

# HOW TO EXTEND THE LIFE OF YOUR BOILER FEEDPUMP

by

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## ABSTRACT

Six topics that should extend the life of a boiler feedpump will be described in detail. These topics will explore both operational and maintenance techniques that, when instituted, have been shown to extend the useful life of boiler feedpumps used in electric power generation. The operational techniques of:

- Properly prewarming a boiler feedpump before rotating the shaft,
- Proper rotor position and vibration monitoring, and
- The proper pump operating envelopes to reduce pump internal distress and wear will be discussed.

The maintenance techniques of:

- Disassembly, which includes the installation and removal dimensions that should be recorded,
- Proper inspection procedures, and
- Proper pump assembly will also be discussed.

In many cases, these techniques will not only extend pump useful life, but will reduce the cost of future pump repairs.

## OPERATIONAL TECHNIQUES

### *Boiler Feedpump Prewarming*

While many boiler feedpumps will not show any initial, visible distress from improper warming, other pumps will show symptoms, varying from difficulty or "hard spots" in rotating the shaft, to total "lockup" with the inability to turn the shaft. Whenever any of these symptoms occur, there is an interference between the rotor assembly and the stationary components. This interference usually occurs between the close clearance areas of the pump. Classically, it is between:

- The stationary case rings and impeller shroud seal diameter (Figure 1(a)),
- The stationary bushings and the impeller hub seal diameter (Figure 1(b)), or
- Between the stationary shaft end seal bushing and the rotating shaft end seal sleeve (Figure 1(c)).

When a boiler feedpump shaft is rotated or the pump is "started" and run under any of the conditions described above, a slight wear occurs in these close clearance areas that increases the clearance. The leakage associated with this increased clearance, therefore, increases. This wear and resulting increase in leakage can vary

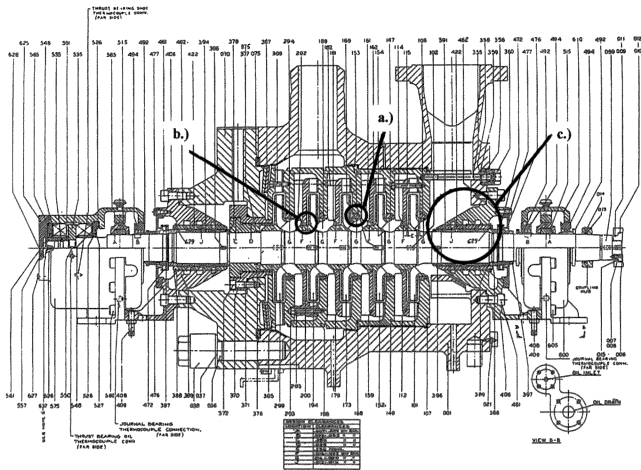


Figure 1. Cross Section View of a Boiler Feedpump (5BDXD520).

from being very slight, in the case of very light rubs, to being more significant, in the case of major rubs. Depending on the number of times a pump is started under these “improperly warmed” conditions and the severity of the condition, there is a corresponding impact on the useful pump life.

As internal wear occurs, internal clearances increase. This causes increased leakage around the impellers that reduces pump efficiency. Similarly, increased clearance of the shaft end seals causes increased seal leakage. In summary, the pump performance deteriorates until a pump refurbishment is required.

The distortion a pump encounters can vary depending on the temperature distribution throughout the pump. Most boiler feedpumps have shafts that are on a horizontal plane, as shown on Figure 1. Usually, a temperature stratification will occur with higher temperatures toward the top of the pump case, as shown on Figure 2. This is the result of the hotter water rising over the cooler water. As a result, the top of the pump case will expand more than the bottom of the pump case. Also, the rotor will take on similar bend. This distortion resembles a “banana” in cross-section. If there is an attempt to rotate the shaft under these conditions, there is a high likelihood that rubbing will occur between the rotating assembly and the stationary components at the small clearance locations described earlier.

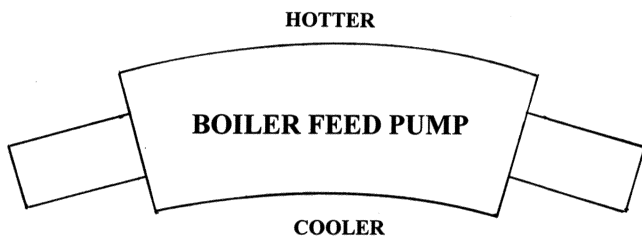


Figure 2. Graphic Representation of the Distortion of a Boiler Feedpump Subjected to Temperature Stratification. The Hotter Water Is at the Top of the Pump Case.

Another type of pump case distortion can occur if cold water is brought into the pump suction of a “hot” pump, or if hot water is brought into the pump suction of a “cold” pump. These conditions can cause a thermal shock to the pump. Since geometrically smaller parts cool faster (shrink faster) when subjected to a cool environment, or will heat faster (expand faster) when subjected to a hot environment, interference can occur at small clearance locations.

