ELECTRIC POWER RESEARCH INSTITUTE

2012 TECHNICAL REPORT

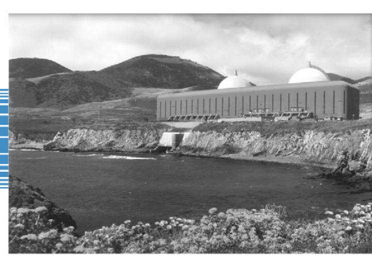
Nuclear Maintenance Applications Center: Vertical Pump Maintenance Guide

Update to EPRI Report 1003467

Plant Maintenance Support

Reduced Cost

Equipment Reliability



Nuclear Maintenance Applications Center: Vertical Pump Maintenance Guide

Update to EPRI Report 1003467

1026475

Final Report, October 2012

EPRI Project Manager
M. Pugh

This document does **NOT** meet the requirements of
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NOTE

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Bryan Sprock	AmerenUE
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John Montgomery	Exelon
Jeff Severyn	FirstEnergy
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PRODUCT DESCRIPTION

Vertical pumps are used in several applications in nuclear power plants, such as condensate, heater drain, residual heat removal (BWR plants only), circulating water, and service water/river water applications.

Background

Results from several annual Electric Power Research Institute (EPRI) Nuclear Maintenance Applications Center (NMAC) maintenance issues surveys indicate that members are experiencing difficulties with various maintenance issues associated with vertical pumps. Some of the problems identified a need for information and assistance with pump and motor alignment issues, predictive maintenance, monitoring recommendations, vibration measurement and diagnosis, overhaul and repair recommendations, and inspection guidelines. In 1994, NMAC issued a report on vertical pump maintenance, *Deep Draft Vertical Centrifugal Pump Maintenance and Application Guide* (NP-7413).

Objectives

- To provide information regarding the maintenance, repair, installation, and troubleshooting of vertical pumps installed in nuclear power plants
- To address specific vertical pump maintenance, reliability, and operational issues that are encountered by nuclear plant maintenance and engineering personnel
- To supplement and augment existing NMAC information on vertical pump maintenance by providing current and state-of-the-art technology and maintenance practices

Approach

In cooperation with the NMAC Pump Users Group and interested NMAC members, a task group of utility engineers and industry experts was formed. This group identified and addressed key design and maintenance issues encountered by plant personnel, and they provided input in the preparation of this report. Experience-proven practices and techniques were identified, summarized, and collected and are also included in this report.

Results

This report provides the end user with an understanding of vertical pumps, which includes elemental component descriptions, common materials of construction, and typical applications. This report addresses such maintenance challenges as pump and motor alignment issues, predictive maintenance and monitoring recommendations, vibration measurement and diagnosis, overhaul and repair recommendations, and inspection guidelines.

Applications, Value, and Use

The information contained in this report represents a significant collection of technical information, including techniques and good practices, related to the maintenance, monitoring, and troubleshooting of this important piece of plant equipment. Industry knowledge from recent experiences and improvements has been included in this revision. Assemblage of this information provides a single point of reference for power plant personnel, both now and in the future. Through the use of this report, EPRI members should be able to significantly improve and optimize their existing plant predictive, preventive, and corrective maintenance programs related to this equipment. In turn, this will help members to achieve increased reliability and availability at a decreased cost.

Keywords

Design engineers
Plant maintenance
Plant operations
Plant support engineering
Pumps

CONTENTS

LIST OF FIGURES

LIST OF TABLES

1
INTRODUCTION

1.1 Background

Results from several annual NMAC Maintenance Issues Surveys indicate that members are experiencing problems in dealing with various maintenance issues associated with vertical pumps. Some of the problems identified a need for information and assistance with:

- Predictive maintenance, troubleshooting, and vibration analysis
- Increased pump and seal life
- Pump-to-motor alignment
- Pump inspections, rebuilding and repair, reverse engineering, and installation
- Maintenance issues related to pump application, design, and operation

Vertical pumps ranked high on the list of topics that members requested NMAC work on to provide additional assistance and guidance. Vertical pump issues have also been identified as a significant maintenance and engineering challenge by the NMAC Pump Users Group.

From an article in *Pump & Systems* magazine, October 2002, by Douglas C. Branham, "What makes a Rolls Royce different from a Chevy? Each of these cars has a drive train, four wheels, a steering wheel, seats, a body, etc. The difference is in the additional workmanship that goes into the Rolls, and the tighter specifications to which it is built. For example, the pistons in a Rolls Royce are weight balanced and matched. This makes for a smoother running engine, a longer life, and lower maintenance costs.

The same type of question can be asked of ANSI or API 610 pumps. Each has a case, shaft, seals, an impeller, a driver, etc. But, why does one user achieve a six-month mean time between failure (MTBF) and another gets eight years? What is the difference between a good machine and a great machine? The answer to these questions is simple—a little extra work and care make the difference."

1.2 Purpose

This *Vertical Pump Maintenance Guide* is intended to provide information regarding the design, operation, maintenance, repair/installation, and troubleshooting of vertical pumps installed at nuclear facilities. The vertical pumps that are targeted are in, but are not limited to, condensate, heater drain, residual heat removal, circulating water, and service water/river water applications. The goal for this report is to address specific vertical pump maintenance, reliability, and operational issues by drawing upon industry expertise and good practices being used currently throughout the industry at the nuclear plants.

This report provides personnel involved with the maintenance of vertical pumps with useful information regarding the following:

- Vertical pump design and description, including elemental component description and information, typical materials of construction, and performance criteria
- Vertical pump vibration monitoring, diagnostics, and troubleshooting techniques
- Maintenance and repair guidelines including:
  - Repair specifications and criteria
  - Pump alignment
  - Balancing
  - Pump lift
  - Cavitation erosion, abrasive erosion and corrosion causes, solutions, and maintenance strategies

This report is intended to supplement the previously issued NMAC report NP-7413, *Deep Draft Vertical Centrifugal Pump Maintenance and Application Guide*.

1.3 Organization

The organization of this report is as follows:

Section 1 is an introduction and general description of the report's purpose and organization.

Section 2 provides a basic description of vertical pumps, their design, construction, and operation, and a discussion of typical power plant applications.

Section 3 provides recommendations for vertical pump vibration monitoring, diagnostics, and troubleshooting.

Section 4 provides maintenance and repair recommendations.

Section 5 provides a list of references used in the development of this report.

Appendix A is a glossary of terms, definitions, and acronyms that are used in this report and that relate to pumps in general.

Appendix B provides an explanation of net positive suction head (NPSH) as it relates to vertical pump installations and the effects of specific gravity and viscosity.

Appendix C through F, I, and J provide additional reference information related to vertical pumps.

Appendix G provides general vertical pump troubleshooting guidance.

Appendix H provides an example of a repair/refurbishment specification for an APM circulating water pump.

Appendix K includes a listing of the Key Points that are identified in this report.

1.4 Key Points

Throughout this report, key information is summarized in *Key Points*. Key Points are bold-lettered boxes that succinctly restate information covered in detail in the surrounding text, making the key point easier to locate.

The primary intent of a Key Point is to emphasize information that will allow individuals to act for the benefit of their plant. The information included in these Key Points was selected by NMAC personnel, consultants, and utility personnel who prepared and reviewed this report.

The Key Points are organized into three categories: O&M Costs, Technical, and Human Performance. Each category has an identifying icon, as follows, to draw attention to it when quickly reviewing the report.

Key O&M Cost Point

Emphasizes information that will result in reduced purchase, operating, or maintenance costs.

Key Technical Point

Targets information that will lead to improved equipment reliability.

Key Human Performance Point

Denotes information that requires personnel action or consideration in order to prevent injury or damage or ease completion of the task.

Appendix K contains a listing of all key points in each category. The listing restates each key point and provides a reference to its location in the body of the report. By reviewing this listing, users of this report can determine if they have taken advantage of key information that would benefit their plants.

2
VERTICAL PUMP DESIGNS AND DESCRIPTIONS

2.1 Vertical Pumps

A centrifugal pump is a kinetic machine that converts mechanical energy into hydraulic energy. The centrifugal pump consists of a set of rotating vanes, enclosed within a housing or casing and used to impart energy to a fluid through centrifugal force. Stripped of all refinements, a centrifugal pump has two main parts:

- A rotating element – including an impeller and a shaft
- A stationary element – made up of a casing, stuffing box, and bearings

In a centrifugal pump (vertically or horizontally mounted), the liquid is forced by atmospheric or other pressure, into a set of rotating vanes. These vanes constitute an impeller, which discharges the liquid at its periphery at a higher velocity. This velocity is converted into pressure energy by means of a volute or by a set of stationary diffusion vanes surrounding the impeller periphery. Pumps with volute casings are generally called volute pumps, while those with diffusion vanes are called diffuser pumps. Diffuser pumps were once quite commonly called turbine pumps, but this term has been more selectively applied to the vertical deep-well centrifugal diffuser pumps usually referred to as vertical turbine pumps.

Vertical pumps fall into two classifications, dry-pit and wet-pit. Dry-pit pumps are surrounded by air or have the volute or case embedded in concrete. Wet-pit pumps are either fully or partially submerged in the handled liquid.

2.1.1 Vertical Wet-Pit Pumps

Wet-pit pumps are the most common vertical pumps installed at nuclear facilities and are the focus of this maintenance guide. Vertical wet-pit pumps are intended for submerged operation and are manufactured in a great number of designs, mainly depending upon the service for which they are intended. Wet-pit pumps can be classified as:

- Vertical turbine pumps
- Propeller or modified propeller pumps
- Volute pumps

The design of vertical pumps illustrates how a centrifugal pump can be specialized to meet a specific application. Figure 2-1 illustrates a turbine design with closed impellers and enclosed line shafting, and another with semi-open impellers and open line shafting.

