MONITORING REPAIRS TO YOUR PUMPS

by

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Mr. Nelson retired from Amoco Oil Company, Texas City, Texas, after more than 36 years of service. Responsibilities in his last position included refinery instrument and electrical repair, weld shop functions, operation of mobile equipment such as large cranes, etc., rotating machinery maintenance and repair, and craft training.

He received a B.S. in Mechanical Engineering from Texas A&M University and is a registered Professional Engineer in the State of Texas. In 1992, he was honored as a "Distinguished Engineering Alumni" by the College of Engineering and as a member of the "Academy of Distinguished Former Students" by the Mechanical Engineering Department at Texas A&M University. He recently had a test cell named in his honor in the Turbomachinery Laboratory research facility at Texas A&M.

ABSTRACT

Many centrifugal pumps are doomed to premature failure because the total pump installation was not evaluated or the repair activity coordinated to correct the basic cause of the initial failure. Pump users have a great need for guidance in repair techniques as pump maintenance is the most expensive aspect of turbomachinery utilization. Because of the hazardous nature of the hydrocarbon processing industry, government regulations require repair facilities to maintain detailed instruction manuals, according to manufacturer, and style of pump to guide the repairs. Some owner companies are also developing guidelines to ensure that quality repairs are achieved. Some basic guidelines to be followed in making repairs are discussed. This presentation intended to serve as an aid in developing detailed guidelines.

INTRODUCTION

Pumps encountered during repair activities are from many different manufacturers and built to several standards of different vintages. The two most common pump standards encountered are:

API

American Petroleum Institute (API) pumps [1] are designed expressly to hand hydrocarbon liquids. They will normally have carbon steel pressure containing parts, closed impellers keyed to the shaft and confined casing gaskets. While guidelines and practices will vary slightly from owner company, these pumps will normally be found in services with pumping temperatures above 350°F and/or working pressures above 350 psi. Most of the discussion is directed toward API type pumps.

Until 1989 (Seventh Edition), API 610 standard "Centrifugal Pumps For General Refinery Service" was written around single stage overhung or back pull out horizontal pumps in hydrocarbon service with 6.0 in and smaller nozzles. The standard now recognizes that there are at least nine types of centrifugal pumps in hydrocarbon service.

ANSI

American National Standard Institute (ANSI) pumps are designed to handle chemical process liquids. Built to ANSI Standard B73.1, 2 or 3, they will normally have ductile iron or stainless steel pressure containing parts, compressed casing gaskets and open impellers screwed to the shaft. Generally, they are applied with operating temperatures below about 350°F and working pressures below about 300 psi.

PUMP DESIGN INTRODUCES PROBLEMS

Some of the problems encountered in pump repair involve the design of the pump. A radially split pump as shown in Figure 1 departs from the design practices of the axially split pump.

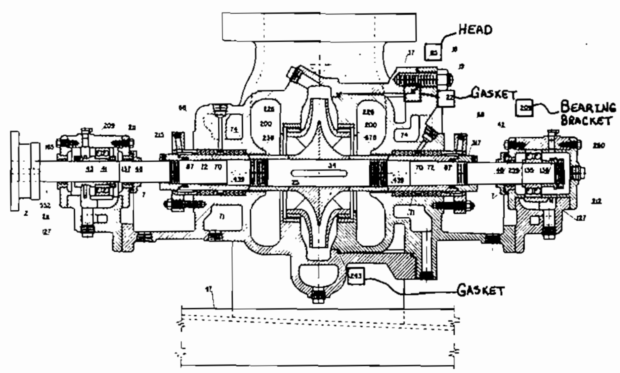


Figure 1. Radially Split API Design Pump.

API Standard 610 [1] requires radially split pump casings under the following conditions:

- When pumping temperatures are above 400°F (a lower limit should be used if thermal shock is probable).
- When pumpage is flammable or toxic with specific gravities less than 0.07.
- When pumpage is flammable or toxic at rated discharge pressures greater than 1000 psig.

Radially split pumps use metallic spiral-wound confined gaskets (Figure 2) between casing sections, whereas the gasket on an axially split joint is not confined. Thus, the probability of a proper and continuous seal is greater with a radially split casing than with

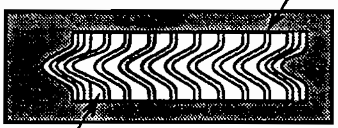


Figure 2. Cross Section of Spiral-Wound Pump Head Gasket.

an axially split design. In addition, the conversion of the filler in spiral-wound gasket to nonasbestos materials has had less impact on the performance of the gasket than the compressed sheet gasket material.

RADIALLY SPLIT PUMP PROBLEMS

A number of problems are inherent with a radially split case as shown in Figure 1 that can be troublesome if not recognized properly.

- *Bearing Alignment*—In radially split designs the bearing bracket(s) is (are) bolted to the casing or heads. Both the bearing bracket *and* the head (or heads) are removed during disassembly, thus requiring all internal alignment to be reestablished each time maintenance work is performed. This is a time consuming and exacting procedure that is not spelled out well in maintenance manuals.

- *Inner Case Alignment*—The liquid path of a radially split pump consists of inner case guides that is contained within the pressure walls of the outer casing. Alignment of this inner casing is very poor. In some single stage designs the guide is cast into the head. In other designs, it is a separate piece.

- *Internal Leakage*—The high pressure (discharge) and low pressure (suction) compartments of both single and multistage pumps are separated by a single invisible gasket. Because the inlet guides or the bundle does not fit tightly in the case bore, the gasket may not function properly, permitting internal leakage. In addition, the gasket is frequently damaged as it passes across the suction nozzle opening in the case, a condition that is hard to avoid and equally hard to detect if it occurs. It is also hard to compress both the external and the internal gaskets the proper amount by tightening the head.

- *Thermal Distortion of Case*—Heat distortion imposes a severe strain on a pump shaft and bearings and usually results in permanent damage. Uneven heat expansion of the pump case is the most serious cause of mechanical failures in hot service pumps.

- *Case Distortion Due to Pipe Strain*—Since the pump case carries the bearing housings, any distortion of the outer casing due to excessive pipe strain is reflected in the location of the rotating element center line as established by the bearings.

- *Misuse of Dowel Pins*—Internal alignment problems can arise after a severe failure of the pump bearings. Frequently, replacement shafts and the bearing bracket bore clearances are built up with weld metal and machined. Proper running clearances are established for the shaft, bearings, and deflector in the shop. If the old dowels were used for final positioning of the bearing housing, misalignment of the bearing housing to the shaft can be created. The result is a loss of internal clearances and bearing failures.

- *Vibration*—is caused by the unbalance imposed on a rotating pump rotor by a shift in mass center of loose impellers. Loosely fitting nonrotating diffuser sections and rotating impellers can generate vibrational waves that travel freely through a pump, its piping and foundation.

PROCESS PUMP GASKETS

Radially Split Casings

Spiral-wound pump head gaskets consist of V-shaped preformed plies of metal, wound up in a spiral with a soft separation of nonmetallic fiber (Figure 2). The V-shape gives spring like characteristics as the metal and soft filler plies flow into gasket surface finish irregularities to give sealing action. The inner and outer metal-to-metal plies must be under equal compression. The compressibility of a spiral-wound gasket is controlled for a specific bolt loading of 30,000 or 45,00 psi by varying the number of metal-fiber wraps. The standard gasket is good for temperatures up to 750°F and should not be reused. Most pump head gaskets are 0.175 in thick and should be compressed to 0.130 ± 0.005 in.

Since the ID of a spiral-wound gasket decreases during compression, there must be clearance of 1/16 in on the ID to permit equal compression of all plies. The OD can fit in the head recess more snugly, but a tight fit cannot be allowed.

Axially Split Casings

With the banning of asbestos as a gasket material, some serious problems have arisen because no substitute for asbestos has demonstrated total superiority to date. The major deficiency in nonasbestos gaskets is long term higher temperature stress-relaxation. This tendency means frequent bolt tightening and the risk of gasket blowout.

A wide variation exists in the quality and ruggedness of available axially split casing joints. The normal practice for these pumps is to machine the bore with a 0.030 in shim between the halves of the casing. The shim is discarded after boring. A 1/32 in compressed sheet gasket is used between the flanges. This material has a thickness of 0.030 to 0.038 in as manufactured with a compressed thickness of 0.025 to 0.032 in. With some nonasbestos substitute gasket material, the compressed thickness can be less than the desired amount.

The gasket is unconfined. Therefore, the proper bolting procedure and sequence must be carefully followed to ensure proper sealing, particularly on multistage pumps and for high pressure applications. The tightness of a horizontally split casing joint can be lost if the pump is thermally shocked (sudden entrance of a hot fluid into a cold pump), and as a result, the unconfined gasket can be blown out, causing a major leak. The tendency for the stress-relaxation of the asbestos substitute materials increases this possibility.

The outer portion of the upper casing flange of some horizontal split pumps appear to be distorted because the flange is not flat by a several thousandths. The appearance of the casing is not distortion. The mating surface on the top half is machined on a taper as shown in Figure 3, so that as the casing bolts are tightened the gasket becomes fully compressed in the critical areas. The flange is given a crown so that the outer edge on one side (either side) is tapered 0.0005 in per inch of distance from the edge of the stuffing box bore to the outside edge of the flange with a minimum of 0.007 in on small pumps. This method is generally used on pumps designed for high pressure and high temperature such as boiler feedwater service. The casing should not be remachined to remove the taper.

IMPELLER ATTACHMENT METHODS

On single stage, single suction API pumps, the impeller is keyed to the shaft to prevent rotation. Both the impeller and a "hook" type sleeve are held in place axially by a fastener threaded in a direction that is counter to pump rotation. This reduces chances of the fastener loosening. A design is shown in Figure 4 that incorporates a square key to drive the impeller and a Woodruff key to drive the shaft sleeve and a socket head cap screw fastener to secure the

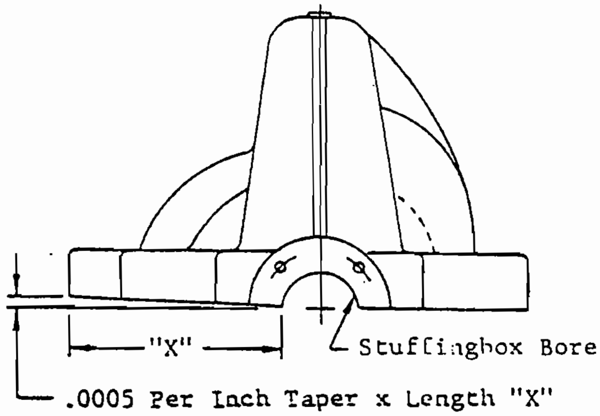


Figure 3. Crowned Faces on Parting Flanges—Multistage Axially Split Pump.

assembly. Flow disturbances also can cause loosened fasteners that will lead to unsecured impellers, a poor impeller hub bore-to-shaft fit, and a leak under the shaft sleeve. The shaft and/or impeller hub bore will then fret. The gasket fit faces of the shaft and hook type shaft sleeve may have to be trued up. The actual fastener method is not clear in most pump drawings but, in general, utilizes run-of-the-mill items that are purchased by the pump manufacturer in bulk. The material can be too hard (>R_c 25) to be compatible with many of the pumped products (chlorides, etc.). The fastener can then break easily during a run. This is especially common with socket head fasteners that are countersunk in contoured cone (Figure 4). Standard socket head capscrews are in the R_c40 range. Lock washers used under the head of hex headed cap screws are also hardened and suspectable to failure in some environments.

Many ANSI impellers are screwed onto or into the end of the shaft without a key. The screwed impeller is particularly vulnerable to backward rotation of the driver or a backflow of product.