There are various pump case locations, based on pump manufacturer, to introduce “warming” water to the pump, as well

as various circulation paths used to circulate the “warming” water through the pump. However, the end result of any of the warming methods is to have a uniform temperature throughout the pump with no thermal stratification. Some warming instructions will indicate a time length for warming water to circulate through the pump. Others will require that a certain indicator temperature be within so many degrees of the inlet temperature, or the suction flow that the pump will experience when placed in service. However, the desired end result of any warming procedure is to have a uniform temperature throughout the pump, and that this temperature be at the temperature of the suction flow that the pump will experience when put into service. Because of the individuality of each installation, the warming instructions may not be sufficient, particularly for “older” pumps.

Since temperature indicators are relatively inexpensive as compared with “pump overhauls,” it is suggested to add temperature indicators on the top and bottom of the case at each end and at the middle of the pump, as shown on Figure 3. This would be six temperature readings. Then, a criterion must be developed based on these six temperature readings, as to when it is safe to turn the shaft and run the pump. This criterion should indicate that these temperature readings should be within so many degrees of each other and within so many degrees of the pump inlet flow. The lower the differences between these temperature readings, the better. However, in reality these temperature differentials will not be zero. One example of the criterion for these temperature readings for a permissive pump startup is to have the case temperature readings within 35° of each other and within 35° of the inlet pump flow temperature. (This value can be 50° in some extreme situations. However, the value should be kept as low as possible.)

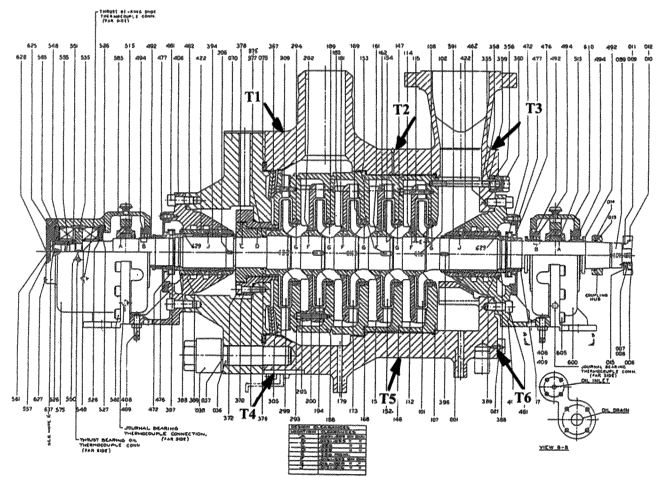


Figure 3. Cross Section View of a Boiler Feedpump (5BDXD520), Showing Six Locations: T1, T2, T3, T4, T5, T6 for Case Temperature Indicators for Determining a Warmed Pump).

**Rotor Position and Vibration Monitoring**

*Axial Position Monitoring*

The monitoring of axial rotor position is another way to extend pump life. The use of axial rotor position monitoring is an excellent method of determining stable operating areas of boiler feedpumps. This is particularly important for boiler feedpumps equipped with a hydraulic balance device containing a radial variable orifice. Consider the example of an “inline” pump having all the impellers facing in the same direction as shown in Figure 1.

The boiler feedpump, shown in Figure 1, is equipped with a balance device containing a variable radial orifice. As the pump is run up to running speed, the hydraulic pressure distribution around each pump stage results in a substantial axial force toward the suction

end of the pump. This axial force is balanced by the hydraulic axial balancing device located after the last stage of the pump. This balancing device contains a variable radial orifice that "self compensates" for the particular axial force at a particular operating point. As an axial force component would slightly increase or decrease due to a slight change in the pump operating point, the proximity of the rotating radial face relative to the stationary radial face changes slightly, i.e., "variable orifice." This change of the relative position of one face with respect to the other, changes the pressure distribution around the rotating portion of the balance device, which provides for variable axial balance of the rotor. Figure 4 shows how axial force varies with the radial face gap or separation.

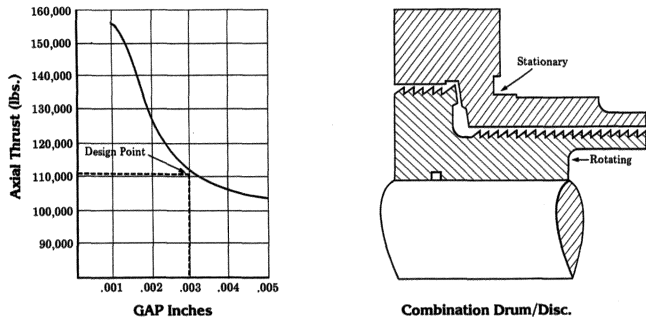


Figure 4. Axial Force as a Function of Radial Face Gap (or Separation) for a Balance Device Containing a Variable Radial Orifice.

The boiler feedpump, shown in Figure 1, is also equipped with a tilting pad thrust bearing. This thrust bearing is spring loaded in such a way as to separate the two radial faces of the balance device at standstill and at low speeds. In this way, there is no possibility for the two faces to contact each other at initial startup of the pump or have a boundary lubrication situation that may cause accelerated face wear.

Considering the above as background, it can be seen how axial position monitoring will extend the life of a boiler feedpump. Figure 5 shows an example of axial position probe installation. It is important that two axial proximity probes are used and positioned at 180 degrees from each other. Their electrical outputs should be biased against each other to eliminate any effects of "lack of perpendicularity" of the target surface.