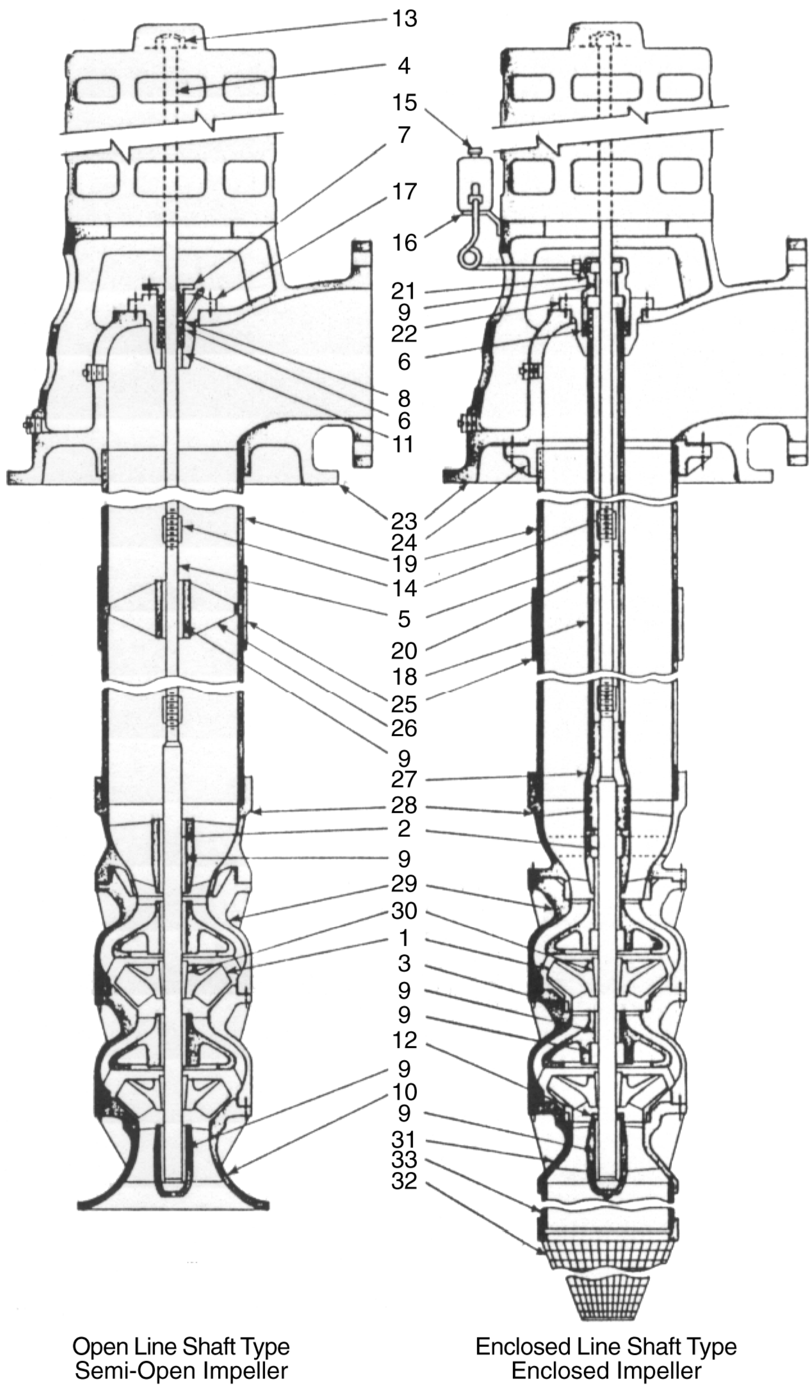


Figure 2-1
Typical Vertical Turbine Pumps

2.1.2 Vertical Dry-Pit Pumps

Many vertical dry-pit pumps are basically horizontal designs with minor modifications (usually in the bearings) to adapt them for a vertical-shaft drive. Bearings for vertical dry-pit pumps and for intermediate guide support are usually antifriction grease-lubricated types to simplify the problem of retaining a lubricant in a housing with a shaft projecting vertically through it. Self-oiling babbitt bearings with spiral oil grooves are used with larger units, for which antifriction bearings are not available or desirable. The pump is connected by a rigid coupling to its motor, which is provided with a line and a thrust bearing.

In nuclear facilities, the most common applications for vertical dry-pit pumps are residual heat removal (RHR) in pressurized water reactor (PWR) plants, and condenser circulating water, which takes suction from cooling towers (see Figures 2-2, 2-3, and 2-4).

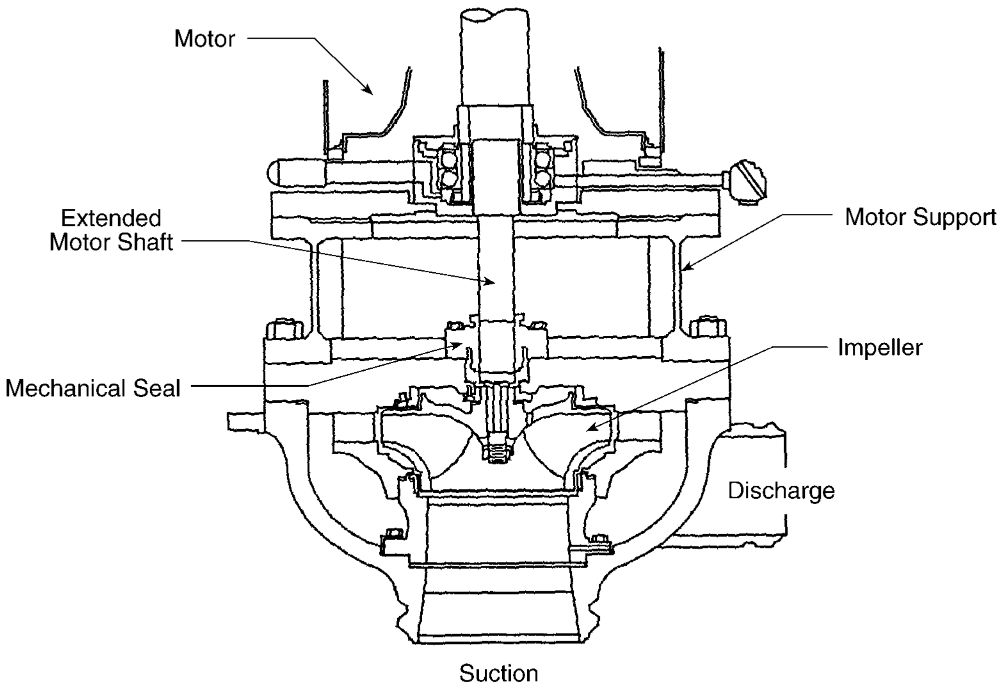


Figure 2-2
Typical WD/WDF Pump in PWR Plant RHR Application

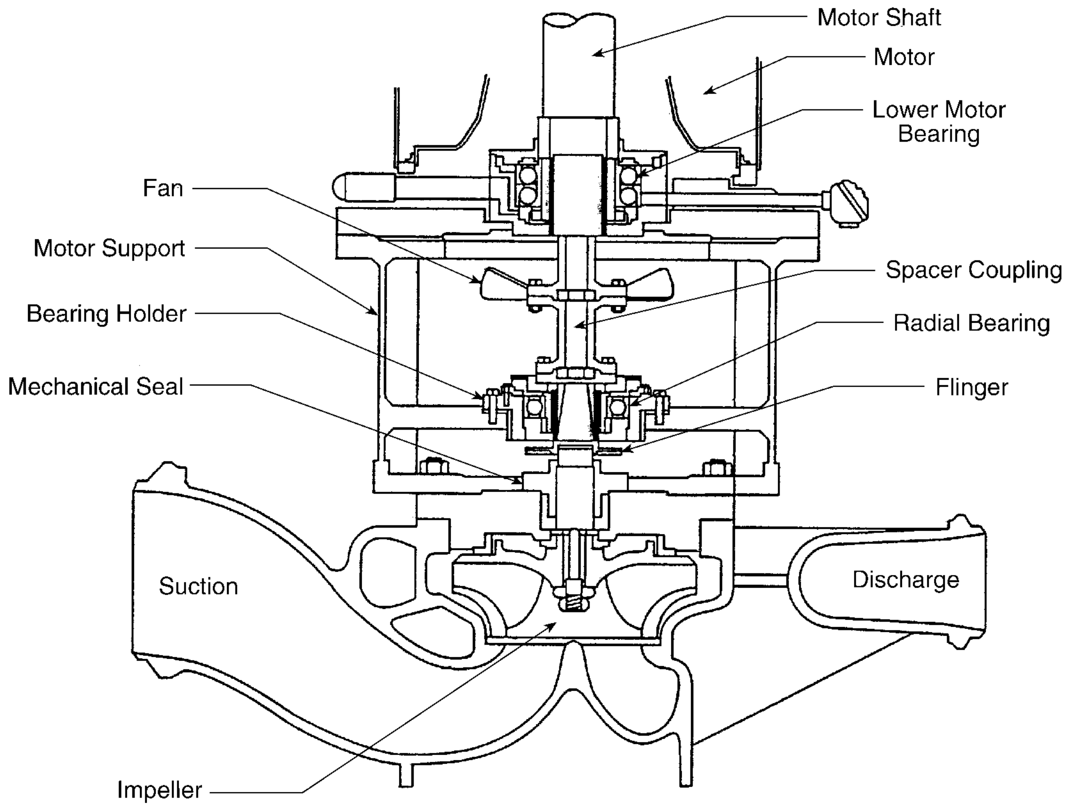


Figure 2-3
Typical WD/WDF Coupled Pump Modification in PWR Plant RHR Application

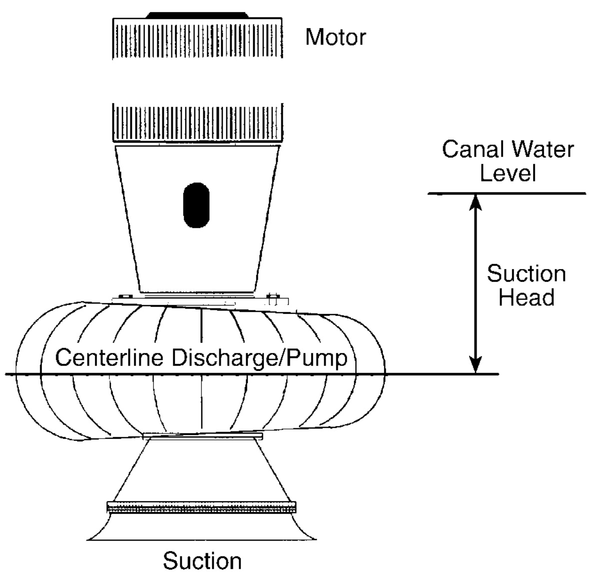


Figure 2-4
Typical Dry-Pit Pump in Circulating Water Application

2.2 Vertical Pump Assembly

Vertical pumps consist of three basic components (see Figure 2-5):

