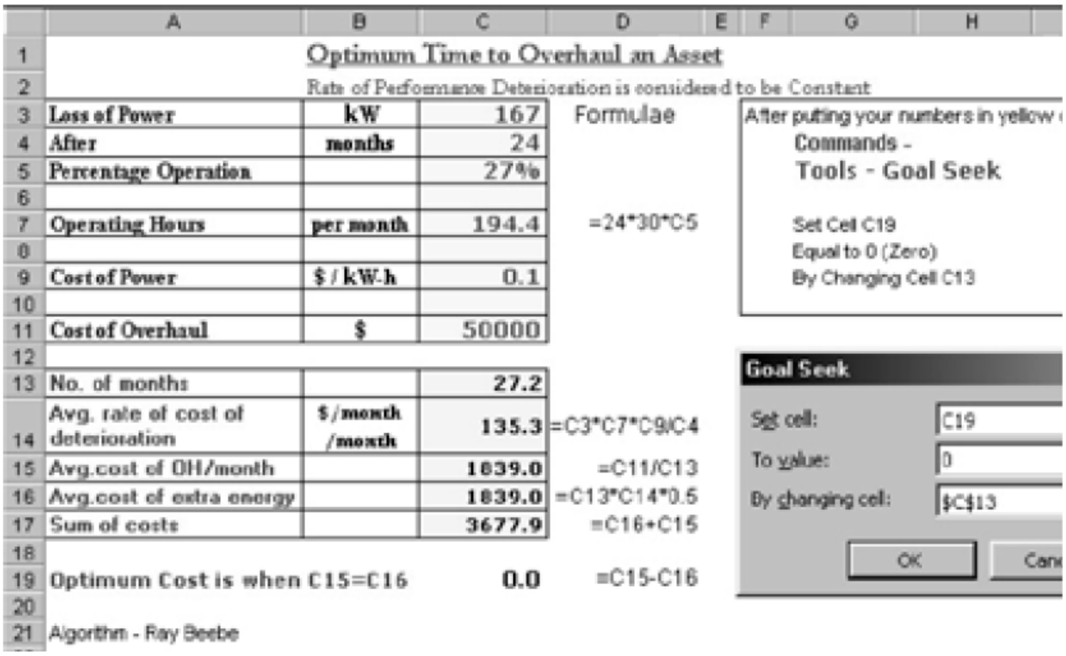
Therefore, the rate of cost of deterioration is spread over 24 months.

The cost of deterioration increases by $3246.6 per month/24 months

This average cost of deterioration is $135.3/month/month.

From here on the process is iterative but can be easily solved using a spreadsheet calculation

The procedure to do it in an MS-Excel spreadsheet is shown in Figure 7.30.



	A	B	C	D
1	Optimum Time to Overhaul an Asset			
2	Rate of Performance Deterioration is considered to be Constant			
3	Loss of Power	kW	167	Formulae
4	After	months	24	
5	Percentage Operation		27%	
6				
7	Operating Hours	per month	194.4	=24*30*C5
8				
9	Cost of Power	$ / kW-h	0.1	
10				
11	Cost of Overhaul	$	50000	
12				
13	No. of months		27.2	
14	Avg. rate of cost of deterioration	$/month /month	135.3	=C3*C7*C9/C4
15	Avg.cost of OH/month		1839.0	=C11/C13
16	Avg.cost of extra energy		1839.0	=C13*C14*0.5
17	Sum of costs		3677.9	=C16+C15
18				
19	Optimum Cost is when C15=C16		0.0	=C15-C16
20				
21	Algorithm - Ray Beebe			

Figure 7.30 – An MS-Excel worksheet calculation to work out the algorithm

The optimum time for an overhaul is when the average cost of overhaul per month is equal to average cost of excess power.

For the iterative process an optimum time is assumed, in this example, say we consider 22 months.

- The average cost of overhaul is: $50 000 ÷ 22 = $2273/month.
- The average cost of extra energy used is: $135 × 1/2 × 22 = $1488/month.

- The total average cost/month is therefore the sum of these two figures, or $3760.7.

Here we see the two costs are not equal and hence the total cost is not optimum.

This process is repeated till both the costs become equal.

This process is repeated and it is found that for this example the optimum duration is computed to be 27.2 months.

The total cost works out to $3677.9.
Note that this calculation is only correct if the wear progresses at a uniformly increasing rate with time. Information may not be available to make any other assumption, but decision makers have to start somewhere.

Note that some relatively small pumps may never justify overhaul on savings in energy use alone, but maybe justified on reduced plant production rate when the pump deterioration does not affect production, at least initially.

For a pump where the speed is varied to meet its desired duty, the effect of wear on power required is much more dramatic than for the case of a constant speed throttle controlled pump. This is because the power increases in proportion to the speed ratio cubed.

Unless the pump output is limited by the pump reaching its maximum speed, or by its driver reaching its highest allowable power output, then no production will be lost.

However, power consumed will increase more dramatically for a given wear state than for a constant speed pump. The performance monitoring tools can be used or portable instruments can accurately indicate the excess power being consumed.

The same method and calculations can be used as before to find the time for overhaul for minimum total cost.
This algorithm is quite general in nature and can be used not just for pumps alone.

Where there is deterioration in performance leading to loss of energy, it is a handy tool for maintenance engineers and managers in their role of managing assets to provide capacity for production.