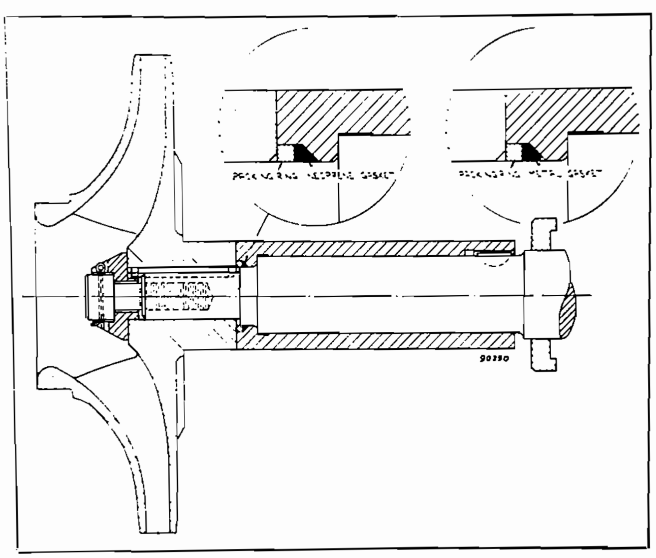


Figure 4. Attachment Method—Impeller and Hook Type Shaft Sleeve.

INSTALLATION INSPECTION PRIOR TO REMOVAL OF PUMP

If the pump to be repaired is an older one, compare it to the designs currently being specified. There may be an inherent hydraulic problem that is causing the failures. Thermal growth misconceptions can result in improper alignment targets, causing vibration problems. Prior to removal of the pump for repairs, eight component areas of a pumping system should be evaluated:

- *The foundation*—Many baseplate designs were very poorly designed in API pumps built prior to the 1981 Sixth edition of API 610. The current Seventh Edition requires that "the pump and its baseplate shall be constructed with sufficient structural stiffness to limit displacement of the pump shaft at the drive end of the shaft or at the register fit of the coupling hub to the values shown in Table 8 of the specification. Look over the baseplate of the failed pump for signs of a poor installation.
- *The driver*—Excitations from the vibrations of the driver (motor, steam turbine, gearing) can be transmitted to other components. Two pole (3600 rpm) motors in particular are prone to mechanical and electrically induced vibrations.
- *Mechanical power transmission*—Excitations from the coupling area, especially due to misalignment of the driver or eccentrically bored coupling hubs. Incorrect positioning of driver and pump such that distance between shaft ends (DBSE) exceeds the axial flexing limits of the coupling.
- *The driven pump*—Design of the pump can greatly influence the hydraulic interaction between the rotor and the casing and, thus, the problems encountered. The Sixth Edition and earlier specifications did not discuss the possible need for an NPSH margin over the $NPSH_R$. Nor was the concept of Suction Specific Speed and its effect on the range of stable flow a factor in the selection and application of pumps. The Seventh Edition also recognizes Gap A and Gap B impeller to casing clearances and provides limits for impeller applications.
- *The suction piping and valves*—Unfavorable incoming flow conditions like cavitation, intake vortex, or suction recirculation due to poor design and layout of suction piping and valves can cause flow disturbances [2, 3].
- *The discharge piping and valves*—Unfavorable dynamic behavior of piping because of loads from dynamic, static or thermal causes including resonance excitation [4, 5].
- *the instrumentation for control of pump flow*—Control system/pump interaction during startups or other periods of low flow can produce pressure pulsations. High pressure pulsations can occur due to hydraulic instability of the entire pumping system.
- *The alignment anchoring devices*—Once it is established, dowels or other devices hold the pump alignment. The Seventh Edition states that "Cylindrical dowels or rabbeted fits shall be employed to align casing components, or the casing and cover, and to facilitate disassembly and reassembly." The thermal growth guiding system of the pump must be effective.

These areas are aspects of the pump installation that are external to the pump rotating element that is normally removed for repairs. They are often not evaluated, resulting in repeat failures.

INSPECTION OF THE OPEN PUMP

Before opening, check the area of the pump curve to determine where the pump is actually being operated. Liquid flow in the impeller internal channels is a very complex phenomenon, especially at off-design operating conditions. The internal flow is unstable and unsteady, with violent changes sometimes occurring from channel to channel. In a presentation at the Fifth International Pump Users Symposium, Gopalakrishnan [6] included the pump curve shown in Figure 5. The curve tabulates all the turbulent flow problems that are encountered over the various operating ranges of a pump. All of these can be detrimental to the mechanical features of the pumping system and could be the cause of the pump failure.

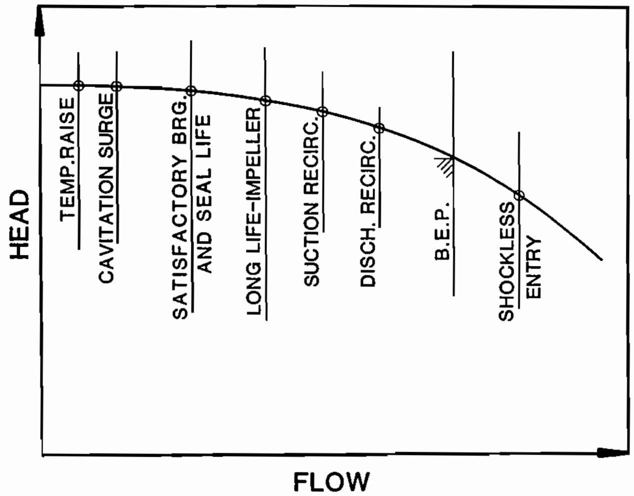


Figure 5. Typical Head-Flow Characteristic of a Centrifugal Pump Indicating Several Flow Rates [6].

As lower flows are approached, the problems accumulate to very destructive proportions. Examine the liquid channels of the impeller(s), the casing and adjacent piping for signs of the failure cause. Flow disturbances can result in the following:

- *Impeller erosion*—The turbulent flows cause severe erosion of the impeller vanes on both the leading and trailing edges in a very short time (well under nine months).

- *Impeller failure*—Low frequency hydraulic pulsations may cause fracture-type failure of the shrouds or covers of the impeller. These pressure pulsations may be in the magnitude of up to 20 percent of the total head at a frequency of about 1.0 to 10 Hz with below about 6.0 Hz being most common. An impeller failure may occur after only a few hours of operation at lower flows.

- *High failure rate of mechanical seals*—The hydraulic pressure pulsations in the impeller channels are very destructive to mechanical seals and may cause opening of the primary sealing faces.

- *High bearing failure rate*—The pump rotor is moved axially by the impeller hydraulic pulsations and can cause impact failure of bearings, particularly ball thrust bearings. In double suction pumps, the pulsations are out of phase on each side of the impeller and can be at varying frequencies, causing the rotor to shuttle back and forth.

THERMAL GROWTH PROBLEMS

The results of a study done by Heald and Perry [7] indicates that several long standing basic concepts of the pump industry about pump thermal growth must be revised. The warming stream flow patterns and rates must be carefully arranged to reduce the temperature differentials across the case of pumps in hot service. The thermal expansion guiding design or dowelling of the case must also be revised. Recommended revisions from their research are shown in Figure 6.

- Pumps larger than 8.0 in suction size should not be dowelled, as per current standard practices.

- Centerline keys that permit free radial and axial expansion towards the driver should be used in larger pumps.

- Three bottom connections—one in the volute and one in each suction passage, should be used with the warming stream flowing out to the suction vessel. Connections of at least 1.0 in should be used to ensure adequate flows.

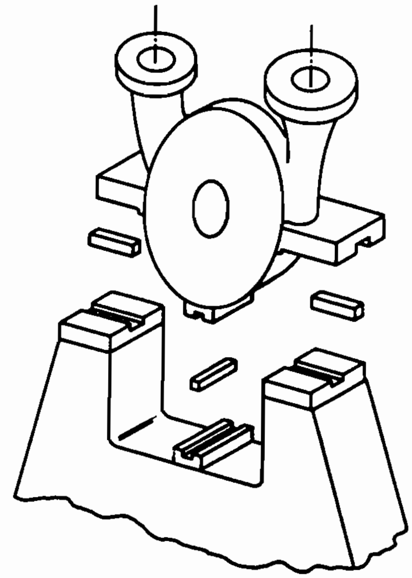


Figure 6. Keys Fitted to Allow Free Radial and Axial Expansion While. Minimizing Distortions Caused by Restricted Movement [7].

- Tangential entry of the warm up fluid stream can go a long way toward reducing the stagnant flow areas with in the pump.

- At temperatures greater than 500°F, extreme care must be taken to ensure that all auxiliary piping be welded to the case and the case rough machined *prior* to final heat treatment. This will prevent distortion due to stress relieving by temperature cycles in service.

Actual Pump Growth Example

A large, nine stage, barrel-type process pump operating at about 500°F can be expected to grow axially almost an eighth of inch. Radial growth is in the range of almost one sixteenth of an inch. The growth of the casing and the rotor tend to off set each other, but the biggest problems involve the tendency for the casing to "hump" causing a "droop" of the bearing brackets. This causes severe alignment problems, both internal and with the driver.

DYNAMIC BALANCING OF COMPONENTS

The objective of dynamic balancing is to make the mass center of the rotor coincide with the geometric center about which rotation occurs. Major residual unbalance limits currently used are roughly compared in Figure 7. The units used are in gram-inches per 100 pounds of rotor weight to permit a more direct comparison.

- American Petroleum Institute (API) Standards 610 (Seventh Edition), 612 (Third Edition) and 617 (Fifth Edition). This unbalance limit standard has been used by the U.S. Navy (U.S. Navy - MIL-STD-167 - Ships) for over 60 years and was adopted by API in the late 1980s. The specification states that the residual unbalance in each correction plane shall not exceed:

$$\text{Unbalance} = \frac{4W}{N}$$

Where:

N is maximum actual running speed, in rpm greater than 1000.

W is static journal weight on each end, pounds.

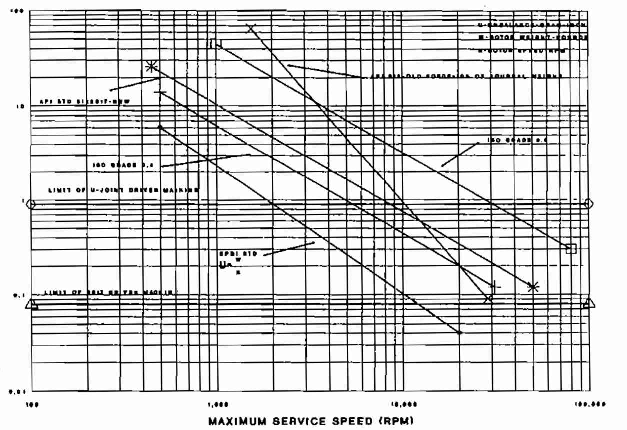


Figure 7. Unbalance Limits Comparisons.

• International Standards Organization (ISO) DR 1940 -This is a direct plot of residual unbalance per unit of rotor mass against service speed. Steam turbine and compressor rotors are assigned the 2.5 Grade. The upper limit of this Grade is about 15 W over N, while the lower limit is about 6.0 W over N. Pump impellers are assigned to the 6.3 Grade, which is about 35 W over N. These limits are not satisfactory for many turbomachinery applications. Putting the API and ISO unbalance limits on the same footing would require using about a ISO Grade 0.7.

The current API limits are more restrictive than the older API specifications and much tighter than the ISO standard. The expense in dynamic balancing is in the set up time, not the actual correction. Achieving a tighter standard is not expensive. A good balancing machine operator can achieve the API tolerances for almost the same manhours as any of the ISO limits. The 4W/N tolerance originated in an era when the readout system of a balancing machine consisted of dial indicators and grease pencils, but it has stood the test of time well. The tighter limits result in a reduction of stresses and forces acting on the machine components, especially the bearings, and makes a good balancing job the best investment to made during the repair of the rotor.