- The bowl
- The column assembly
- The head assembly

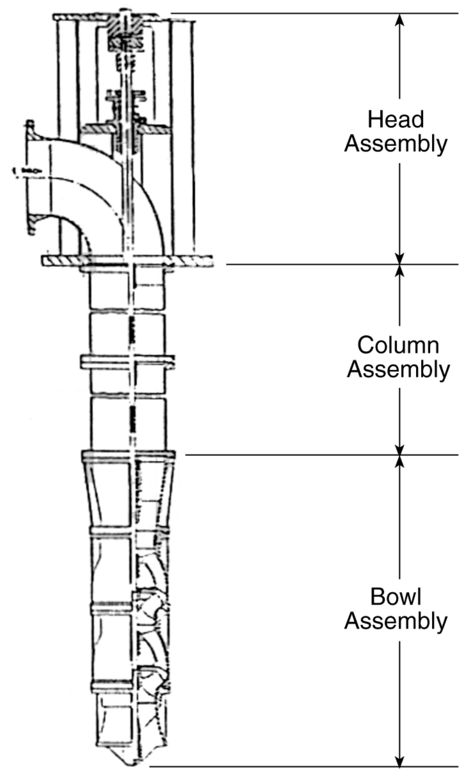


Figure 2-5
Three Basic Assemblies of Vertical Pumps (Bowl Assembly, Column Assembly, and Head Assembly)

2.2.1 The Bowl Assembly

The simplest bowl assembly configuration consists of a straight shaft with taper collet mounted impellers, and bowls that are joined together with straight threads and furnished with a shrink fitted bearing. The impellers can be either the:

- Enclosed design with both a front and a back shroud (Figure 2-6)
- Open design without a front or back shroud (Figure 2-7)
- Semi-open design without a front shroud (Figure 2-8)

Figure 2-6
Enclosed Impeller

Figure 2-7
Open Impeller

Figure 2-8
Semi-Open Impeller

The bottom case bearing is normally permanently grease-lubricated; the other bearings are lubricated by the pumped liquid. This design lends itself well to smaller pumps with up to an 18 inch bowl diameter and a 2 inch shaft diameter. However, the following limitations must be noted:

- Taper collet mounting of impellers, depending on shaft diameter and material combinations, is only recommended for handling liquids from 0°F to 200°F. The same temperature limitation applies to semi-open impellers.
- A grease-lubricated bottom bearing is only recommended for ambient temperature water service; otherwise, lubrication with the pumped liquid should be used with filtration when required.

For larger pumps or when handling hot or cold liquids, the following design practices are recommended:

- The bowl joints should be flanged and bolted. Metal-to-metal joints with O-rings for sealing are preferred.
- Male and female registers between mating flanges with strict tolerances for register fits, concentricity, parallelism, and perpendicularity required to maintain pump internal alignment (that is, centerline) should be used.
- The impellers should be mounted on the shaft with key drives, and secured axially with split rings and thrust collars.

When selecting closed versus semi-open impellers, the following must be noted:

- Closed impellers should always be used for handling hot or cryogenic liquids.
- Closed impellers exhibit lower down thrust when in the axially unbalanced configuration.
- Closed impellers are easier to assemble for large pumps with more than three stages.
- Semi-open impellers are more efficient due to the elimination of disc friction from the front shroud.
- Efficiency loss due to wear on the semi-open impeller vanes can be regained by adjusting the impeller setting at the adjustable pump-to-driver coupling.
- When semi-open impellers are axially balanced to reduce axial thrust, down thrust can be maintained over the full operating range. This prevents shaft whip from up thrust with associated bearing wear.
- Semi-open impellers can readily be hard surfaced for erosion protection.
- Semi-open impellers are less likely to seize when handling sand or foreign material.

Key Technical Point

Adequate clearance between impeller and suction liner/shroud must be maintained to allow for thermal growth and shaft stretch due to thrust loads, and rotor and fluid weight.

2.2.2 The Column Assembly

The column assembly consists of three primary components:

- The outer column, which serves as the conduit and pressure boundary for the flow from the bowl assembly.
- The column shaft, or line shaft, which transmits torque from the driver to the impellers on the pump shaft and carries the hydraulic thrust from the bowl assembly to the thrust bearing in the head/driver assembly.
- The shaft enclosing tube, or inner column, which houses the column bearings, serves as a conduit for bearing lubrication, and protects the shafting. The type of liquid that is pumped determines whether or not a shaft enclosing tube is required.

Outer Column

The simplest outer column construction consists of pipe sections, normally 10 feet in length, with straight thread on both ends and joined with pipe couplings. This design is commonly used for column diameters that are 12 inches or less. For handling relatively clean liquids, bearings of a rubber compound are located in housings with a three-legged or four-legged spider and a mounting ring, which is centered within the column coupling and clamped between the column pipe ends. Metal-to-metal contact with O-rings provides an adequate liquid seal. This configuration is often referred to as *open line shaft construction*.

For larger column sizes, or where corrosive or other properties of the pumped liquid make threaded joints undesirable, flanged column joints are used. Registered fits are used to provide alignment, with O-ring gaskets for sealing, because they provide metal-to-metal flange face contact. Bearings that are housed in spiders can be clamped between column faces or provided with a design that incorporates spiders welded (stress relief required) into the outer column, with the flange register and spider bore machined in the same operation. When a shaft enclosing tube is required, the larger column sizes require a radial stabilizing spider clamped or welded to the column with a snug, machined fit around the enclosing tube. A tensioning device is required at the top end of the threaded enclosing tube to stretch and stabilize the enclosing tubes.

Column Shaft

Shaft sections that are less than 4 inches in diameter are commonly joined by threaded couplings, which transmit both torque and axial thrust. For pumps with this construction, it is imperative that drivers be checked for correct rotation before being connected to the pump. Threaded couplings run in the reverse rotation will unscrew and the resulting jacking motion can cause serious pump and/or motor damage. However, reverse rotation from backflow through the impellers will not cause the couplings to unscrew because the direction of shaft torque remains the same as for normal operation.

For column shafts that are 4 inches and larger in diameter, a keyed sleeve coupling should be used for transmitting torque. Axial thrust should be carried through split rings retained by thrust or retaining collars. Flanged bearings, both in the column and the bowl assembly, are recommended for bores that are 4 inches and larger.

Shaft Enclosing Tube

When abrasives or corrosive properties prohibit the pumped liquid from being used for flushing and lubricating the column bearings, the bearings should be fitted inside a shaft enclosing tube. The bearing holders can be threaded on the outside diameter (OD) and serve as joiners for the 5 foot long enclosing tube sections. Bearing alignment is provided by placing the enclosing tube assembly in tension through a threaded tensioning device located in the discharge head. The enclosing tube is stabilized within the outer column by spider-type supports, the hub of which fits tightly around the enclosing tube with the legs attached to the inside of the outer column. The desired bearing lubrication, which can be oil, grease, clean water, or any fluid that is compatible with the pumped liquid, is injected at the top end of the enclosing tube assembly.

To overcome the increasing number of assembly and handling problems that can occur with an increase in column size, an enclosing tube integrally welded with ribs at the top and bottom of the outer column is a preferred design. Alignment is assured with simultaneous machining of the registered column fits, the inner column joints, with slip fit and O-rings, and the bearing seats. Furthermore, the need for a tensioning device in the discharge head can be eliminated with proper axial restraints.

2.2.3 Discharge Head Assembly

The discharge head of a vertical pump functions to efficiently change the direction of flow of the liquid from the vertical axis to the horizontal axis of the discharge pipe. The discharge head can accommodate all modes of drivers, including hollow shaft and solid shaft motors, right angle gears, diesel engines, and vertical steam turbines.

The discharge head is designed to:

- Support the pump’s stationary parts
- Support the suspended, liquid-filled weight of the pumping unit
- House the shaft sealing device
- Provide support for the driver
- Be the main structural support positioning all components, rotating and stationary

The shaft sealing device must be suitable for the maximum pressure to which the pump can be subjected, which would most likely be the sum of the pump shutoff (zero flow) pressure from the performance curve and maximum suction pressure. The sealing device is located in a stuffing box that can either be placed in the discharge stream for flushing or mounted externally for cooling and flushing. The actual sealing can be done with packing or a mechanical seal. A pressure breakdown bushing, with bleed-back to pump suction, can also be included in the sealing device for high-pressure applications.

The standard drive coupling for vertical pumps with solid shaft drivers is a rigid design, capable of transmitting the maximum torque from the driver and the combined axial force from hydraulic thrust plus rotating element weight. To permit adjustment of the impeller setting within the bowls, the coupling typically incorporates a disc (adjusting nut) threaded on to the top end of the column shaft and clamped between the two coupling halves. For drivers such as a hollow shaft motor with limited thrust carrying capability, a thrust bearing might be incorporated into the discharge head design. For pumps using hollow shaft drivers, torque is transmitted to the top column shaft or head shaft through a coupling at the top of the motor; impeller adjustment is made by a nut seated on top of the clutch.