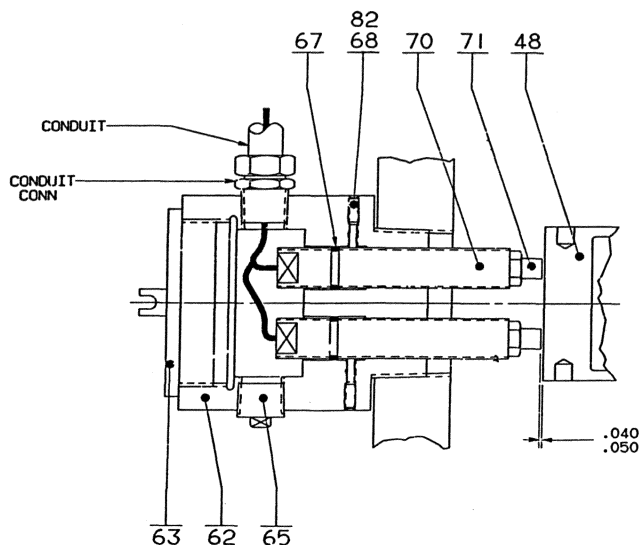


Figure 5. An Example of a Dual Axial Probe Installation. Item 48 Is an Enclosed Check Nut That Mounts on the End of the Shaft, and Item 71 Is the Proximity Probe.

Axial position probes will provide an "online" monitoring of axial rotor position. Although axial balancing devices are designed so that no contact should occur between the rotating and stationary surfaces, contact may occur under some upset conditions. By tracking the axial position readings, the amount of wear occurring to the radial faces becomes a known value. Therefore, knowing the amount of wear as a function of time, and the amount of wear, the remaining balance device life can be predicted. If axial position is tracked with respect to "operating events," the events that have the greatest effect on wear will be determined. By reducing the frequency of these events, the pump life can be extended.

Another observation that can be made from axial probe readings is the operating point where the balance faces come in close proximity to each other. This can be a point that the rotor just becomes axially balanced (floats axially) and can be unstable to the extent that the radial faces could contact and cause wear. With the implementation of axial probes, the exact point that the axial balance device starts to balance axial thrust can be determined. These operating points should be avoided because of the potential for contact between the faces and the associated wear.

Axial probes can also be a valuable asset on pumps with balancing devices without a variable radial orifice. These types of balancing devices are designed for a specific operating point or design point. For other pump operating points, the difference in pump hydraulic axial force and the compensating balance device axial force, a resultant residual force, are compensated for by the thrust bearing. These types of pumps usually have larger thrust bearings because of the magnitude of these residual forces. The direction of this residual force will determine the direction that the thrust bearing is loaded. When axial probes are installed on these types of pumps, this loading direction is easily observed. Again, the operating points that exhibit this change in direction should be avoided, because they tend to be unstable and can cause accelerated distress of the thrust bearing. The axial probes will also indicate the amount of wear occurring and provide online thrust bearing clearance.

Radial Vibration Monitoring

In general, radial proximity probes are the recommended method of radial pump vibration monitoring. Information from a properly instrumented pump can also be used as a diagnostic tool in determining possible root causes of vibration.

It is preferred to have two radial proximity probes, 90 degrees to each other, near each radial bearing. This probe configuration, in conjunction with a key phaser, will not only provide the magnitude of shaft radial displacement relative to the bearing bracket, but also provide the information to determine shaft orbit shapes.

There has been a great deal of information published about radial vibration monitoring. Although the magnitude of vibration is very important, it must be remembered that the frequency of vibration is just as important in determining a root cause for the vibration. Radial vibration monitoring will not automatically increase pump life. However, the use of this radial vibration information can also indicate a pending problem that may be extraneous to the pump. In these situations, the pump can be shut down in a controlled manner to correct the problem. The pump can then be restarted, avoiding a more costly pump repair in the future.

Consider the following example of this type of pump vibration problem. A boiler feedpump started experiencing high vibration at the coupling end bearing location. The frequency of vibration was determined to be at running speed. The phase angle of the peak amplitude changed each time the pump was shut down and then restarted. This pump had a grease packed, gear tooth type coupling (with a spacer between the coupling hubs) that had been in service many, many years without being maintained. It was hypothesized that the coupling had extensive tooth wear that permitted the spacer to take a different radial position each time the pump was shut down and restarted. This hypothesis was verified when the

coupling was inspected. The coupling was replaced and when the pump was restarted, the vibration problem was gone. If this vibration problem was not addressed, the vibration would probably continue to grow, and eventually cause internal pump wear and possibly trip the pump. If this were the case, a pump repair would probably have been required.

### PUMP OPERATING ENVELOPES TO REDUCE PUMP INTERNAL DISTRESS AND WEAR

This section of the Tutorial will address operating envelopes of boiler feedpumps that are the least stressful on the pump, with respect to pump wear or damage. Pumps are fixed geometry machines that are primarily designed to provide design point conditions, rather than off-design conditions. At design point conditions, the flow fields are designed to enter the impeller and diffusers or volutes without any mismatch of flow and vane angles. Also, the flow field uniformly fills the flow passages. This is exemplified in Figure 6.