API Technical Paper 684, "Tutorial on the Standard Paragraphs for Rotor Dynamics and Balancing," 1994, covers balancing in very practical terms. It should be a reference for any one working in the machinery industry.

Nelik and Jackson [8] discuss which balance standard to use. The author's opinion favors the position taken in conclusion No. 8 by the authors. ISO 6.3G is far too liberal and, therefore, uneconomical for most pump rotors. It is time for a single standard, not widely diverging standards.

ROTOR COMPONENT BALANCING GUIDELINES

Impeller Checks

Centrifugal pump impellers are sand castings with physical variations of dimensions and mass. The quality of any dynamic balancing operation depends upon the control of radial runout and the elimination of internal couples along the length of the rotor. Impellers must be precision balanced on a *solid* mandrel to API 610 specifications. The strength of a rotating impeller can be adversely affected by improper grinding for balancing. The hub, a highly stressed part, should be avoided. Ground areas should be spread as much as practical and sharp corners or deeply ground holes avoided. To accomplish this balancing, the following guidelines are important:

• The impeller to shaft fit must be an accurately controlled slight interference fit. Many impeller castings are not properly stressed relieved so bore dimensions may change while in hot service or during heating for removal. Careless or excessive application of heat during installation or removal can also cause distortion and loss of fit.

• Impeller bores that have been reclaimed by a weld buildup and remachining are very susceptible to thermal stress. All welding should be done with a low heat input and it may be necessary to stress relieve the impeller before final machining.

• The shaft, thrust collar, pressure reducing sleeve, and all other rotating parts must be balanced to API tolerances.

• Prepare halfkeys as required for the balancing of individual impeller(s) on a mandrel. These keys must precisely fill the open keyways at the impeller bores the same amount as the keys used at final assembly.

• A precision balancing mandrel for impeller(s) must be made. A typical design is shown in Figure 8. The actual geometry should match the minimum pedestal spacing and the roller configuration of the balancing machine. The mandrel design guidelines are as follows:

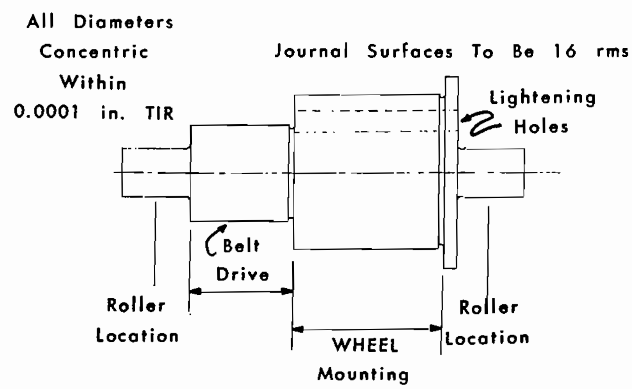


Figure 8. Precision Balancing Mandrel [3].

· The mandrel should preferably be made of a low alloy steel, i.e., AISI 4140 or 4340, which has been stress relieved.

· The journal surfaces should preferably be hardened and ground, with a finish not poorer than 16 rms.

· All diameters must be concentric within 0.001 in TIR.

· The diameter of the section where the impeller is to be mounted, should be established on the basis of heating the impeller hub to a temperature of approximately 300°F for installation and removal.

· Keyways are not incorporated in the mandrel.

· The impeller balancing mandrel should be checked to assure that it is in dynamic balance. Make corrections on the vertical faces only.

· The impeller balancing mandrel should be free of burrs and gouges.

· Mount each individual impeller, together with its halfkey, on the balancing mandrel by careful and uniform heating of the impeller hub, using a rosebud-tip torch, to a temperature of approximately 300°F. A temperature indicating crayon stick should be used to monitor the heating operation. Cool the impeller by directing a flow of shop air against the hub. When the impeller and mandrel have cooled to room temperature, install the mounted impeller in the balancing machine. Identify the required dynamic corrections with the balancing machine operating at the highest speed for the impeller diameter.

· Make the required dynamic corrections to the impeller by removing material over an extended area, with a relatively fine grade grinding disc. Blend in the ground area with adjoining contours. Under no circumstances should holes be drilled in the impeller for balancing.

SINGLE STAGE, DOUBLE SUCTION BRONZE IMPELLER PROBLEMS

When removing a hollow cast-bronze impeller, remove the plug in the passageway into the hollow area before heating to allow water/steam to escape without causing a rupture/explosion. If the impeller has been in hydrocarbon service, remove hydrocarbon prior to heating. If there is no plugged hole, drill into hollow area prior to heating (Figure 9).

If it is necessary to straighten bent impeller shrouds or vanes, do so *cold.* Hot bronze is subject to shattering when hammered or shocked. For the same reason, do not try to remove a heated bronze impeller by hammering, it may crack or shatter, use a steady force such as a press or strong-back.

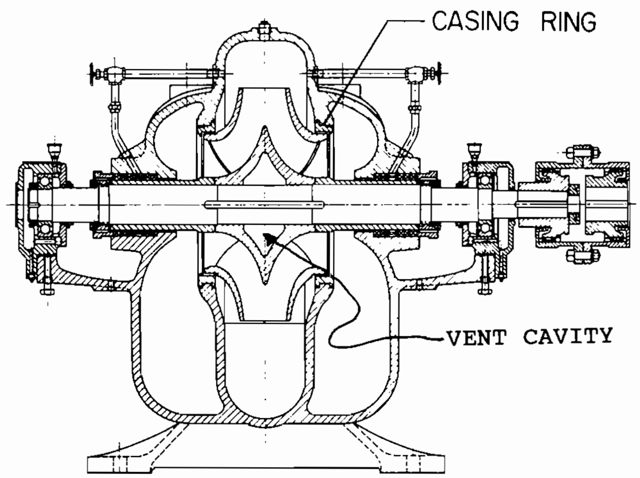


Figure 9. Venting of Bronze Impeller

EFFECTS OF GAP A AND GAP B

Careful machining of the volute or diffuser tips to increase Gap B (vane to volute or diffuser tip) while maintaining Gap A (shroud to casing) has been used for a number of years to reduce the vane-passing frequency vibration greatly. The pulsating hydraulic forces acting on the impeller can be reduced by 80 to 85 percent by increasing the radial gap from 1.0 percent to 6.0 percent. There is no loss of overall pump efficiency when the diffuser or volute inlet tips are recessed. Some slight efficiency improvement results from the reduction of various energy-consuming phenomena: the high noise level, shock, and vibration caused by vane-passing frequency, and the stall generated at the diffuser inlet.

Recommended dimensions are given in Table 1 for the radial gaps of the pump impeller to casing.

Trimming of Pump Impellers

The head and flow developed by a centrifugal pump can be adjusted by trimming the impeller diameter. The resulting performance of the pump is the subject of much confusion. Earlier publications [3, 9, 10] give some guidelines of the effects of impeller trimming on head, NPSH, flow, efficiency, and vibration.

Table 1. Recommended Radial Gaps for Pumps.

Type Pump Design	Gap A	Gap B - Percentage of Impeller Radius		
		Minimum	Preferred	Maximum
Diffuser	50 mils	4%	6%	12%
Volute	50 mils	6%	10%	12%

$$*B = 100 \; \frac{(R^3 - R^2)}{R^2}$$

R^3 = Radius of diffuser or volute inlet

R^2 = Radius of impeller

NOTE: If the number of impeller vanes and the number of diffuser/volute vanes are both even, the radial gap must be larger by about 4%.
Source: Dr. Elemer Makay

The "affinity laws" calculations generally dictate too great a cut on the impeller diameter because:

· The "affinity laws" assume that the impeller shrouds are parallel. In actuality, the shrouds are parallel only in lower specific speed pumps.

· The liquid exit angle is altered as the impeller is trimmed so the head curve steepens.

· There is increased turbulent flow at the vanes tips as the impeller is trimmed due to increased shroud to casing clearance or Gap A.

· Impeller diameter reductions for radial designs should be limited to about 70 percent of the maximum diameter. For pumps of higher specific speed values (2500-4000), trimming should be limited to about 90 percent of the maximum impeller diameter.

What to Trim

No hard and fast guidelines for the mechanical aspects of impeller trimming exist. For volute type pumps, it is best to cut the impeller vanes obliquely, as shown in Figure 10, leaving the shrouds unchanged or cut the vanes only, as Figure 11. Either method tends to even out the exit flow pattern and reduce the recirculation tendencies at the exit area. Gap A should be about 0.050 in (radial) for minimum axial vibration.

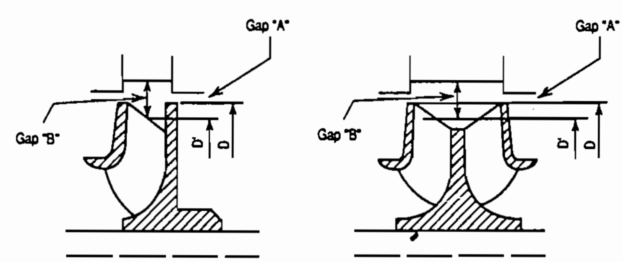


Figure 10. Trimming Impellers—Oblique Cuts of Vanes [3, 9, 10].

In most diffuser type pumps, it is best to trim only the vanes in order to control tip recirculation and the ill effects of an increased Gap A. This cut yields a more stable head curve due to a more uniform flow distribution at the exit area. Structural strength of the shrouds is a factor in the decision. There may be too much unsupported shroud left after a major reduction in diameter. The

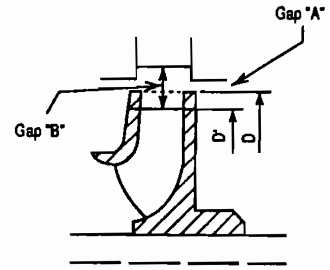
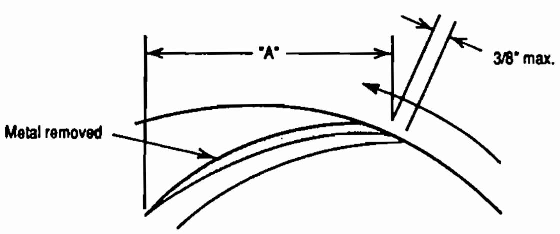


Figure 11. Trimming Impellers—Cutting Vanes Only [3, 9, 10].

double oblique cut of Figure 10 leaves the shrouds unchanged and solves the structural strength problem as well as improving the exit flow pattern.

Correction of Vane Shapes

Cutting impeller vanes results in blunt vane tips that cause disturbances in the volute and hydraulic "hammer" even when the impeller OD is the correct distance from the cut water (Gap B). Corrections can be achieved by tapering the vanes by "overfiling" or removal of metal on the *leading* face of the vane as shown in Figure 12. By sharpening the underside or "underfiling" edge of the vane as shown in Figure 13, the outlet area of the liquid channel can be enlarged. This will result in about five percent more head near the best efficiency point, depending on the outlet vane angle. At least $^1/_8$ in of vane tip thickness must be left. Sharpening the vanes also improves the efficiency slightly.



Length of blend for over filing	
Impeller diameter, in	"A" distance of blend, in
10 & below	$1\frac{1}{2}$
$10\frac{1}{16}$ through 15	$2\frac{1}{2}$
$15\frac{1}{16}$ through 20	$3\frac{1}{2}$
$20\frac{1}{16}$ through 30	5
30 & larger	6

Figure 12. Sharpening of Impeller Vanes by Overfiling [3, 9].