Most vertical pumps are a structurally flexible design. This means that the structural natural frequency of the first order is the same order of magnitude as the operating speed. A careful analysis must therefore be made of the discharge head design, in relation to its foundation and the connected driver and system piping, to ensure that the combined natural frequency does not coincide with the pump operating speed. Similarly, deflection calculations for the unit must be made to ensure that pump alignment is not impaired when it is subjected to nozzle loads and the liquid filled weight.

Discharge heads are available with above ground or below ground discharge (see Figures 2-9 and 2-10) and are constructed of cast iron or fabricated steel. Cast iron discharge heads are used for low-pressure service that does not exceed 175 psig. For pressures that exceed 175 psig, or for applications requiring alloy construction, fabricated discharge heads are used.

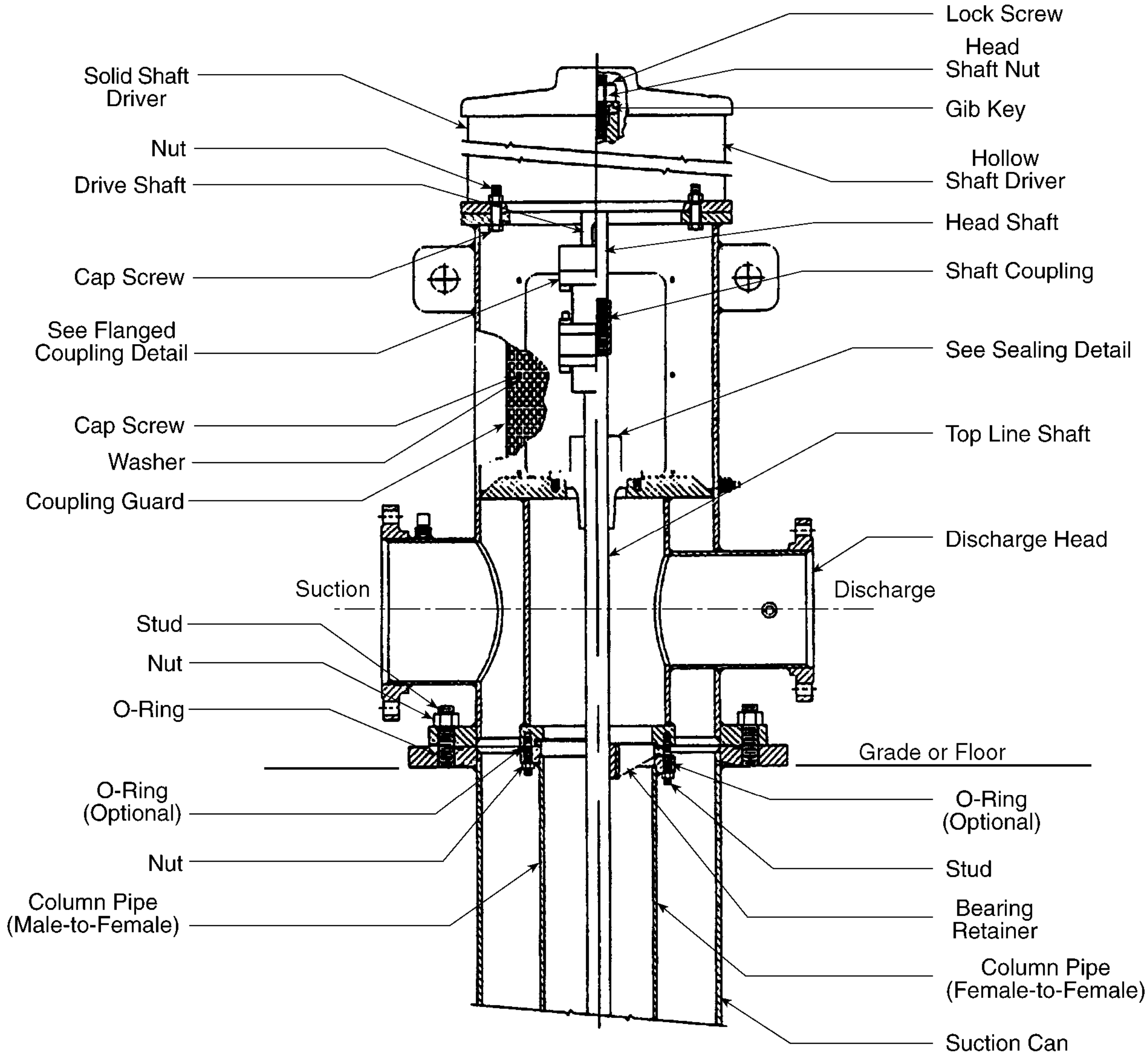


Figure 2-9
Fabricated Discharge Heads with Above Grade Suction

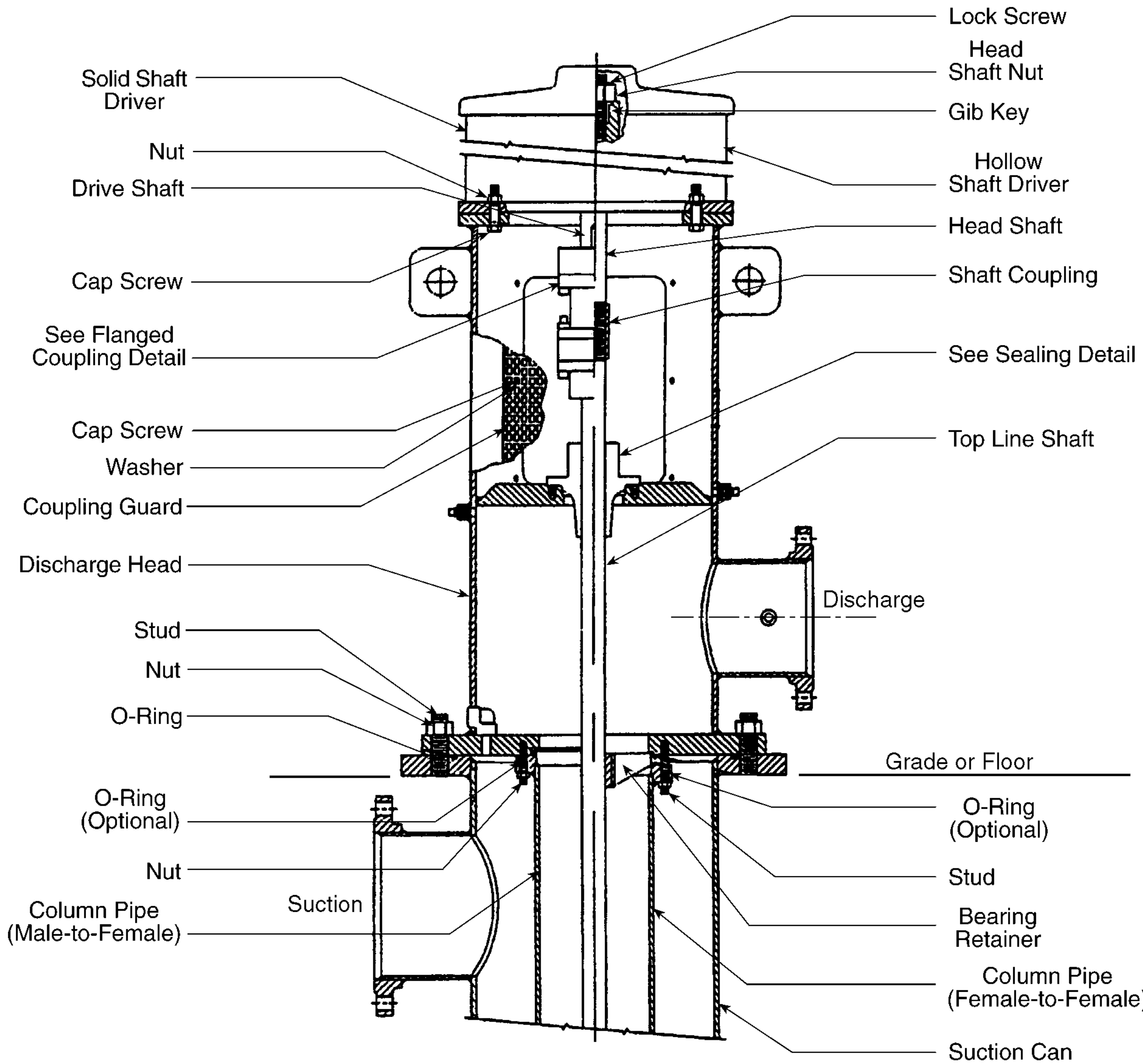


Figure 2-10
Fabricated Discharge Head with Below Grade Suction

2.3 Pump Component Descriptions

2.3.1 Impellers

Impellers are classified according to the major direction of flow in reference to the axis of rotation of the impeller. Centrifugal pumps can have radial-flow impellers (Figure 2-11), axial-flow impellers (Figure 2-12), or mixed-flow impellers (Figure 2-13). Mixed-flow impellers combine the principles of design of both radial-flow and axial-flow impellers. Impellers are further classified as either single or double suction. Single-suction impellers (Figure 2-14) have a single inlet on one side, while double-suction impellers (Figure 2-15) have symmetrical inlets on both sides.