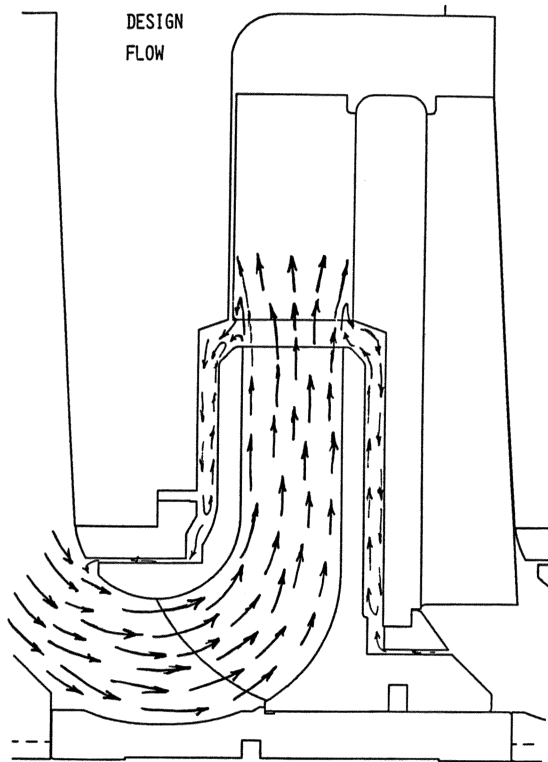


Figure 6. Meridional View of an Impeller Flow Field at Design Flow.

As the flow is reduced at constant speed, there is a point where the flow field within the flow passages can no longer remain organized and orderly. Flow angles no longer match the fixed geometry vane angles and contours. Flows begin to separate from boundary surfaces, and at lower throughflows they begin to reverse, as exemplified in Figure 7. These recirculating flow fields are not uniformly recirculating. Instead, they can be going in one direction between two vanes and in an opposite direction between two other vanes. Also, these "flow cells" that are not uniformly distributed circumferentially, can get swept downstream in a somewhat random fashion. Under most circumstances, these recirculating flows tend to be violent in nature, resulting in pressure pulsation and vibration in the lower frequency range. (Although recirculation occurs in all centrifugal pumps at low flows, the onset can be minimized with careful hydraulic design and pump selection.)

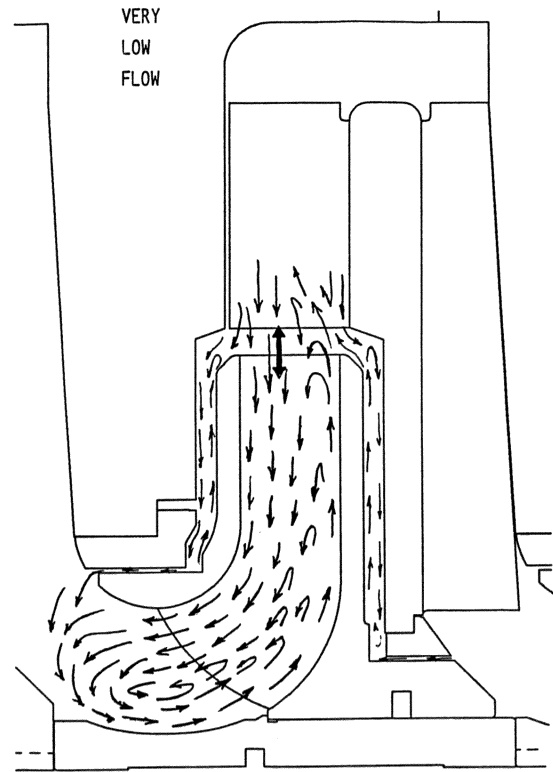


Figure 7. Meridional View of an Impeller Flow Field at Very Low Flow, Showing Suction Recirculation and Discharge Recirculation.

In summary, recirculation, sometimes referred to as hydraulic stall, is the result of a disorganization of the internal flow field, resulting from reduced throughflow. If this disorganized flow exits out the impeller eye near the shroud and reenters near the hub, it is called suction recirculation. If this phenomenon occurs near the impeller discharge area, it is called discharge recirculation. In many cases, suction recirculation and discharge recirculation are interconnected. Both of these phenomena can be violent at their onset and cause major pressure pulsation and pump vibration. (In many installations, the existing pump instrumentation will show a step increase in pulsation or vibration, which is an indication of suction or discharge recirculation.) This results in distress to the pump and can cause accelerated wear to the close clearance components of the pump, which will result in more frequent pump repairs.

Since pump distress occurs with recirculating flow fields (more violent at their onset), and since action can be taken not to operate the pump at these conditions, it is reasonable to expect longer pump life for a pump not subjected to these adverse conditions. It is very possible that the point at which the onset of suction or discharge recirculation begins is at a flow that is greater than the published minimum flow. In these situations, the pump throughflow should be kept at values that are higher than those for the onset of suction and discharge recirculation. This means that a modulating valve in the pump recirculation pipe should be opened earlier than normal to prevent distress to the pump.

Therefore, the pump operating envelope would require modification to avoid the flow regimes where suction and discharge recirculation occurs. Although this translates into a smaller range of operation for the pump, it usually has minimal impact on overall operation. The increase in pump life and increase in pump reliability usually justifies the impact of reduced operating range.