IMPELLER WEAR RING MOUNTING

Impeller wear rings are shrunk on and locked in place to prevent movement due to rubbing or a partial seizure. Interference fit between impeller wear rings and impeller is about 0.001 in to 0.0015 in for ring diameters ranging from $2^1/_2$ in to 6.0 in; 0.002 to 0.0025 in is used for rings $6^1/_2$ in to 12 in in diameter. Two methods of locking the rings are in general use. One method is to drill and tap either radially or axially for screws at the joint between the impeller and the impeller wear ring (Figures 14 and 15). After the screw is tightly bottomed, it is sawed off, peened and dressed down with a fine file. API 610 specifications limits the diameter of the pin to one-third the width of the ring. This method has some problems. Threads must be tight, which causes the wear ring to bulge at the pin with soft materials and may crack stellite and

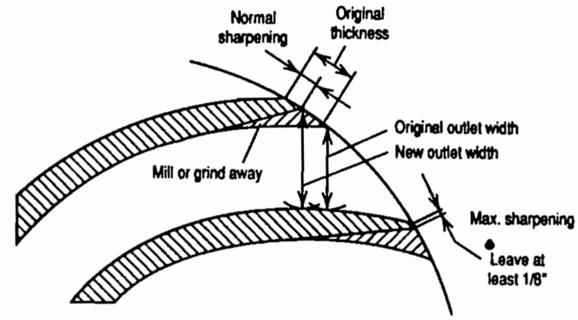


Figure 13. Sharpening of Impeller Vanes by Underfiling [3, 9, 10].

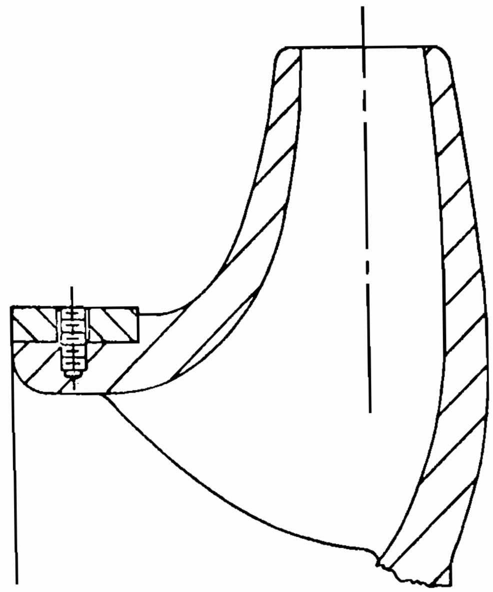


Figure 14. Installation of Wear Rings—Radial Pinning.

colmonoy hard surfacing (Figure 16). The second method substitutes tack welding for the screws. Thermal stresses and metallurgical changes in the impeller metal must be avoided. API specifications frown on this method but properly done it is a good method. After the rings are locked in place by any method, they should be checked for radial runout.

CASING WEAR RINGS MOUNTING

In most horizontally split pumps, the casing wear rings are mounted with a tongue and groove joint as shown in Figure 17. The wear rings "float" under hydraulic pressure of leakage to a running position and are locked in place by discharge pressure. The critical dimensions are on the bore and vertical flange face. The "float" clearance should be enough to accommodate about half of the variation of compressed gaskets. On smaller pumps clearance should be 0.004 to 0.006 in on the diameter of the rings. In a large circulating water pump, the clearance may be a little larger. In no instance should the case rings be clamped tightly when the cover is made up.

In radially split pumps, there are two ways of getting the proper wear ring clearance during repairs. One is to install new casing and impeller wear rings. The more economical alternative is to remachine the impeller wear ring by mounting the impeller in a lathe and taking a clean up cut on the circumference. After turning, the impeller ring should be measured and a new casing ring machined

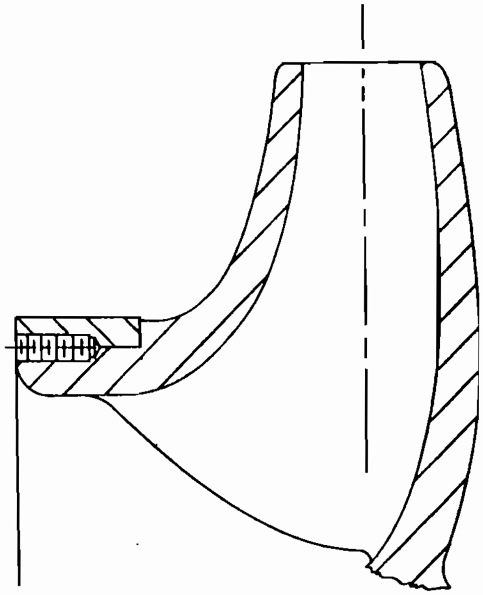


Figure 15. Installation of Wear Rings—Axial Pinning.

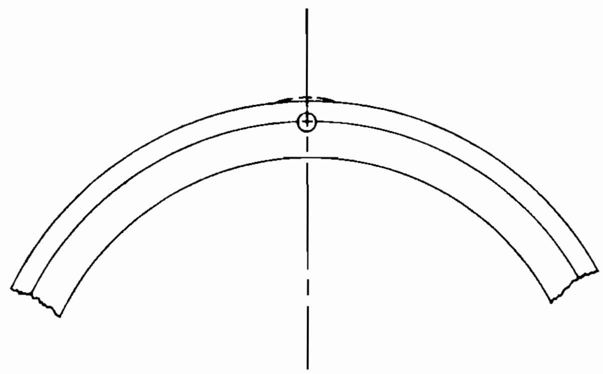


Figure 16. Installation of Wear Rings—Problems with Axial Pinning.

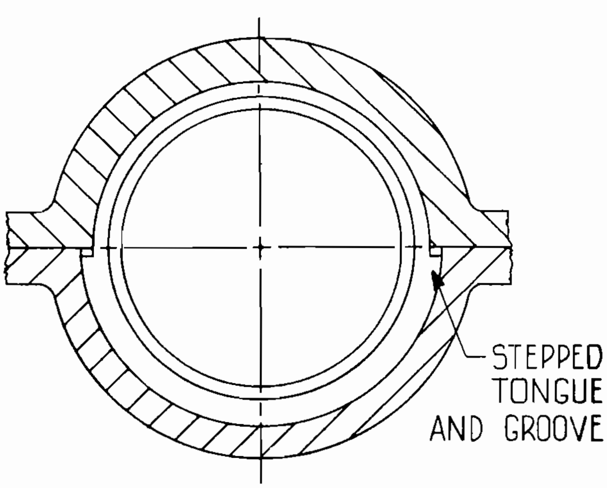


Figure 17. Installation of Case Wear Rings— "Floating" Design.

to dimensions for proper clearance. This is the reverse of normal renewal practices in horizontally split pumps.

Most casing rings for radially split pumps are plain bands with a shrink or tight fit in the case. The seal area of the ring to the casing is on the OD of the ring. The OD must be increased to take care of any corrosion or damage to the case if the pump has suffered a seizure-type failure.

"Ell" shaped rings are sometimes used in vertical split pumps. The static seal area is on the vertical face (Figure 18).

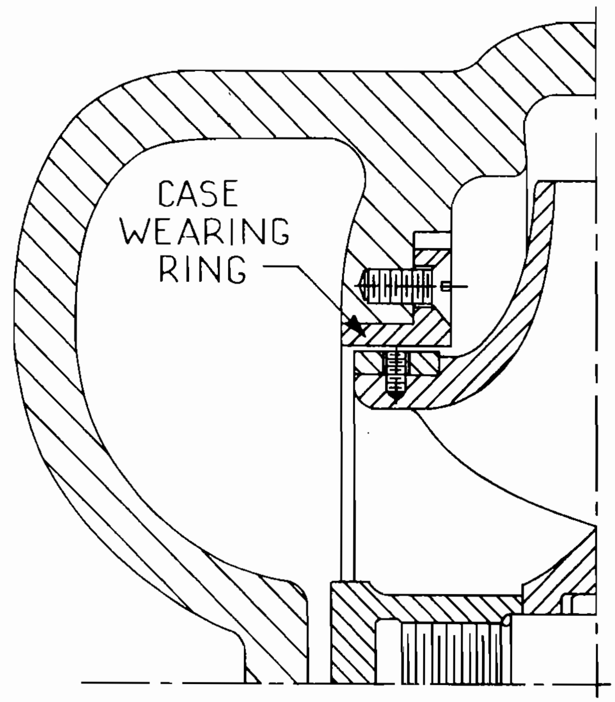


Figure 18. Installation of Wear Rings—Ell Shaped Case Ring.

Wear Ring Grooving

Abrasion is a real enemy of wear ring clearance. Surface hardened materials, stellite overlay, and heat-treated materials are effective to varying degrees in reducing wear. To further assist, spiral grooves in the softer wear ring are often used. Apparently small particles of abrasive matter rotating with the liquid and trying to enter the wear ring clearance, find these grooves and follow them to their outlet. Tests indicate that a 35 percent decrease in leakage can be achieved. Three grooves per inch, 3/32 in wide by 1/32 in deep with a right-hand spiral are commonly used. Experience indicates that the effectiveness is the same regardless of whether the spiral runs counter to or with the flow.

RUNNING CLEARANCES

The important consideration in establishing running clearances is dependability of operation and freedom from seizure under operating conditions, even at a possible small sacrifice in initial hydraulic efficiency. For cast iron, bronze, hardened 11 to 13 percent chromium, and materials of similar galling tendencies, the clearances found in Table 2 are typically used for operating temperatures below 500°F. For vertical pumps, the running clearances of Table 3 apply to the clearance of steady bearings or interstage bushings when materials of low galling tendency are used.

Wear Ring Replacement

For most pumps in the medium flow and head range, doubling the wear ring clearance will increase power requirements about

Table 2. API Wear Ring Clearances.

For cast iron, btonze, hardened 11 to 13 per cent Chromium, and materials of similar galling tendencies, the following running clearances should be used:

Diameter of Rotating Member at Clearance Joint (Inches)	Minimum Diametral Clearances (Inches)
Under 2.5	0.011
2.500 to 2.999	0.012
3.000 to 3.499	0.014
3.500 to 3.999	0.016
4.000 to 4.499	0.016
4.500 to 4.999	0.016
5.000 to 5.999	0.017
6.000 to 6.999	0.018
7.000 to 7.999	0.019
8.000 to 8.999	0.020
9.000 to 9.999	0.021
10.000 to 10.999	0.022
11.000 to 11.999	0.023
12.000 to 12.999	0.024
13.000 to 13.999	0.025
14.000 to 14.999	0.026
15.000 to 15.999	0.027
16.000 to 16.999	0.028
17.000 to 17.999	0.029
18.000 to 18.999	0.030
19.000 to 19.999	0.031
20.000 to 20.999	0.032
21.000 to 21.999	0.033
22.000 to 22.999	0.034
23.000 to 23.999	0.035
24.000 to 24.999	0.036
25.000 to 25.999	0.037

Notes: (1) For materials with severe galling tendencies such as 18-8 stainless steel or operating temperatures above 500° F, add 0.005 in to these diametral clearances.

(2) There should be a minimum 50 Brinell hardness difference in mating materials.

Table 3. Sleeve Bearning or Bushing Clearances.

Vertical pumps using pumpage lubricated sleeve bearings and throad bushings of bronze or carbon running against steel, 416, or 18-8 shafts or sleeves should have the following running clearances:

Diameter of Rotating Member	Clearance
0.75 to 1.50″	0.004 to 0.006″
1.50 to 2.50″	0.006 to 0.008″

five to seven percent. It is economical to renew a wear ring whenever the initial clearance is doubled.

REPAIR OF SINGLE STAGE HORIZONTAL PUMPS

Sharpening of techniques for disassembling and assembling pumps should reduce shop returns to a minimum. A little attention to a few details and suing some simple tools like a dial indicator can make the difference.

In order to have proper mechanical seal performance, the pump shaft must run "true." The shaft must be coaxial with the seal gland bore, *and* perpendicular to any seal faces held in the gland. If the shaft is not "true," the rotating seal face will oscillate axially as it attempts to maintain face contact with the stationary member. In addition, this axial oscillation will cause "fretting" or chafing of the shaft sleeve under the static seal of the rotating element of the mechanical seal. To confirm that the shaft is running "true" checks should be made.