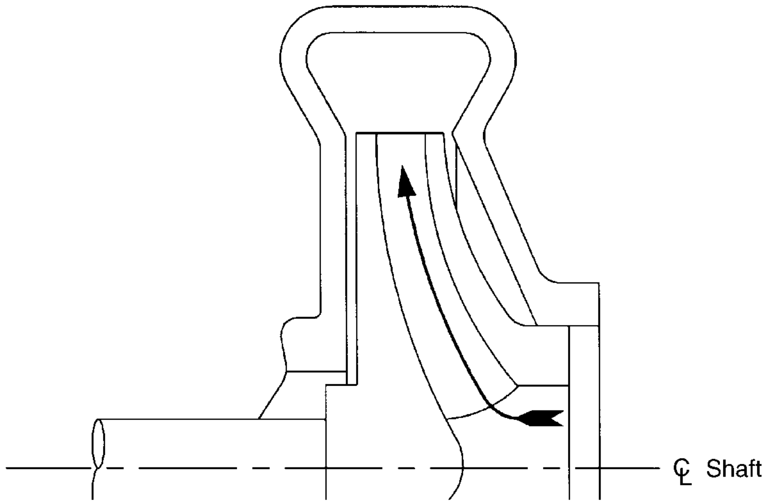


Figure 2-11
Radial-Flow Impeller

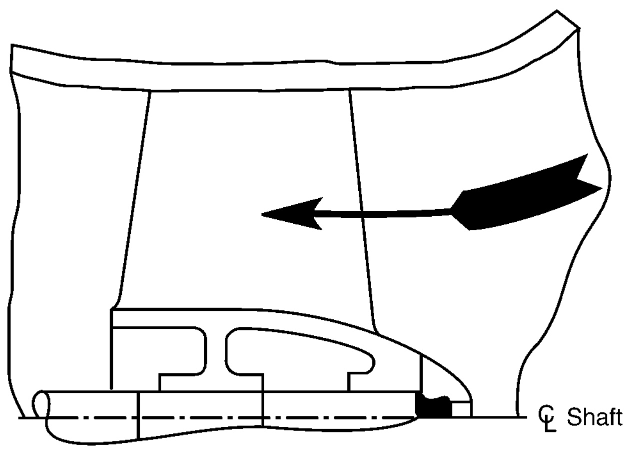


Figure 2-12
Axial-Flow Impeller

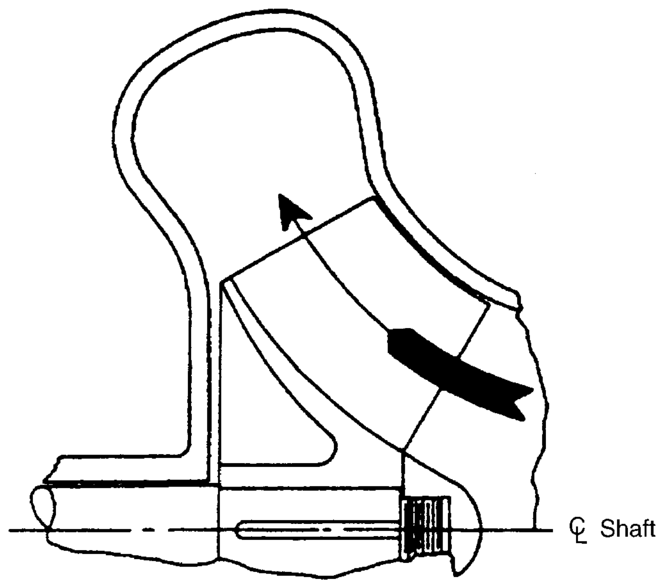


Figure 2-13
Mixed-Flow Impeller

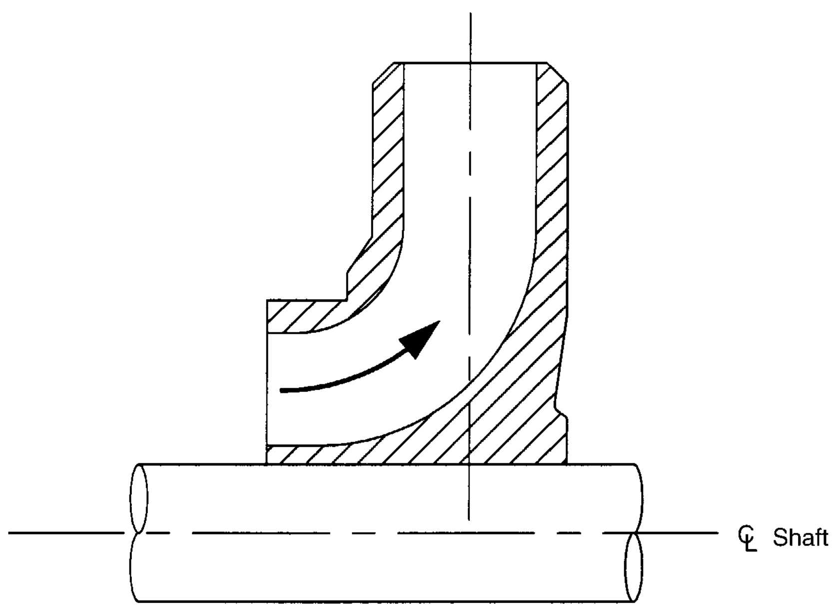


Figure 2-14
Single-Suction Impeller

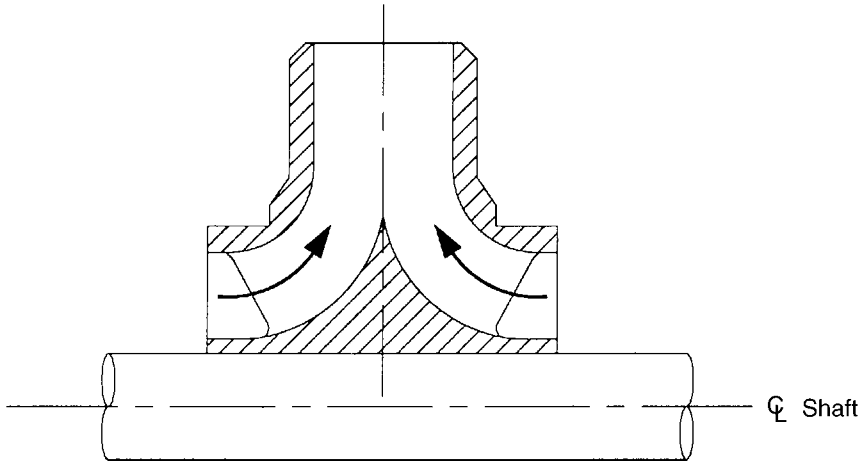


Figure 2-15
Double-Suction Impeller

Impellers can also be classified by the shape of their vanes (see Figure 2-16):

- Radial-Vane impeller
- Francis-Vane or screw-vane impeller
- Mixed-Flow impeller
- Axial-Flow propeller or impeller

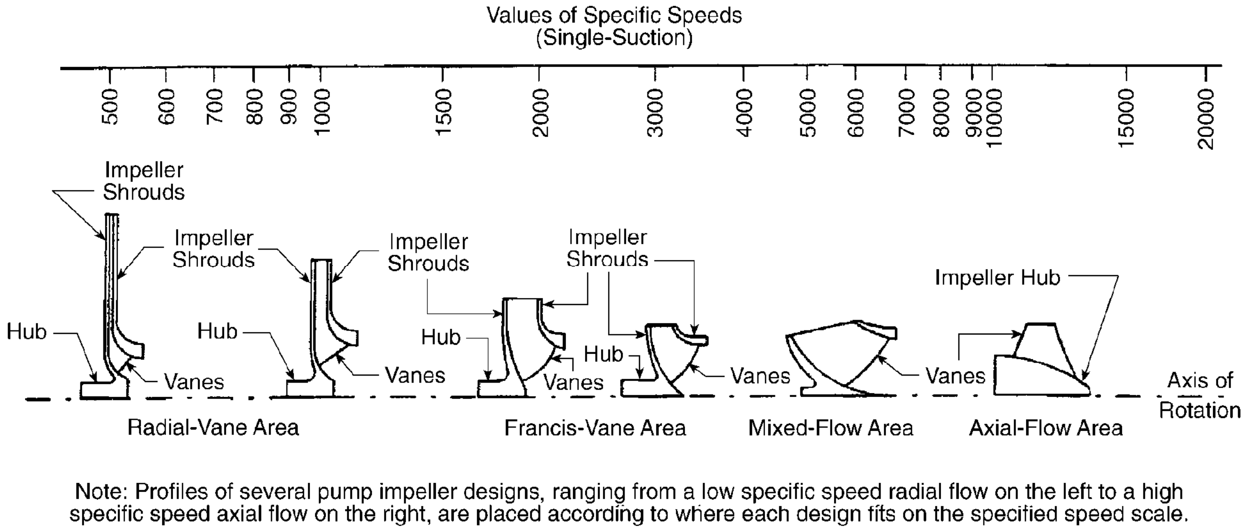


Figure 2-16
Comparison of Pump Profiles

There are four primary methods for impeller-to-shaft attachment (see Figure 2-17):

- Drive collet
- Lock collet
- Split ring and retaining ring (snap ring)
- Lock collar