### MAINTENANCE TECHNIQUES

A list of any operation and maintenance issues known during the last run interval should be maintained and reviewed. Also, review

the pump's long-term history to understand the timing of repeat activities such as seal or bearing replacement. Effectively addressing root-cause issues leads to longer runs. A life span of two, three, or four years, as historically experienced, may seem to be all that is possible. However, it is likely that the life span of the main item(s) responsible for the current accepted overhaul frequency can be significantly improved. In the late 1970s, an electric utility set a goal to extend the run life of high-energy boiler feedpumps to five years. They not only realized but exceeded that goal. Seven-to-nine-year run intervals are now experienced.

Each overhaul is an opportunity for original equipment manufacturers (OEM) and non-OEM resources to make recommendations concerning improvements in materials, clearances, part configurations, operating parameters, etc., which will improve the durability of the pump, therefore, extending the run intervals between overhauls.

#### *Disassemble Pump*

The only teardown measurements of much practical value would be a thrust bearing travel check and a shaft end-to-end measurement. The thrust bearing check is done to get first impression of conditions that may require new thrust parts. The shaft end-to-end measurement is an as-found reference and is used to anticipate an axial move activity. Checking shaft end-to-end distance, a thermal and coupling freedom consideration, should be a reassembly step after running position is set. Correction of this requirement will take place early in the coupling alignment process. If desired, float without thrust bearing could be taken on pumps with balance disk to determine disk wear, therefore as-found running position. Element lift checks, if made at both ends simultaneously and indicators as close to element as possible, can provide an indication of wear if information is needed sooner than shop inspection results will become available.

All parts should be examined during disassembly and any evidence of rotating to stationary contact noted. Keep track of the specific location and orientation of this evidence to collaborate with the element components. These observations may tie together later and help prove the existence of a problem. Such evidence will be necessary when convincing all involved parties to allocate time and money to correct this problem if it is mechanical, or change procedures if it is operational.

During disassembly work, careful attention should be paid to protect the shaft from damage. Bearing journal and vibration-probe-sensing band areas are especially important. Note condition of all positioning dowels and holes. The dowel pins are usually tossed into the bolt bucket with everything else; these really should be wrapped to protect from impact damage and stored to avoid getting bent. Utilize all tools (fixtures, skid plates, shaft extenders, lift rigging, etc.) to extract the element from the case in a controlled fashion.

If inspected spare bearings are not on hand, the babbitt bearing bores should be measured, when available, so spin casting repairs could get started, if necessary.

The element should be placed in a shipping skid, supporting element and shaft. Use a clamping device to prevent the shaft from sliding or bouncing in transit. The driver should not be allowed to tie down across the shaft with a chain or strap when securing the load to a truck.

#### *Case Areas to Inspect*

- Low pressure sealing face for any washing or cutting across the surface
- Discharge area for erosion
- Head gasket seal face for concentric grooving, about 250 or less microinches surface roughness, and no wash or cut areas
- Inside surfaces of case for cracking near external attachments or major changes of wall cross section or thickness

- Seal housing for erosion
- Case radial fit diameters and compare with radial fit areas of the element
- Head to case radial fit diameter
- Inboard seal housing to inboard end of pump case register fit diameter

#### *Overhaul of Element*

- Look for handling or shipping damage
- Prepare to unstack element using vertical pit
- Establish indicator zero with the rotor down
- Record float readings as each potential-float-limiting part is removed
- Record adequate dimensional data to evaluate rotor axial position at optimum setting for each stage individually. This has varying degrees of difficulty depending on pump design.

Determine axial corrections needed from disassembly data. In addition, measure each component involved (at the following places) for comparison of like parts and also to choose easiest or preferred place for correction.

- Impeller hub shoulder to center of discharge vane
- Split keeper ring thickness or snap locating ring shoulder depth
- Shaft impeller locating groove to groove distances

It has to be decided which components you wish to be kept consistent, and which ones are acceptable for alteration. Axially centering can be upset by any new parts of rotor assembly introduced during the life span of the pump, or by reused parts being positioned in different locations during overhauls.

#### *Head Areas to Inspect*

- Diameter of register fit to case
- Gasket surface condition for concentric grooving, about 250 or less microinches and no leakage cuts or handling dents across gasket width
- Sweep check in machine to check concentricity of head register to bore and head register to seal housing register fit area
- Diameter of register fit to seal housing
- Stationary balance bushing bore perpendicular to head gasket surface
- Seal housing mounting flange parallel to head gasket surface
- Stationary balance bushing installed—then checked for bore perpendicular to head gasket surface, swept axially for any possible high spot on wall, and skim cut to eliminate any minor problems

#### *Stationary Balance Bushing*

- Inspect for rubbing
- Check material hardness

#### *Seal Housing Inspection*

- Inboard housing register fit diameter to inboard case end
- Outboard housing register fit diameter to head
- Bore size for seal bushing
- Bore to mounting flange perpendicular