• *Shaft Runout*—Shaft runout, deflection or lift should be checked by mounting two dial indicators at each end of the bearing housing and locating the stems, as shown in Figure 19. Rotating the shaft by hand will show radial runout at the seal end. Observing both indicators will show whether or not the shaft is bent. Lightly lifting the shaft may show a greater reading than shaft runout, indicating wear in the bearings or a poor fit of the bearings in the housing.

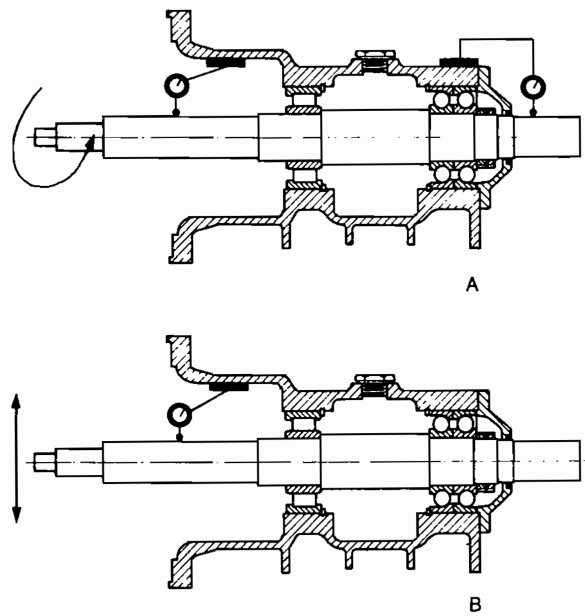


Figure 19. Checking of Shaft Runout or Deflection.

• *End Play*—With the dial indicator mounted on the pump bearing housing and the stem located against the shoulder of the shaft or sleeve, attempts should be made to move the shaft from end to end. End play normally be between 0.001 to 0.004 in (Figure 20). Different pump manufacturers apply various tolerances to the design allowance for thermal growth of the thrust bearings. Excessive end play is detrimental to the mechanical seal. Normally, hydraulic forces keeps the shaft thrusting in the same direction. On equipment that is subject to start/stop routine the contact faces of the seal can be damaged.

• *Sleeve Concentricity*—If the shaft runout is satisfactory, the shaft sleeve should be mounted in position. Allowing for tolerances on both the bore and the OD of the sleeve, the total runout should not exceed 0.004 in (TIR) (Figure 21). If runout is more than this, the defect should be corrected. Excessive runout can cause oscillations of the seal faces and variation of the fluid film thickness (a major problem for light hydrocarbons). Vibrations can also occur, possibly causing bearing failures.

• *Stuffing Box Bore Concentricity*—This check is frequently overlooked on between bearing, double suction, radially split

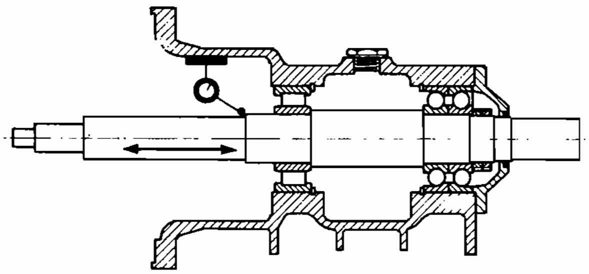


Figure 20. Checking of Shaft End Play.

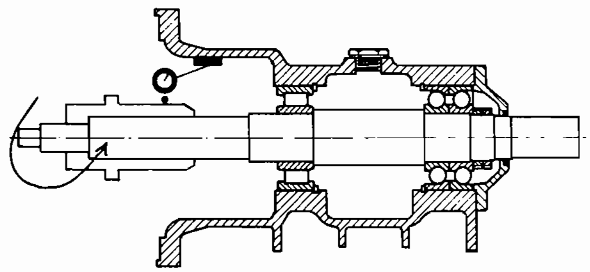


Figure 21. Checking of Sleeve Concentricity.

pumps. Because the bearing brackets are removed during repairs, they can be shifted. If the stuffing box bore is not concentric, the bearing brackets can be moved slightly and redowelled to center the shaft in the stuffing box. TIR should not exceed 0.004 in (see Figure 22 set up). Again, excessive runout can cause tracking problems of the seal face, resulting in uneven wear and eventually a failure.

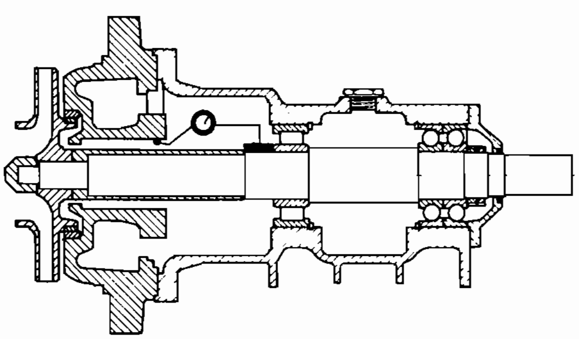


Figure 22. Checking of Concentricity of Stuffing Box Bore.

• *Stuffing Box Face Squareness*—A final indicator check is made with the pump completely assembled but without the seal(s) installed. Mount the dial indicator on the shaft and the stem located on the stuffing box face(s), as shown in Figure 23. When the dial indicator and shaft is rotated, TIR should not exceed 0.003 in. If the stuffing box face is not normal to the center line of the shaft, the stationary seal face will also not be square. This in turn will cause wobbling of the rotary faces. The excessive runout can cause a loss of contact and fretting wear between the seal faces.

BALL BEARINGS

A ball bearing is a piece of precision equipment; manufactured to extremely close tolerances. In order to obtain the maximum service from a bearing, the shaft and housing must be machined to the same exacting tolerances that are used in making the bearing. One of the bearings is fixed axially, while the other is free to slide. The outboard bearing (the closest one to the coupling on a pull out

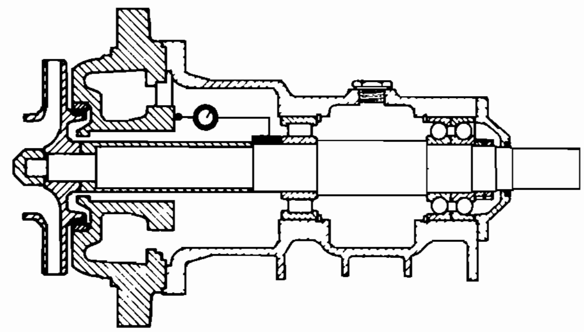


Figure 23. Checking of Squareness of Stuffing Box Face.

design) is fixed axially. The inboard bearing is free to slide within the housing bore to accommodate the thermal expansion and contraction of the shaft. Since the outboard bearing is fixed in the housing, it carries the axial thrust in addition to radial thrust.

The numerical code used in bearing identification is mostly standard among the various bearing manufacturers. However, the alphabetical prefixes and suffixes are not. Care should be exercised when identifying bearings from codes for the purpose of interchanging bearings that the meaning of all numbers and letters is determined, so that an *exact* substitution can be made [11].

Ball Bearing Fits

Both the bearing housing and bearing clearances should always be checked during the assembly of any pump. Unfortunately, many pump manufacturers do not indicate the proper bearing fits for either the shaft or the housings to guide shop repairs. The original dimensions of both the housing and the shaft will change from time to time from oxidation, fretting, damage from locked bearings, and other causes. Every bearing handbook has tables to aid you in selecting fits. The vibrational effect of looseness on the bearing fits is different for the housing and the shaft.

Housing Fits

Ball bearing fits in the bearing housing are of a necessity slightly loose for assembly. If this looseness becomes excessive, vibration at rotational speed and multiple frequencies will result. Do not install bearings with ODs outside of the given tolerance band, since this might result in either excessive or inadequate outer race looseness. The following table of rules of thumb are good guideline for looseness (Table 4).

Shaft Fit

A loose fit of the shaft to the bearing bore will give the effect of an eccentric shaft, at a one times running frequency vibration pattern. The objective of the shaft fit is to obtain a slight interference of the antifriction bearing inner ring when mounted on the shaft. The bearing bore should be measured to verify inner race bore dimensions. Do not install bearings with an ID outside of the given tolerance band, since this might result in either excessive or

Table 4. Rules of Thumb: Bearing Housing Fits.

1.	Bearing OD to housing clearance - About 0.00075 inch loose with 0.0015 inch maximum.
2.	Bearing housing out of round tolerance is 0.001 inch maximum.
3.	Bearing housing shoulder tolerance for a thrust bearing is 0 to 0.005 inch per inch of diameter off square up to a maximum of 0.002 inch.

inadequate shaft tightness. Good rules of thumb for guidelines are given in Table 5.

Table 5. Rules of Thumb: Bearing Shaft Fits.

1.	Fit of bearing inner race bore to shaft is 0.0005 inch tight for small sizes: 0.00075 inch tight for large sizes.
2.	Shaft shoulder tolerance for a thrust bearing is 0 to 0.0005 inch per inch of diameter off square up to a maximum of 0.001 inch.

PUMP GASKET COMPRESSION

Many single stage process pumps incorporate the back pull out feature, so that the pumping element may be removed from the case without breaking the piping connections. This construction also permits centerline mounting of the casing to minimize thermal expansion problems allow operation at higher pressures and temperatures for hydrocarbons. There is one fact that cannot be overlooked in working on this type of pump. The head must be made up square. This is important for two reasons:

- To obtain correct compression of the gasket for pressure holding capabilities.
- To maintain correct internal alignment and positioning of the rotating element with respect to the stationary casing components.

Starting in 1981, API Standard 610 specification required that the heads be machined such that when the heads are pulled "metal-to-metal," the gasket will be compressed the correct amount. Pumps manufactured before 1981 may have a separation when the gasket is made up properly. The head can be cocked if the separation is not maintained uniformly by use of feeler gauges.

MULTISTAGE DOUBLE CASING TYPE PUMP REPAIR

Special Tool Requirements

Some special tools are needed to properly assemble a multistage, double casing pump such as that shown in Figure 24.

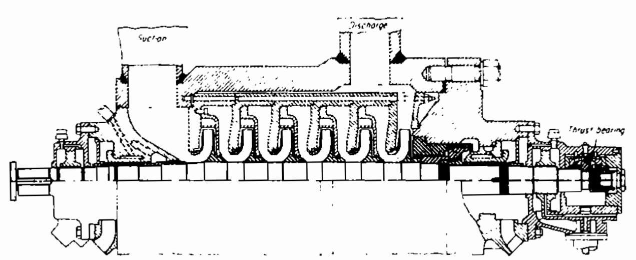


Figure 24. Typical Multistage Double Casing Pump.

The following is a typical listing of those tools.

- Micrometers for measuring bores of individual diffuser cover bushings and wear rings.
- A mandrel for checking alignment of the bushings in all cover or diffuser sections after they are stacked.
- The plates and tie rods that make up the inner casing stacking jig.
- Rails that are used to insert or extract the inner casing from the outer casing.
- Several mandrels used for balancing impellers and pressure reducing sleeve or balancing drum.
- Pressure Reducing Sleeve (balancing drum) lock nut tool.
- Air ring used for cooling the shrink fits on the impellers.

SHAFT RUNOUT CHECK

The grooves cut for split impeller locator rings makes the shaft tend to develop runout very badly. Checking this runout is very important. The following is good practice for this step.