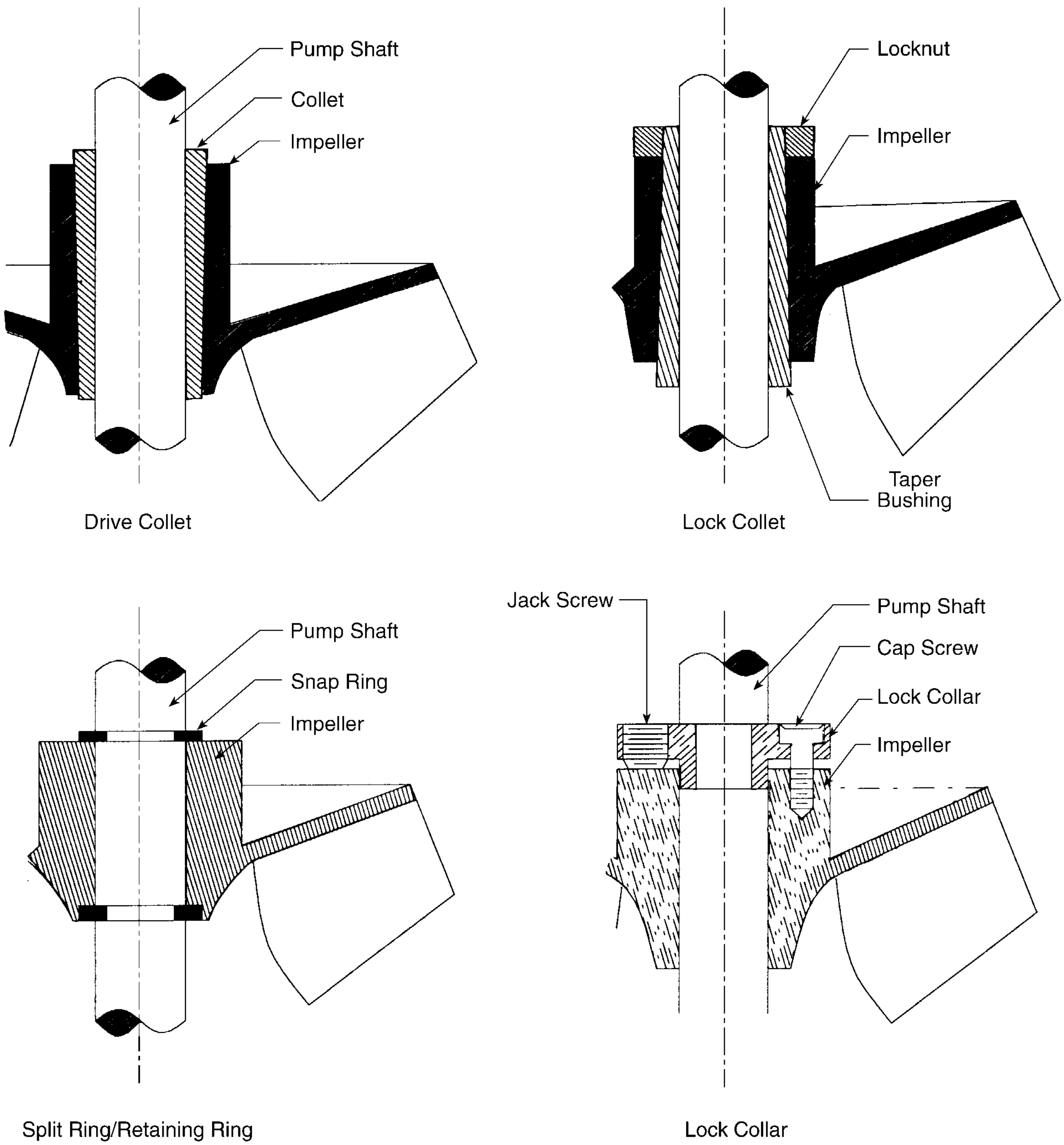


Figure 2-17
Impeller-to-Shaft Attachment Methods

The drive collet is the simplest of the four arrangements and is typically used on small units. It is essentially a tapered collet that is driven between the shaft and the tapered fit of the impeller bore. The drive collet assembly is not a controllable operation. The quality of the attachment depends on the environment and the ability of the individual assembling the pump. If the collet is not tight enough, or if dirt or oil is present in the impeller or shaft fits, the impeller will slip under load and pump performance will be adversely affected. The one advantage of the drive collet is the lower initial cost.

The lock collet is similar to the drive collet except that a threaded nut is added on the small end of the collet. This results in the collet having a tighter fit and higher reliability. Although the lock collet has a tighter fit than the drive collet, it still can have assembly problems and the possibility of slippage under load.

The next arrangement uses a split ring/retaining ring combination to secure the impeller on the shaft. The split ring transmits axial down thrust, and the retaining ring transmits axial up thrust. Torque is transmitted by a key in the shaft. This method is substantially more reliable than the drive and lock collet. The disadvantage of this design is that it requires two registered fits to ensure tightness between the impeller and the shaft. Any manufacturing error will result in a loose fit, causing the impeller to *chatter* during operation and increasing overall pump vibration.

The last attachment method is a split collar device. The collar fits in a shaft groove and is attached to the impeller hub by cap screws. Once again, torque is transmitted by a key in the shaft. This design accurately places the impeller at the proper location and rigidly holds the impeller under all operating conditions, including reverse rotation. This design is the recommended method for impeller attachment.

2.3.2 Pump Bowl/Case

The liquid leaving the impeller has high kinetic energy and discharges into the pump bowl. The bowl assembly contains axially disposed diffuser vanes to efficiently transform this kinetic energy into pressure energy. The bowl can be fitted with wear rings and have one or more bearings. The functions of the bowl are to:

- Guide flow
- Limit energy recovery
- Connect the suction bell and column
- Support the pump bearings

See Figures 2-18 and 2-19 for examples of the suction bell and bowl.

Figure 2-18
Typical Suction Bell and Bowl/Case Arrangement

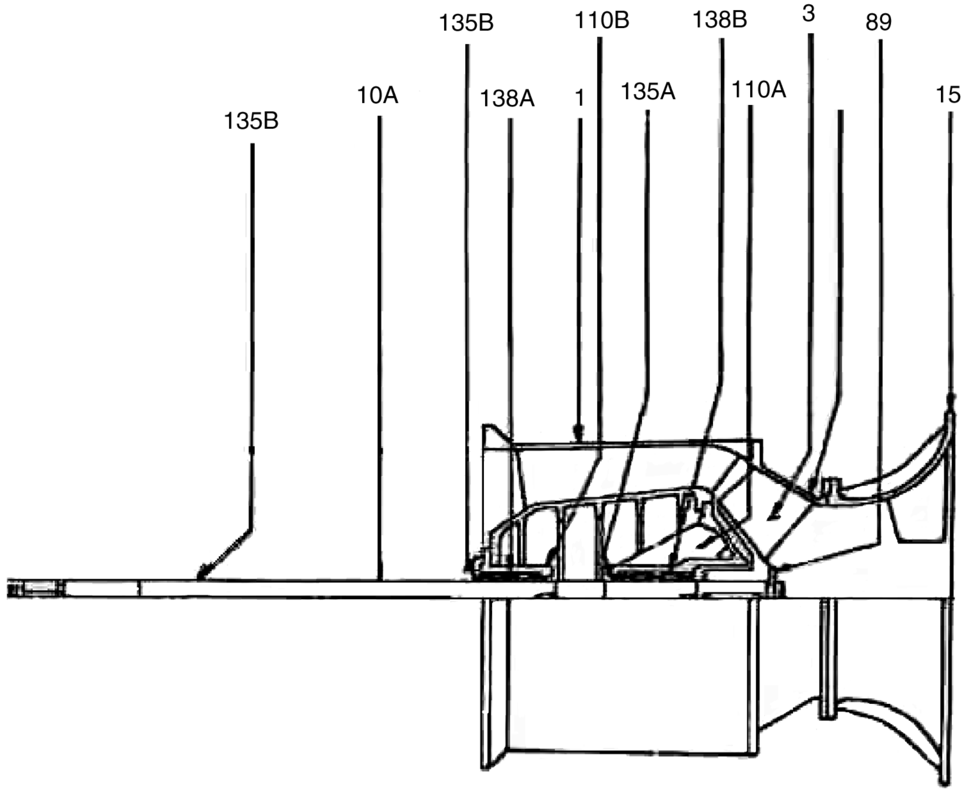


Figure 2-19
Vertical Pump Wet End Cross-Section (Item 1: Bowl/Case, Item 3: Impeller, Item 89: Shroud/Liner, and Item 15: Suction Bell)

2.3.3 Wear Rings

Wear rings provide an easy and economically renewable leakage joint between the impeller and the casing. A leakage joint without renewable parts is shown in Figure 2-20. Restoring the original clearances in this design requires buildup of the worn areas through welding or other means, followed by machining to the design clearances, or the purchase of new components. The option does exist, when repairs are required, for the impeller or bowl or both to be modified for renewable wear rings. This could be more economical than welding or replacing existing parts and eliminates the problem for the next repair.

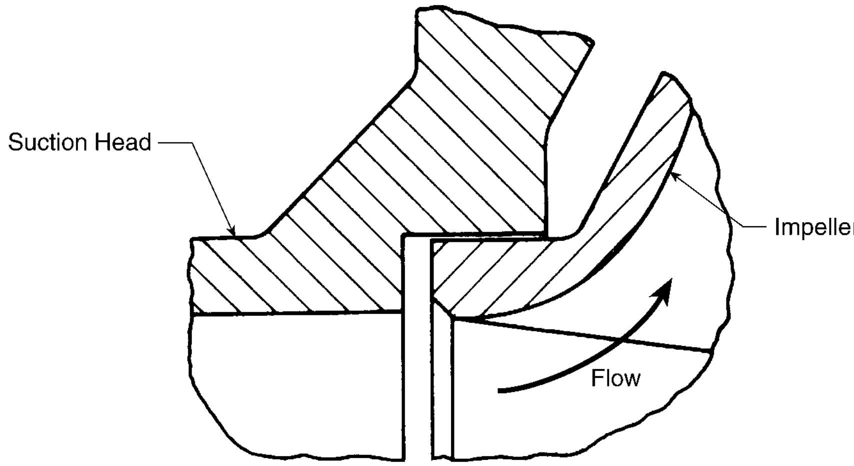


Figure 2-20
Plain Flat Leakage Joint with No Renewable Parts

Key O&M Cost Point

Provisions for renewable wear rings might increase the initial cost but can improve the pump refurbishment process during the life of the pump, thereby reducing total maintenance costs.

Wear rings are available for enclosed impellers and bowls. This provides a way to reestablish initial operating clearances and recover pump efficiency at the lowest cost. For longer life, wear ring improvements such as material upgrades, hardening by heat treatment, hard surfacing, and hardness differential between wear surfaces are also available. Impeller and bowl wear rings might both be options offered by a pump manufacturer but, typically, renewable bowl wear rings are provided, while impeller wear rings are offered as an option (see Figures 2-21 and 2-22).