#### *Seals*

If still using floating ring design and having problems or if you have depleted parts inventory supporting floating rings, consider

changing to fixed bushing (labyrinth) design. A seal water injection system with separate temperature controls for each end is highly recommended; however, some locations that have switched to fixed bushing design have stayed with pressure control seal water system with success. During installation of spiral grooved rotating sleeves, do not trust markings alone. The correct orientation of the spiral groove will result in water being pulled back into the pump with rotation. Examine rotating and stationary parts for rubbing. Also inspect dimensions of the following:

- Seal housing bore diameter
- Stationary seal bushing outside diameter for proper fit in seal housing
- Stationary seal bushing inside diameter for running clearance
- Rotating seal sleeve outside diameter for running clearance
- Rotating seal sleeve inside diameter for proper fit to shaft

#### Shaft Inspection

- Nondestructive evaluation (NDE), especially at threaded areas and keyways
- Surface condition of journals
- Diameters at:
  - Journals
  - Impeller fit areas
  - Thrust collar
  - Seal sleeve locations
- Runout
- Signs of rubbing
- Known locations of bands for proximeter-type vibration probes

#### Diffusers

- Examine condition of inlet vane tips (cracking, breakage, erosion)
- Measure diffuser channel width,  $b_3$
- Measure diffuser diameter,  $D_3$
- Measure outermost diameter for fit in diaphragm or case inner diameter

#### Volutes

- Examine condition of volute tongue (cracking, breakage, erosion)
- Diameter of volute circle

#### Impellers

- Measure and inspect condition of bores
- Measure discharge channel width,  $b_2$
- Measure diameter of impeller vane discharge tip circle,  $D_2$
- Inspect hidden side of inlet vanes for cavitation
- Interference fit to shaft
- Sidewall thickness (overlap issues)
- Discharge vane areas—cracking in corners
- Runouts—radial and sidewall axial
- Signs of rubbing
- Hardness of hub outer diameter
- Hardness of eye outer diameter

#### Spacer Sleeves (Impeller Loose Fit Designs)

- Hardness
- Ends perpendicular to bore
- Bore size for fit to shaft

#### Wear Rings and Bushings

- Absolute hardness, max hard equals brittle
- Locking mechanisms sound, grubscrews or tack welds
- Wear ring clearances—dampening and efficiency
- Bushing clearance—dampening
- Differential hardness of 10 Rockwell C scale to impeller
- NDE
- Grooved surfaces
- Signs of rubbing
- 420F stainless steel

#### Radial Bearings

- Babbitt bond good
- Measured bore size acceptable
- Bearing shell dowels close tolerance for upper and lower half alignment
- Tilt pad rockers not flattened

#### Thrust Bearing

- Babbitt bond good
- Pad support buttons not flattened
- Leveling assembly rockers not worn between contact with each other

#### Gap B

Gap B relates to the distance between the exit of the impeller vane tip and the inlet vane tip of the diffuser or volute. It is expressed as a percentage (Figures 8 and 9 show details).

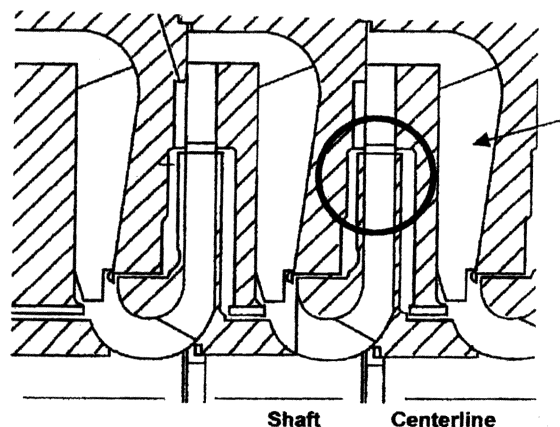


Figure 8. Cross Section of Multistage Centrifugal Pump.

If too tight, unnecessary shock forces are created that add to the stress of all neighboring materials. The pressure pulsations cause accumulated damage and show up as high vibration at vane pass frequency. Cracking of impeller and diffuser vanes will develop. Piping systems can also suffer damage. Addressing this issue can take a major life-robbing piece out of the premature failure equation.

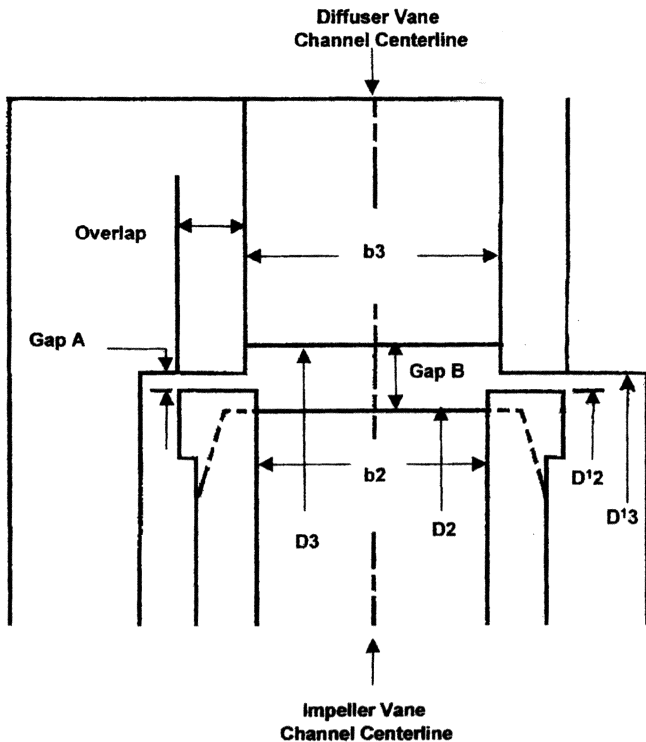


Figure 9. Important Parameters to Note on Each Stage.