- The shaft should be checked for proper diametral fits.
- If the shaft has been thermal spray or chrome coated on the journals, an area adjacent to the journals should be used to support the shaft providing there is no runout.
- Despite its symmetrical nature, the shaft must be balanced.
- Prepare halfkeys for the keyways of the bare shaft. These should be carefully taped in position, using high-strength fiber-impregnated tape; several turns are usually required. *Note*: Tape sometimes fails during spinning in the balancing machine. It is, therefore, important that adequate tape be used for the protection of personnel against the hazard of flying halfkeys.
- Mount the bare shaft, with halfkeys in place, in the balancing machine with the supports at the journal locations. Spin the bare shaft at a speed of 300 to 400 rpm for approximately ten minutes. Shut down, and check the radial runout (TIR) at mid-span using a 1/10 MIL dial indicator; record the angular position of the high spot runout value. Spin the bare shaft at a speed of 200 to 300 rpm for an additional five minutes. Shut down, and again check the radial runout (TIR) at mid-span; record the angular position of the high spot and runout value. Compare the results obtained after the ten-minute and five-minute runs; if they are the same, the bare shaft is ready for further checking and balancing. If the results are not repetitive, additional spinning is required; this should be continued until two consecutive five-minute runs produce identical results.
- Check the radial runout (TIR) of the bare shaft at least three impeller locations, approximately equidistant along the bearing span, and near the shaft ends. Record the angular position of the high spots and the runout values at each location. The shaft is satisfactory if both of the following conditions are satisfied.
  - The radial runout (TIR) at the section of the shaft between journals does not exceed 0.001 in.
  - The radial runout (TIR) outboard the journals does not exceed 0.0005 in.
- With the balancing machine operating at about 800 rpm, make the required dynamic corrections to the bare shaft using wax. When satisfactory balance is reached, start removing material at the face of the step at each end of the center cylindrical section of the shaft. Under no circumstances should material be removed from the section of the shaft *outboard* of the journal bearings.
- If the balancing machine is not available for the above shaft runout checking, the following method can be used.
  - Mount the shaft on "V" blocks at bearing areas to confirm mechanical runout. Maximum runout should be 0.001 in.
  - With the shaft still mounted in the "V" blocks, check electrical runout at the probe burnished points. Maximum runout should be 0.00025 in.

PRELIMINARY STACKING OF ROTOR (IMPELLERS ONLY)

The pump rotor must be assembled for balancing and runout checks *without* the diffuser sections in place.

- The rotor assembly should be stacked vertically, with the shaft held in a special fixture.
- All impellers, the thrust collar and pressure reducing sleeve (balancing drum) should be heated to 250°F for mounting. Under no circumstances should a temperature of 450°F be exceeded.
- Measure the bore of the impellers, thrust collar and balancing drum to assure uniform expansion.
- Install the match-marked split rings and key for the first stage. It may be necessary to use a dab of grease to hold these pieces in place.
- Install the first stage impeller using these procedures.
  - Heat the impeller to about 250°F.
  - Place the cooling air ring on the OD of the impeller vane tips with the air turned off. Cooling air flow is through the impeller vanes to suction eye.
  - Move the impeller into position. It will be easier to "hit" the key in the shaft if the keyway position of the impeller is marked with a felt tip pen. The keyway in the impeller isn't visible while installing it on shaft.
  - When the impeller counter bore makes contact with the split rings, maintain constant downward pressure on the impeller. Turn on the cooling air. After about 30 seconds the downward pressure can be released. *Note*: Cooling air *must* remain on until the impeller is at ambient temperature.
- Carefully remove the rotor to "V" blocks for runout indication. With the journal areas mounted on the "V" blocks maximum runout should be about 0.001 in.
- If the rotor exhibits no appreciable runout changes after the mounting of the first impeller, the decision may be made to mount all remaining impellers without checks after mounting of each individual impeller.
- Install the remaining impellers on the shaft using these same procedures.
- After the rotor is stacked, check it for runout. If it is satisfactory, then tighten the impeller nuts and check the runout again.
- If all checks are satisfactory, final balance the rotor.
- If excessive runout is present, check the faces of the shaft nuts. Reclaim the faces as necessary. All runouts must be brought into tolerance before balancing the rotor.

ROTOR ASSEMBLY BALANCING

After the rotor is assembled, it is ready for final dynamic balancing. Observe the response of the balancing machine at several speeds and select the best speed. For most rotors of the lengths and diameters encountered in this style pump, about 500 to 800 rpm is probably best for the balancing machine.

1. Spin the rotor and record the first run numbers. If the first run is below 25.0 gr/in of imbalance on either plane, proceed to step 2. If the imbalance is above this limit, take the following steps.
   - Reduce the balancing machine speed to 200 rpm.
   - Heat each impeller backside (discharge side) fit with a #3 Rosebud torch tip placed 1.0 in from the hub bore for 30 seconds. Heat all impellers in succession this way. The rotor must be rolling at 200 rpm throughout the entire heating and cooling process. DO NOT stop turning the rotor or allow rotor to turn any faster than 200 rpm until entire element has cooled to ambient temperature to prevent any rotor bow.
   - After the entire rotor has cooled to ambient temperature make a balancing run at about 1100 rpm. If readings are still above 25.0 gr/in, unstack rotor to determine the cause of the excessive unbalance. If the readings are below 25.0 gr/in, proceed to step 2.
2. Install suction nut hand tight and make another balance run. (Readings should go down when mass is added.) If readings go up, the nut is probably out of balance (step 10). Make additional runs using putty to determine correction location and weight. Do step 3 before drilling correction hole. Drill correction holes only in line with the existing spanner wrench holes on the suction locknut.
3. Tighten suction nut with spanner wrench. If readings go up, it indicates fits are off. Try another nut or correct existing nut.
4. Install pressure-reducing sleeve or balancing drum only after preheating to 250°F. Tighten the locknut snug to hold the pressure reducing sleeve in position until cool. After the element has cooled to ambient temperature, remove nut and make another run. Readings should again go down. Proceed to step 5. If they go up, either the pressure reducing sleeve is out of balance or fits are off, remove and check.
5. Install the pressure reducing sleeve locknut snug, make another run. Readings should go down. If they do, proceed to step 6. If they go up, either the locknut is out of balance, or the face fit to the pressure reducing sleeve is off. So, tighten the nut and make another balance run. If the readings go up again, the fits are still off. Try another locknut or correct the face fits on this one. If readings go down but are still higher than before adding the nut to the element, make a correction to the nut. Use same procedure as for the suction nut to determine correction location and weight. Drill correction holes only on the small diameter fit of the pressure reducing sleeve locknut (see step 10).
6. Tighten pressure reducing sleeve locknut. Make another balancing run. If the readings go up, fits of the locknut are off. Try another nut or correct fits. If the balancing run readings stay the same or go down, proceed to step 7.
7. Preheat thrust collar to 250°F, install with along with the locating spacer and make the locknut up snug. Make another balancing machine run. If the balance readings go down, proceed to step 8. If they go up, try tightening the locknut more. If imbalance readings go up more, this means locknut faces are off. Repair the thrust collar, spacer sleeve or locknut fits. If readings go down proceed to step 8.
8. Do a six-point residual imbalance test. Record results on residual imbalance form and run sheet.
9. Scribe mark with an air pencil all components for precise reinstallation at final stacking.
10. After the rotor is dynamically balanced, reconfirm the runouts.
    - Mount the rotor on the "V" blocks at the journal areas and reconfirm the mechanical runouts. Maximum runout shall be about 0.001 in again. Check the thrust collar face runout at this time-maximum 0.0005 in.
    - Reconfirm the electrical runout at the probe burnished areas. Maximum runout shall be about 0.00025 in.

Note: The suction locknut and pressure reducing sleeve locknuts are the only components on the rotor that are not dynamically balanced. All other parts are precision-balanced and no corrections should be made on them after assembly on the rotor. If locknuts require balancing, remove only enough material to reach last balance amplitudes and phase readings (before nut was installed).

FINAL ROTOR AND DIFFUSER STACKING

For the final assembly of the rotor and diffusers the following guidelines should be followed.

- Carefully unstack the rotor.

- Vertically stack the rotor and the diffuser elements using the same procedure as the impeller only stacking described above.

- Vertically lift check the shaft after each cover is installed to check the bumping or float distances. Jack can be placed under coupling end of shaft. Design lift should be maintained through out the stacking procedure.

- Since diffuser covers should have 0.005 in interference fit, some heat may be needed to install them. If the cover doesn't drop into fit, heat preceding cover around entire circumference for approximately one minute with a small rose bud. Then tap lightly with a lead hammer. Verify alignment of cover to outer casing key as each succeeding cover is installed while the lower cover is still warm.

- After pressure-reducing sleeve is installed and cooled, tighten nut to match-marks established during balance procedure. Leave nut snug until pressure reducing sleeve or balancing drum is cool. Take bump or float dimensions so that dimension "Z" shown on Figure 25 can be set readily by the thrust collar position.

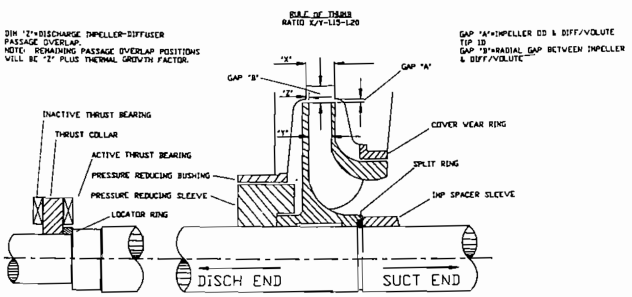


Figure 25. Impeller-Diffuser Passage Overlap in Multistage Pump.

- Install inner casing stacking jig after element has cooled to ambient temperature.

PUMP ASSEMBLY

- Install the inner casing and the rotor in the outer casing using a guide fixture.

- Install the expansion gaskets and spacers between the inner casing and the outer head as shown in Figure 26. Note that various combinations of gaskets and spacers are used (four gaskets and three spacers; three gaskets and two spacers). *Caution*: Failure to ensure that the inner head gasket spaces that were supplied with the pump, or precise replicas, are utilized in the reassembly could cause serious problems. Improper makeup of the intermediate diffuser covers, the suction spacer, and the discharge diffuser spacer could cause liquid recirculation within the pump, and in turn cause decreased efficiency and erosion, particularly at the high pressure end. The thickness of each spacer should be measured along with the gasket thicknesses to achieve approximately 0.015 to 0.020 in compression on each gasket (about half of the compression when used as a head gasket).

- Install head and tighten. Most heads will go to a metal-to-metal fit internally so the gap at the head will be approximately 0.048 to 0.058 in.

- Install bearing housings with the lower half journal bearing in place. Move the coupling end housing until suction impeller is centered in wear rings within 0.001 in, using full length feeler gauges. Move thrust end housing up or down until pressure reducing sleeve is centered in the mating bushing within 0.001 in, again using feeler gauges. Record feeler gauge clearances. Take dial indicator readings of seal chambers bores and faces. Record all readings.

- Ream bearing housing dowel holes.

- Install thrust collar with the locating spacer and the inboard thrust pads to check rotor position. The locating spacer should be machined to control the final position.

- Check the journal bearing shell "crush" or "clamp" by adding 0.005 in shim to the bearing cap gasket (normally in thick).

- Remove the bearing housings. Install mechanical seal assemblies, and leave them loose on the shaft until bearings and thrust are finally set.

- The thrust bearing clearance or float of about 0.010 to 0.012 in is controlled by machining the bearing cover or adding gasket thickness.

PUMP THERMAL GROWTH PATTERNS

There are many misconceptions about relative thermal growth between the driver and the pump, but there are greater misunderstandings about the relative growth of components within the pump. Thermal distortion of the pump case and rotor is routine in hot service pumps. The relative growth of a typical multistage hot service pump of the double case design is shown in Figure 27. The suction end of the case is anchored by a socket welded to the case that fits over a locating pin welded to a support "cradle." The discharge end of the case is fitted with a key which fixes the

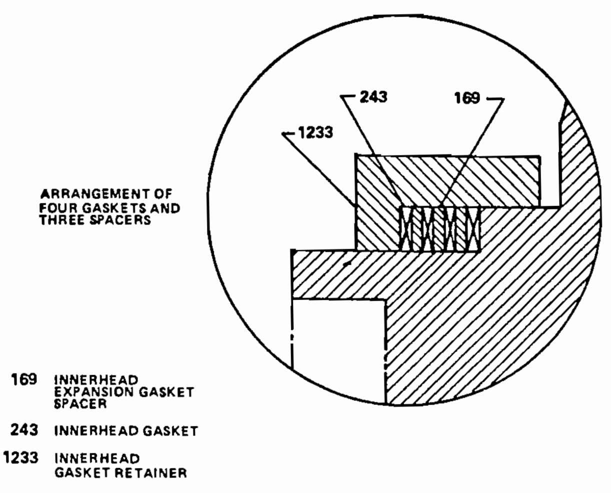


Figure 26. Expansion Gasket and Spacer Arrangements.