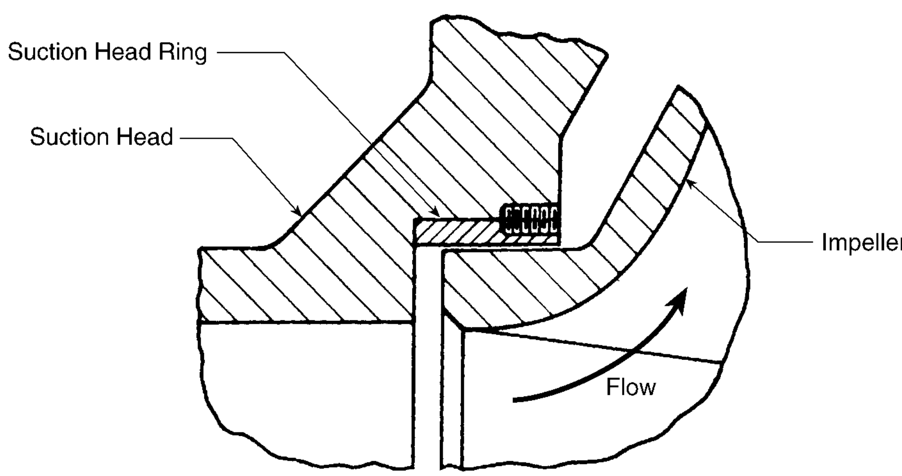


Figure 2-21
Single Flat-Casing-Ring Construction

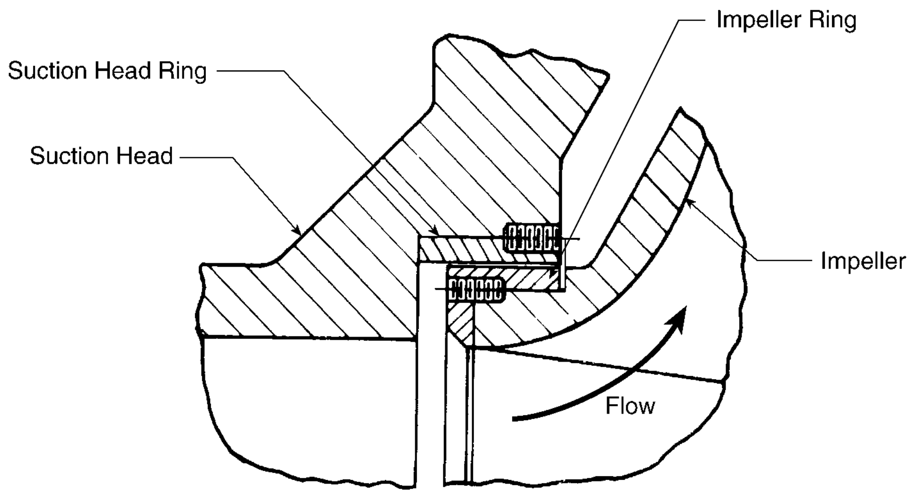


Figure 2-22
Double Flat-Ring Construction

2.3.4 Column

The column assembly is the spool piece connecting the bowl assembly to the discharge head; it can consist of one or more sections. Column sections in larger pumps are flanged and provided with rabbet fits for positive alignment. Column lengths depend on shaft construction type (open or closed). For open line construction, the column length matches the bearing span (to enable bearing support through the use of a spider located at the flange) unless this length exceeds the machining capability of the manufacturer and/or the handling capacity in the field.

2.3.5 Shaft

The basic function of a pump shaft is to transmit the torque produced during startup and operation of the centrifugal pump, while supporting the impeller(s) and other rotating components. The shaft must perform this function with a deflection that is less than the minimum clearance between the rotating and stationary parts. Pump shafts can be rigid or flexible. A rigid shaft design for a centrifugal pump is one with an operating speed that is lower than its first critical speed. A flexible shaft is one with an operating speed that is higher than its first critical speed.

For most vertical pumps made in the United States, axial thrust is transmitted to the motor thrust bearing through the line shaft. Shafting is precision machined for trueness, balance, straightness, and surface finish to minimize shaft vibration and maximize bearing life.

2.3.6 Shaft Sleeves

Pump shafts are usually protected from erosion, corrosion, and wear at stuffing boxes and bearings by renewable sleeves. The most common shaft sleeve function is to protect the shaft from wear at the stuffing box, particularly where packing has been used, and at the line shaft, bowl, and suction head bearings. Shaft sleeves are typically keyed to the shaft with a slight clearance fit between the ID of the sleeve and the OD of the shaft.

2.3.7 Bearings

Bearings in centrifugal pumps maintain the shaft and rotating elements in correct alignment with the stationary components under the action of radial and axial loads. The following three basic types of bearings are used in pumps:

- Sleeve bearings
- Antifriction ball bearings
- Thrust bearings

Sleeve bearings can accommodate only radial loads. Antifriction ball bearings accommodate radial loads, and some types can also accommodate axial thrust loads. Some special types of bearings are designed to accommodate only thrust loads.

Vertical pumps are typically equipped with sleeve bearings at all locations at which the shaft passes through the pump discharge head, at each bowl, and within the pump column sections (depending on the shaft length). Vertical turbine pumps often include an additional bearing located at the bottom of the shaft in the suction head. Sleeve bearings are typically designed to be lubricated by the pumped fluid.

All vertical pumps require a thrust bearing to support the weight of the shaft, impellers, and the unbalanced hydraulic thrust. In U.S. designs, this bearing is usually located in the motor, while most non-U.S. pump manufacturers incorporate the thrust bearing into the pump directly under the driver.

Sleeve bearings consist of a bushing mounted inside a rigid cylindrical sleeve that provides dimensional stability for the bushing. Bushings for condensate pump applications are typically manufactured of a low-friction, longwearing material such as carbon, graphalloy, or nitrite; the shaft sleeve is typically manufactured of stainless steel. Most other applications use bronze bearings, rubber, and marine-style bearings. Gaining in popularity is synthetic polymer alloy bearing materials (for example, those manufactured by Thordon Bearings, Inc.).

2.3.8 Stuffing Box

The stuffing box has the function of protecting the pump from leakage at the point where the shaft passes through the pump case. If the pump operates with a suction lift and the pressure at the interior stuffing box end is below atmospheric (as is the case in most condensate pump applications), the stuffing box must prevent air leakage into the pump. If this pressure is above atmospheric, the stuffing box prevents liquid leakage out of the pump. For vertical pumps, the stuffing box is located in the discharge head for above ground applications and in the motor support for below ground applications. Two stuffing box sealing methods are typically employed: packing and mechanical seals.

2.3.8.1 Packing

For packing, the stuffing box is usually a cylindrical recess that can accommodate a number of packing rings around the shaft or shaft sleeve (see Figures 2-23 and 2-24). If sealing of the stuffing box is required, a lantern ring or seal cage can be added between equal numbers of packing rings (see Figure 2-25) to allow for injection of seal water. Compression of the packing by a packing gland provides the desired fit on the shaft or shaft sleeve. The sealing liquid supplied to the stuffing box usually comes from the pump discharge piping, or an external clean water source.

Stuffing box packing functions as a pressure breakdown device and must be somewhat elastic so that it can be adjusted for proper operation. Packing must also absorb energy without failing or causing damage to the shaft or shaft sleeve. It comes in numerous forms and materials and is supplied in continuous coils of square cross-section or in preformed die-molded rings. The die-molded packing rings are preferable because they are available in exact sizes and in sets. This ensures an exact fit to the shaft or shaft sleeve and to the stuffing box bore, while also ensuring equal packing density throughout the stuffing box. Common packing materials include a graphite impregnated Teflon coating or aromatic polymide.