#### Gap A and Overlap

Gap A relates to the distance between the impeller sidewall diameter and the stationary rim of the diffuser. It is a radial gap and is expressed in inches. In some volute designs, there is not any case material in close proximity to the impeller to easily visualize this Gap A area.

If too large, high vibration will occur at lower than synchronous frequency. Problems also will be seen as low-flow axial shuttling. Thrust bearing damage will develop. Internal part damage occurs due to axial loading reversals.

Proper Gap A and overlap is an orifice that filters/restricts the occurrence of pressure fluctuations between impeller sidewall and case sidewall. Troublesome axial issues are not as likely to develop (Figures 8 and 9 show details). The continuous usable flow range is expanded. There is research concluding that Gap A is of the most benefit to single stage double suction pumps.

#### Device Balance

The rotating drum type is a cylinder running with close radial clearances. Flow across this area experiences a large pressure drop that counteracts most of the hydraulic axial forces within the pump.

The balance disk type has a drum section and a vertical face, which is essentially a water lubricated thrust bearing. The taper face configuration has proven to be more reliable than straight face.

#### Component Balance

The objective is to address all large imbalances at the point of origin, instead of correcting assembly only at end planes or mid and end planes.

#### Assembly Balance

The residual imbalance level should be brought down to  $4W/N$  (or  $W/N$  if preferred).  $W$  is rotor weight and  $N$  is speed in rpm. The units work out to be in ounce-inches and the result is total for rotor. If three balance planes were available, you would divide the result by three for the tolerance to use at each plane.

#### Stack Element

The use of a vertical stacking pit is optimum. All pieces must be deburred and cleaned. A dry moly spray lubricant should be used on the clearance running areas to help the stainless steel materials survive handling, transportation, and installation. Be careful of antiseize products to which dirt can adhere, because grit can compromise verification checks while stacking or during installation, especially centering the element later in the field.

Each stage is checked for proper impeller-to-diffuser-centerline alignment as well as float, and these data are recorded to evaluate final compromise running position, where all stages are within tolerance.

The element is clamped end-to-end and laid on side for final checks. The radial freedom lift check verifies proper bore line through stationaries. We expect to see approximately diametrical clearance at each end—both ends lifted together and indicators as close as possible to the element. The element is checked at starting position, then assembly is rolled 90 degrees and the test is repeated. The shaft is also turned while centered to feel for any dragging. Total float is recorded.

The completed element needs to be protected from debris as soon as practical and stay protected until time of installation.

Transport only with stable shipping skid support and the shaft restrained from any chance of axial travel or vertical bouncing.

#### Installing Element in Case

The float of the element can be checked right out of the box. Try to take most of the rotor weight off during check. We have utilized previously mentioned data to establish:

- The element radial diameters are compatible with case
- Head fit to case within tolerance
- Inboard seal housing to case inboard end fit
- Outboard seal housing to head fit
- Seal running clearances
- Bearing running clearances based on micrometer readings
- Thrust bearing components in useable condition
- Balance device radial running clearance
- Shaft end-to-end distance, as-found and installation requirement from outline drawing, including tolerance
- Optimum running position from shop report
- Differential hardness of wear ring and bushing material to impeller eye and hub areas, respectively

A critical prerequisite to assembly is having the pump case at uniform temperature. If a thorough isolation cannot be accomplished, the unit may have to be taken offline. The heat conducted through the suction and discharge lines into the case can be enough to seriously jeopardize element centering activities, due to distortion. Centering and turning the element at one thermal condition does not assure in-service success. The pump may even lock up on turning gear. Hot water leaks make an accurate reassembly nearly impossible. Further verifications include:

- Gaskets with proper rating for the required temperature and pressure service,
- Low-pressure seal face surfaces are clean, and
- Antiseize product is applied wherever the pump element will drag in the case.

The major assembly steps can proceed by:

- Installing the element
- Rigging the head to hang plumb
- Pulling the head up evenly to push the element squarely
- Checking the float (lift with a sling to take off some weight)

Pacific, Ingersoll Rand, Byron Jackson, Sulzer Bingham, and Worthington have similar designs in regard to the fact that the outboard end of the element is centered by the head. The inboard end of the element is centered by the case, essentially for all manufacturers.