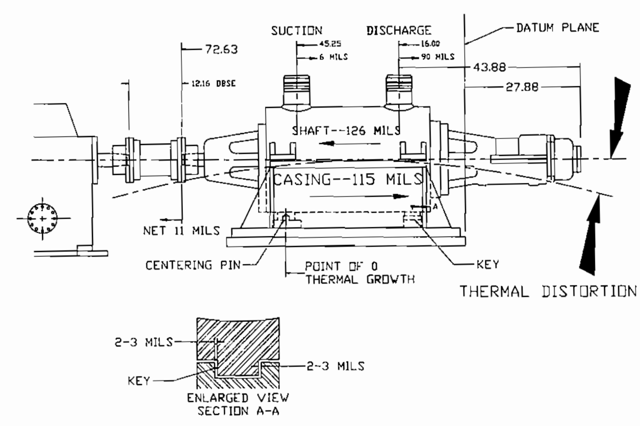


Figure 27. Marginal Version of Guiding System for Thermal Distortion of a Multistage Pump.

vertical centerline relative to the foundation. The support feet are near the centerline of the case. During thermal expansion the case grows radially as well as axially. Correction of major problems with this expansion design was the subject of study by Johns [12, 13]. An even worse design is shown in Figure 28. In this design, radial expansion of the outer casing spread the pedestals apart and the adjustable centering guides actually fell out of position. In addition, both sets of cap screws in the foot rail were tightened very tight and the case "humped" badly. This caused contact between the rotor and the stationary element.

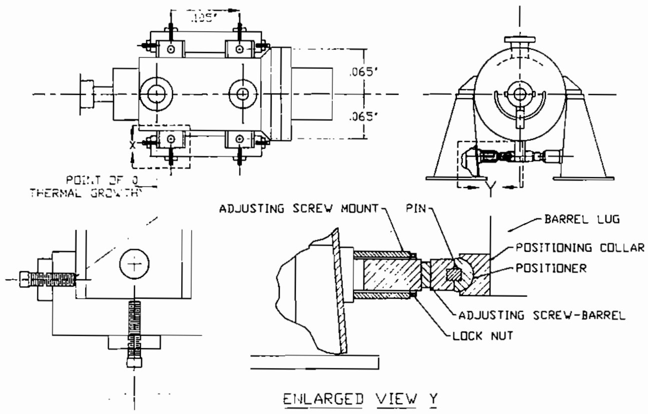


Figure 28. A less Satisfactory Version of Guiding System for Thermal Distortion of a Multistage Pump.

As a result of machining and welding variations, the cradle and the case are custom tailored to each other and must remain together. The cradle cap screws must not be too tight to permit sliding of the casing.

VERTICAL PUMP REPAIR

Vertical pumps present maintenance problems that are distinctively different from those of horizontally mounted pumps. Many more parts are required to rebuild a vertical turbine pump since typically four stages of this mixed flow design are required to produce the head of one stage of the radial flow horizontal pump. Because of more wetted parts, a vertical process pump made out of alloy materials is considerably more expensive than an equivalent horizontal process pump.

Vertical pumps consist of three major components as shown in Figure 29 and Figure 30.

- *Head and Driver Assembly*—An electric motor and a cast or fabricated base from which the column and bowl assembly is suspended.

- *Column and Shaft Assembly*—The column pipe which suspends the pump bowl assembly from the head assembly and serves as the conductor for the liquid from the pump bowl to the discharge. Within the column pipe is the line shaft that transmits the power from the driver to the pump impellers. The line shaft bearings in the column are generally lubricated by the liquid being pumped.

- *The Pump Bowl Assembly—The Pump Proper*—Each bowl unit or stage has an impeller.

VERTICAL PUMP PROBLEMS

The vertical pump's rotor is not gravity stabilized. The gyroscopic effect of rotation can cause lots of damage to the rotor and the casing when problems arise. There is a great potential for cost savings if proper repairs that result in increased reliability and extended runs can be made.

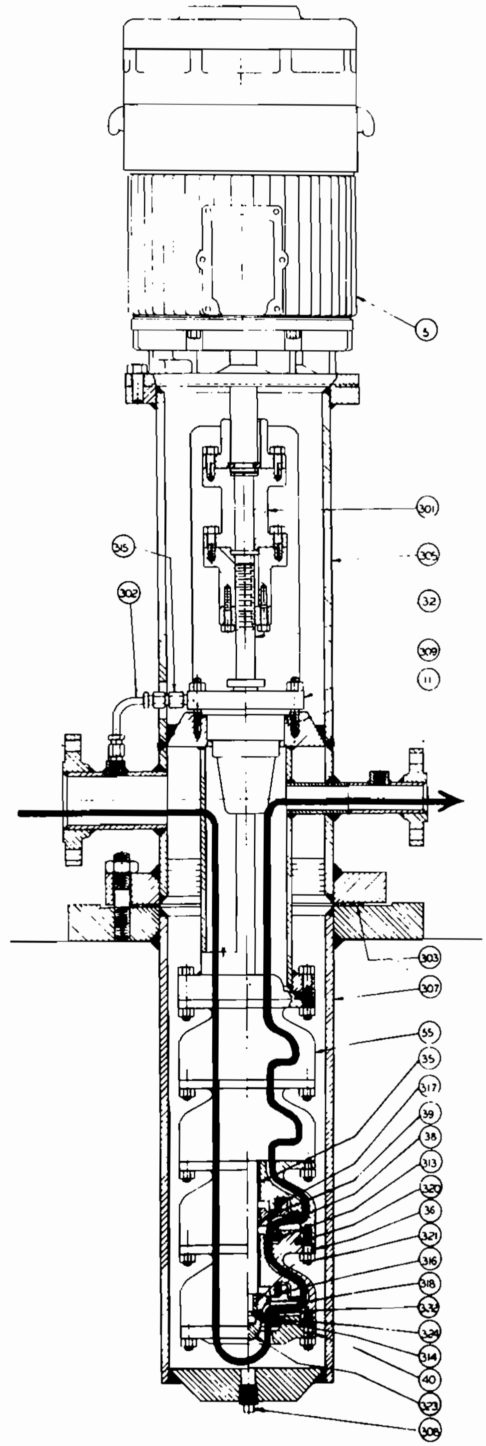


Figure 29. Typical Multistage Vertical Process Pump.

Critical Speeds in Vertical Pumps

Critical speeds are usually not a factor in centrifugal pumps. The stiffening and damping effect resulting from a pressure drop across close clearance spaces of wear rings and interstage bushings is large and raises the critical speed far above what is known as the "dry" value. The bowl assembly may have a high "wet" critical;

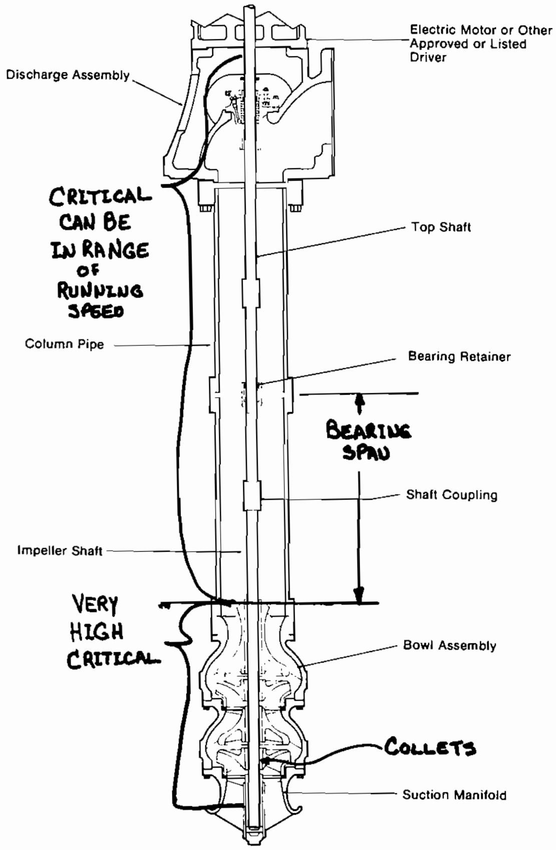


Figure 30. Rotordynamics of a Typical Vertical Pump.

but the column assembly can have a very low critical, as depicted in Figure 30. The only liquid force influencing the column shaft is a small amount of hydrodynamic wedge in the bushings.

Vertical pumps have been manufactured with extremely large spacing between the shaft guide bushings. This reduces the wet criticals of the shafting to below the operating speeds of the vertical pump. This "whip action" drastically increases the chance of seal failures and other mechanical problems due to the runout on the shaft.

Problem pumps may be helped by repairs that involve shortening the bearing span. Column spacing between shaft guide bushings will be as indicated in Paragraph 2.9.2.1 of API-610, or less, as shown in Figure 31. Adherence to these guidelines should keep the first critical of the drive shafting at least 130 percent to 150

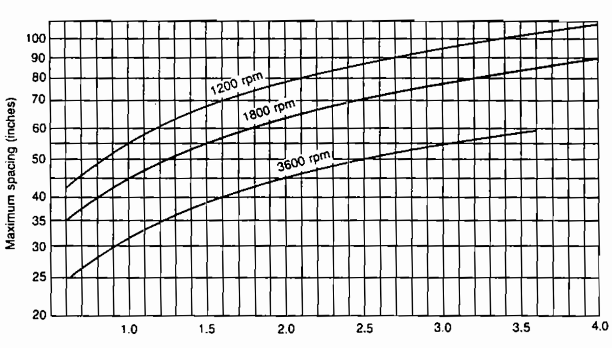


Figure 31. Maximum Spacing Between Shaft Guide Bushings Vertical Pumps [1].

percent higher than its operating speed to allow for the effects of wear on the bushings.

Startup Considerations—Upthrust

When a vertical pump is first started, the pump has a momentary upthrust hydraulic action. The equipment including the motor must be designed to take this upthrust. When a vertical pump is operated at very high capacities on a continuous basis, it very likely will have a continuous upthrust. Damage can include buckled line shafts, seal leakage, impeller rubs, and motor bearing failures.

Impeller Locking Devices

There are five impeller locking devices to position impellers on the shaft. These generally determine how pump is disassembled.

- *Split Collet*—generally should be limited to smaller pump impeller diameters (under 6.0 to 8.0 in)
- *Locknuts*—Mounted on each end of shaft and spacers between each impeller
- *"Gib" Key*—An ell shaped key fitted in a stopped keyway
- *Spring Type Ring*—A snap ring located in grooves in the shaft
- *Threaded Pin*—A pin goes through the impeller hub into the shaft
- *Split Rings*—Split rings located in a wider groove and retained by a shrunk on or bolted on keeper ring.

Bearing Troubles

The pumpage lubricated lower bearing of a vertical turbine pump is especially vulnerable because there is almost no pressure differential across the bushing to provide lubrication. A flushing line from the discharge as shown in Figure 32 will extend the pump life considerably. If the fluid being pumped contains abrasive material, a clean fluid from another source may be used provided the process fluid will not be contaminated by the flushing fluid. A cyclone separator like that used on mechanical seals can be utilized to clean up the pump discharge so that it can be used for flush. The external piping must be added.

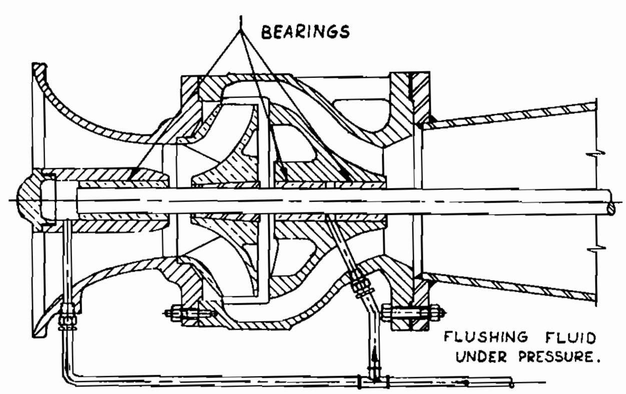


Figure 32. Vertical Pump Bowl Designed for Flushing Fluid.

BUSHING AND BEARING MATERIALS

Before replacing the bushings, evaluate the original material selection and its suitability for your current application.

- *Bronze*—Most common material used *except* it should not be used on any corrosive service, above 200°F or below -20°F, or sandy dirty water sump service.

- *Carbon*—Second choice after bronze. Use for corrosive products and product lubricity is low (propane containing hydrogen sulfide) and below -20°F. Do not use for dirty pumpage. Remember, the coefficient of thermal expansion for carbon is very low.

- *Cast Iron*—Use on mildly corrosive service (sour crude), temperatures above 200°F, when particulate matter is present in the product, and with products of fair or better lubricity.

- *Rubber*—Applicable only to water pumps. Use in conjunction with open impellers on sandy water or dirty sump service. Do not use in hot service or where any product detrimental to neoprene is present.

- *Teflon, Glass Filled or Otherwise*—Use only as last resort and with Engineering guidance. Carbon is the best suggestion for most services where Teflon is specified.

- *Metallic Filled Carbon*—A graphite that has been filled with a metal to alter its characteristics. Copper, nickel, babbitt and powdered iron are a few of the alloying metals. Commonly called by the trade name "Graphalloy".

DISASSEMBLY OF PUMP

The basic construction principles of a vertical pump influence the repair methods. Some of the differences actually will determine from which end you start to disassemble the pump. Pump bowls are: 1) individually screwed together, 2) bolted together with a series of bolts around the bowl flange, and 3) bolted together with tie bolts which reach from the bottom of the pump to the top through the bowls (sometimes called channels) or case. Method number 2, a bolt circle in a flange, is generally the design encountered.

DISASSEMBLY PROCESS

- Before starting the disassembly process, prepare a data sheet for the pump. Record the name plate data, noting any instructions regarding "lift" and other items pertaining to this particular pump.

- Set the pump on the floor with the discharge flange facing down. Grind flats at all mating joints in the discharge column and the pump bowls, 180 degrees from the discharge. Starting from the bottom or suction, match-mark all joints at the ground flats and record stencil configuration as to letter height; etc. Be careful not to confuse any previous markings with current ones.

- With the pump thrusting toward the suction, the upper shaft extension length to the motor adapter face and adapter face configuration.

- Measure and record total pump shaft float.

- When a pump that utilizes a split collet bushing to lock the impeller on the shaft is disassembled, a measurement *must* be made from the end of the shaft to the impeller and recorded for proper reassembling.

- The bowls and impellers match-marks and numbering should be carefully recorded as the pump is disassembled to ensure proper reassembly. Measure and record the float at each stage.

- Match-mark the impellers and the collets (or other mounting devices such as keys, locking pins, split keepers, and keeper plates) before removing the impellers from the collets and shaft in order to maintain their relative positions. This will help in machining new wear rings and in positioning the impellers for balancing. If this is not done, past experience has shown that the impeller skirt will runout considerably. When reinstalling the impeller on the collet, be sure the match-marks are lined up.

- Match-mark lineshaft couplings to their mating shaft on each end.

IMPORTANT! Lineshaft couplings should be removed by using pipe or chain tongs on the mating shafts only. Wrenches applied to the coupling can collapse the coupling causing it to seize on the shaft. If normal efforts (including heating) to remove the lineshaft coupling fail, split the coupling with a cutoff wheel in a grinder. This permits salvaging the more expensive pump shaft sections.

- The rigid coupling halves from the motor and the pump should be checked to be sure they run true. The spacer spool should also be checked for square. All couplings should be steel. Cast iron is subject to frequent failures.

- The shaft should be checked for straight after complete disassembly of the pump.

AFTER DISASSEMBLY

- All clearance dimensions should be taken, i.e., impeller wear ring to bowl or channel, spider bushing to shaft, throat bushing to shaft.

- All wear ring and bushing clearances should be sized properly utilizing the clearances specified by API 610, correcting as needed for product temperature.

- All bearing spider bores should be checked for parallelism with the shaft column.

- All bushings in the spider should be locked in by a method consistent with the material and design of the bushing. Most vertical pumps have brass throat bushings and brass spider bushings.

- All bushings, because of the tight clearances to the shaft, should have a spiral groove on the bore in order to provide for lubrication.

- Most vertical pumps with bottom suction (or normal flow pattern) should have a sand collar and a plug in the bottom bowl.

- All impellers should be individually balanced to compensate for uneven wear of the casting, pieces broken out of the vanes, etc.

- Where possible, a one-piece shaft is preferred. If couplings are required, a radial weep hole should be drilled in the center of the coupling. Screw the shaft together before assembling the impellers to check the shafts for runout. This can be done on the knife edge rollers.

- A sleeve shrunk on the shaft in the area of the spider bushings or an eccentric spider bushing can be used instead of shaft replacement where shaft wear is a problem.

- Wear rings should be grooved in the direction of rotation, starting on the outboard side of the wear ring and threaded towards the impeller. A course pitch with two or more leads will reduce leakage across wear rings by as much as 35 percent.

SHAFT INSPECTION

The pump section and the drive shaft should be supported in a horizontal position on knife edge rollers for checking.

- *Sag*—The vertical movement of deflection of the shaft on the rollers as a result of the effects of gravity, a flexing under its own weight. The condition is not permanent, due to the elasticity of the material. If the shaft is rotated, the sag will move to a new spot. Sag is not runout and is not shaft bow.

- *Shaft Bow*—This is the amount of permanent deflection of a shaft beyond the elastic limit of a material. The shaft has been permanently yielded to by an external force. Unlike sag, this bow will not relax or return to a new position as it is rotated. The deflection must be removed by either thermal or mechanical means.

• *Shaft Runout*—This is the measure of shaft roundness combined with any permanent bow or bend. The effect of shaft runout is greatest on a mechanical seal or packing due to the orbital movement or path of the shaft. A shaft with 0.003 in runout will move the seal faces 0.003 in per side or a total of 0.006 in. Permissible runouts should be limited to about 0.002 in with some small leeway for lower pressures and speeds of less than 3600 rpmi.

If the shaft must be replaced because of excessive wear in the bushing areas and/ or bows, the shaft quality should be selected carefully. The following is a suggested specification for purchasing stock [14].

Pump Shaft Quality (PSQ)
Type 416 Stainless Steel

Process: Hot rolled bar, turned, ground and polished

Specifications: ASTM - A581, A582 and AMS 5610

Diameter Tolerance:	0.750 thru 1.500	+ 0.000"/- 0.002"
	1.625 thru 2.437	+ 0.000"/- 0.003"
	2.687 thru 2.937	+ 0.000"/- 0.004"
	3.000 thru 4.000	+ 0.000"/- 0.004"

Length: 20' to 24' Random lengths

Straightness Tolerance: 0.0015" per foot - Guaranteed F.O.B. point of shipment

Out of Round Tolerance: One half of the diameter tolerance

Surface Finish: 16 RMS

Chemical Analysis:	C	MN	P	S	SI	CR	MO
Min				.15		12.00	
Max	.15	1.25	.06		1.00	14.00	.60

Mechanical Properties: Tensile Strength - 100,000 PSI Minimum
Yield Strength - 85,000 PSI Minimum
Brinell Hardness- 262 Maximum

MACHINING OF SHAFT SECTIONS

Care must be taken in the handling of shafts during the machining process to avoid damaging or bending. When the shaft is "chucked" in a lathe, soft copper pads should be used under the jaws and the shaft must be supported to avoid bending as it is rotated.

REASSEMBLY PROCESS

• The pump section should be assembled in a *vertical* position. If a multistage vertical pump is assembled in a horizontal position, the tendency for both the bowls and the shaft to develop a bow is very great. With only about 9.0 mils bowl bushing clearance, the shaft comes in contact with the bushing after two or three bowls are installed because many bowls will have clearance (0.002 in to 0.003 in) in spigot fits. Vertical assembly will randomly distribute this clearance. Pumps have been encountered which had over $^1/_4$ in curvature from the top to bottom of a sixteen stage section after assembly.

• The jack bolt in the suction piece must be used to position the end of the shaft so that accurate spacing of the impellers to the casing can be maintained. Any load applied on previously installed impellers can bump them off the collet while installing the next impeller. Several impellers have come loose in this manner and destroyed themselves as well as the bowls when the pump is run.

• Impellers have also been cracked in the hub area due to over tightening of the collet. Collet-held impellers require much more care and attention in mounting than those held with split rings, ell keys, or snap rings as locking arrangements.

• All bushing and wear ring fits should be coated with nickel-based ant-seize compound to aid in avoiding damage during startup.

• After the pump is reassembled, it should be checked to determine if it has the proper lift. Rotate the rotor assembly by hand to check for possible rubs.

• After the pump bowls are assembled it may be necessary to assemble the column sections in a horizontal position because of their excessive length. The assembled portions should be rotated 180 degrees as each additional component is installed. This will stagger alignment clearances and fits through the length of the columns and cause less total deviation of the centerlines.

• Upon installation the discharge head, compare the shaft extension with the measurement taken before disassembly. Deviations of $^1/_{16}$ in to $^1/_8$ in are acceptable. Minor corrections can be made by installing or roving gaskets of differing thicknesses from mating faces. As a extreme, the upper shaft section must be machined.

• The shaft extension should be supported and the shaft locked for shipment. If it is necessary to leave the jack bolt in the suction bell in place, a warning tag should be attached to that the bolt *must be* removed prior to installation. If the bolt is left in place at installation, the pump will be wrecked upon startup.

ALIGNMENT AT INSTALLATION OF VERTICAL PUMPS

Since vertical pumps have relatively loose fitting bearings and the shafts are fairly small in diameter and flexible, alignment of the motor to the pump is a problem. The pump shaft should really be an extension of the motor shaft. The "coupling" is rigid and does not flex or in any way allow for misalignment. To be sure that the pump assembly is correctly positioned, a special bushing must be used when reinstalling the pump as shown in Figure 33. The split bushing is preferred.

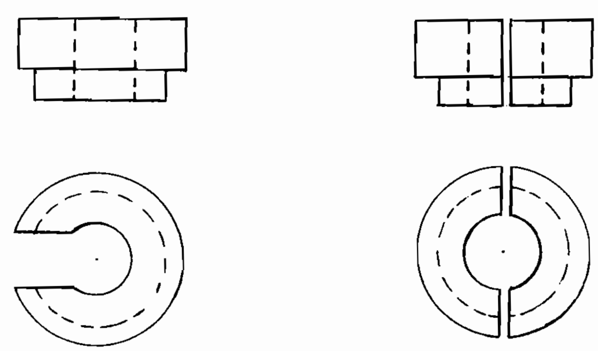


Figure 33. Alignment Bushings for Vertical Pumps [14].

CONCLUSIONS

Achieving quality pump repair is a matter of paying attention to details and taking many measurements to determine the relationship of mating parts. Above all, the individuals involved in the repair work must think about what they are doing at all times.

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