The bearing housings are mounted and the shaft is supported in bearings. Center the shaft on one end, then the opposite end. Then repeat centering the first end, and repeat the opposite end. Inspect dowel holes for any step at joint line, ream as necessary—pins must be in very good condition. These pins and holes are the mechanism to accurately repeat position of the bearing housing. A pump element does not have any internal seals that are thin rows of teeth like in a steam turbine. All close-running clearances are substantial cross sections of material, so a pump is not forgiving on centering errors. Small radial centering mistakes result in minor rubs that do “rub in,” resulting in minor damage to new parts, and cause higher leakage rates due to the large gap opposite the rub. Larger mistakes cause heavy rubs that will affect efficiency rapidly after startup and may be noticed in overall vibration levels. Gross mistakes may not permit the element to survive very long after startup, components may get hot enough to expand to the point of galling and seizure, or they may crack into pieces that cause any number of secondary failure damage scenarios. Any reassembly variable that falls outside an acceptable tolerance range ultimately results in a less efficient as well as a shorter run interval.

Now is a good time to verify bearing clearance, if you prefer using the lift check method. The only components in place have larger clearances than the bearing. Be sure to leave out oil seals, flinger rings, or anything else that could cause a false result. Also, because bearing running clearance and bearing to housing “pinch” checks are related, both tests should be accomplished. Bearings need a tight fit within housing, from about on size to two mils interference. Meeting this requirement, the bearing running clearance should not be compromised; otherwise, the pinch is too much and should be corrected and rechecked. One common cause of too much pinch is not bearing outside diameter to housing bore problems, but actually interference between the antirotation pin and the hole in housing. If plastigauge is to be used for final bearing running clearance check, both these steps can be done later.

The bearing housings are removed for seal housing installation and then reinstalled. The thrust bearing maintains the running position (impellers axially centered in diffusers at optimum compromise of all stages of the element). Shimming at the thrust collar is done to set the element at the targeted running position.

If not already done, bearing running clearance and pinch checks can be done now using plastigauge. Use of plastigauge in the journal clearance area is using the product as intended to determine the clearance present. A correct pinch check actually overcrushes the smallest range of plastigauge wider than the minimum gauge width on the cover envelope. If too much clearance (1.5 mils or more) is detected, a thinner joint gasket should be used; a shim should be placed on top of bearing, or the bearing housing can be repaired by machining. Remember that joint sealant compounds add to gasket thickness and decrease pinch as compared with no sealant assembly checks.

The thrust bearing clearance is established by using shims or gasket material with the end cover. Sometimes light skim cuts are taken on the bolt circle surface of the end cover to close excess clearance.

On Delaval designs, both ends of the element and the head register to the barrel. After the head is on and the float of the element is checked, the balance disk can be installed. It is hydraulically pumped onto the shaft and the float is checked to achieve targeted running position. If the disk has to be pushed past being flush with shaft shoulder, ensure that the pusher tool has been relief cut to clear shaft shoulder. A word of caution: awareness of starting standoff and amount of travel of the disk is critical. Too little interference fit and the disk may not be rigid

enough to withstand loading without fretting the shaft. Too much interference fit and the disk could crack (depending on the material properties) during installation or later in service.

The bearing housing mounting flange is part of the seal housing. The bearings are then moved to center the shaft at each end within the seals. Repeat the centering activities at each end. Plastigauge should be used to verify bearing-to-housing pinch checks and bearing running clearances to be simultaneously correct, as described previously. It takes a lot more judgement to get meaningful results using plastigauge on tilt-pads than straight cylindrical bearings for running clearance checks. Checking for running clearances by using a journal-sized mandrel and performing lift checks at a workbench works well. The actual shaft can also be used before or after the element is in case. As with the mandrel bench check, the shaft position is held and the bearing housing is lifted. Remember that with a five-pad bearing, support is at six o'clock. However when lifted, the journal is going between pads at 12 o'clock. Lift-check results will be larger than actual pad-to-journal clearance in line with support rockers.

The thrust bearing clearance is established by correct thickness backing plates. The spring supported inboard set of leveling plates is to allow travel of rotor assembly during startup and shutdown that accomplishes balance disk face separation.

## CONCLUSION

The main steps in obtaining the maximum life of a boiler feedpump are understanding its design, respecting the startup requirements, and staying within its operating parameters. Once this is in place, the life can be extended and maintenance costs reduced by incorporating proven component modifications as required at overhaul opportunities. Longer runs allow manpower and maintenance dollars, previously dedicated to the boiler feedpump, to be used elsewhere.

## NOMENCLATURE

$D_2$	=	Outside diameter of impeller discharge vane tip circle
$D_3$	=	Inside diameter of diffuser inlet vane circle
$b_2$	=	Impeller discharge channel width
$b_3$	=	Diffuser inlet channel width
$D_2^{1_2}$	=	Outside diameter of impeller sidewall
$D_3^{1_3}$	=	Inside diameter of diffuser/volute sidewall
Gap A	=	$\frac{D_3^{1_3} - D_2^{1_2}}{2}$ , generally .050 + inches
X	=	Overlap ratio = $\frac{\text{overlap length}}{\text{Gap A}}$ , generally 3 to 5
Gap B	=	$\frac{D_3 - D_2}{D_2} \times 100$ , depends upon impeller and diffuser/volute vane combination
		Diffusers, generally 4% minimum, 6% preferred
		Volutes, generally 6% minimum, 10% preferred
		Constraints to increasing Gap B are impeller diameter requirements and the length to diameter ratio of the diffuser
$b_3/b_2$	=	Hydraulic channel width ratio, generally not less than 1.15

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