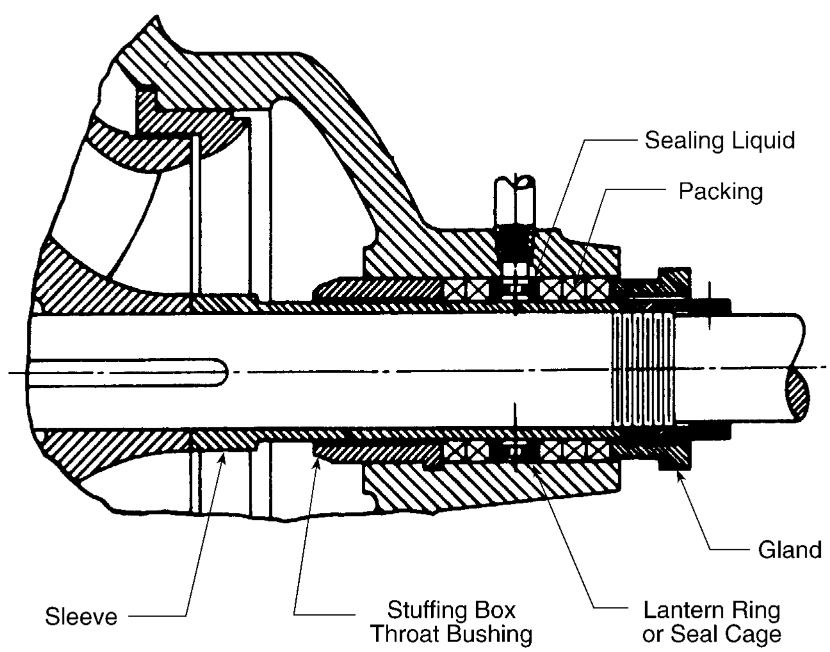


Figure 2-23
Conventional Stuffing Box with Throat Bushing

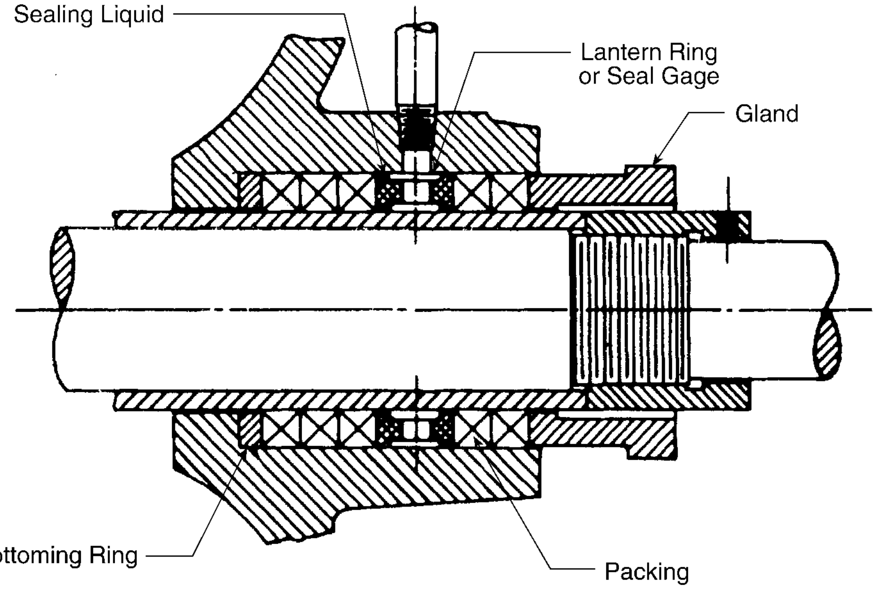


Figure 2-24
Conventional Stuffing Box with Bottoming Ring

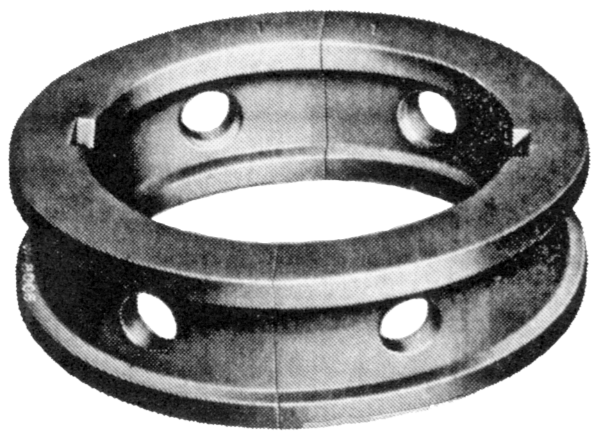


Figure 2-25
Lantern Ring (also called Seal Cage)

2.3.8.2 Mechanical Seals

A mechanical seal is a sealing device that forms a running seal between rotating and stationary parts. The wide variety of styles and designs, together with extensive experience, allows the use of seals on most pump applications. A mechanical seal must seal at three points in the following ways:

- Static seal between the stationary part and the housing
- Static seal between the rotary part and the shaft
- Dynamic seal between the rotating and stationary seal faces

Note: Additional information on mechanical seals can be found in EPRI report 1000987, *Mechanical Seal Maintenance and Application Guide*.

Figure 2-26 shows a basic seal with these components:

- The stationary seal part positioned in the housing with preload on the O-ring to effect sealing and prevent rotation.
- The rotating seal part positioned on the shaft. The O-ring seals between the rotating part and the shaft, and provides resiliency.
- The mating faces precision lapped for a flatness of 3 light bands and a surface finish of 5 micro inches (0.1 micrometer).
- The spring assembly, which rotates with the shaft and provides a force to keep the mating faces together during periods of shutdown or lack of hydraulic pressure.
- The driving member, which positions the spring assembly and the rotating face and provides the positive drive between the shaft and other rotating parts.

As wear occurs between the mating faces, the rotating face must move along the shaft to maintain contact with the stationary face. The O-ring must be free to move.

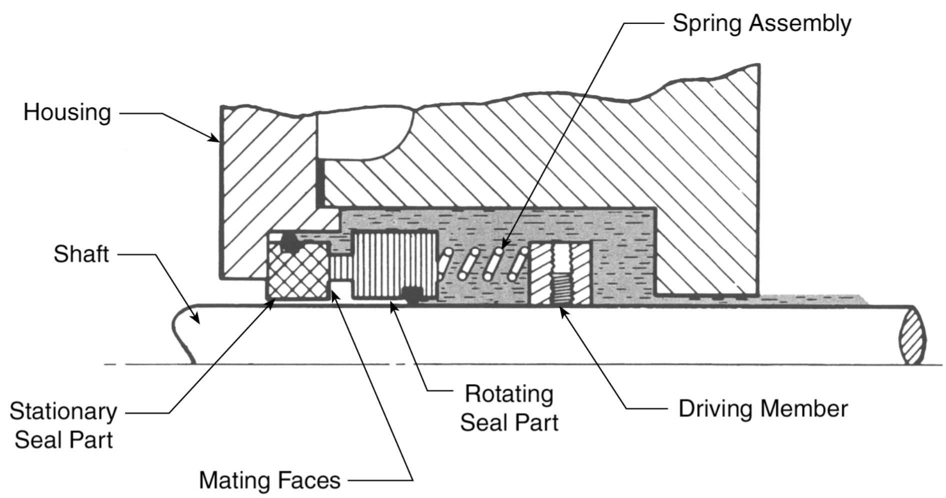


Figure 2-26
Basic Mechanical Seal

2.3.8.3 Comparison of Packing and Mechanical Seals

When properly applied, packing can be a simpler, more economical, more reliable, and easier maintenance choice than mechanical seals. The most important concept regarding packing is that leakage is required across the packing to provide lubrication and cooling. Most packing problems occur when maintenance personnel adjust the gland tightness to reduce or eliminate packing leakage. Packing suppliers recommend leakage rates ranging from 30 to 60 drops per minute.

When properly applied, mechanical seals can offer the following advantages over conventional packing:

- Reduced friction and power losses
- Zero or limited visual leakage of product
- Elimination of shaft or sleeve wear
- Reduced maintenance
- Ability to seal higher pressures

However, the disadvantages of mechanical seals include the following:

- Higher initial and replacement costs
- More difficult installation
- Sensitivity to vibration, shaft alignment, and wear of other pump components
- A more catastrophic failure mode than packing

Because mechanical seals are less forgiving of wear, vibration, and misalignment of the pump components as compared to packing, repeated failures of mechanical seals might be indicative of problems elsewhere in the pump.

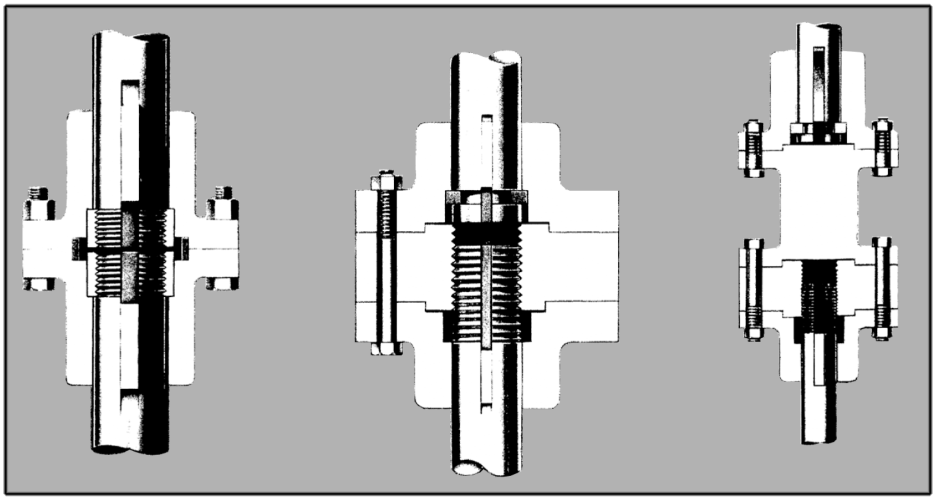
Key Technical Point

If a pump experiences multiple mechanical seal failures, be sure to extend the analysis of the problem beyond the seal itself.

Also, mechanical seal installation must be completed after the pump rotor is installed to account for shaft stretch. The mechanical seal must be relocated every time there is an axial change in shaft position as a result of impeller or shroud wear. All of these characteristics of mechanical seals should be taken into consideration before considering the application of mechanical seals for an existing pump.

2.3.9 Couplings

Coupling types used in pump drive systems can include rigid or flexible couplings. A coupling is used wherever there is a need to connect a prime mover (motor) to a piece of machinery (centrifugal pump). A common application for rigid couplings is in vertical pumps, where the prime mover (motor) is positioned above the pump. Two common types of rigid couplings are used. One type consists of two flanged members, each mounted on one of the connected shafts (see Figure 2-27). A second type is known as the split rigid, which is split along its horizontal centerline (see Figure 2-28). In this case, both machines can use a common thrust bearing, which is usually located in the motor. It should be noted that the use of rigid couplings requires precise alignment of machine bearings because there is no flexibility within the coupling to accommodate misalignment between shafts.



Rigid Flanged Coupling	Adjustable Coupling	Adjustable Spacer Coupling
Used to couple the pump to the vertical hollow shaft driver. Impeller adjustment is performed on the adjusting nut located on top of the motor.	Used for the vertical solid shaft driver. Impeller adjustment is made by using an adjustable plate in the coupling.	Performs the same function as a Type A coupling with the addition of a spacer. The spacer may be removed for mechanical seal maintenance without disturbing the driver.

Figure 2-27
Flanged Coupling Arrangements

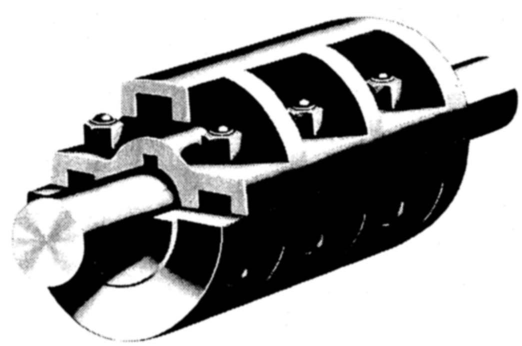


Figure 2-28
Horizontally Split Rigid Coupling

2.3.10 O-Ring Material Considerations

A common O-ring material supplied in balance of plant pumps is ethylene propylene diene monomer (EPDM). This material provides good service but, when used in applications around 350°F or higher, will become hard and brittle with age. Some EPDM compounds can be used at higher temperatures; for example, Parker compound E0962-90, “Kem Therm,” is rated up to 500°F.

Viton O-ring material has a higher service temperature but is not recommended for use with water.

Kalrez is another high-temperature O-ring material. However, there are some reports that this material has contributed to stress corrosion cracking.

An application that sees temperatures around 375°F is in a vertical condensate booster pump head to barrel O-ring fit. To eliminate a concern with O-ring material at this connection, the flanged connection could be modified to accept a spiral-wound gasket.

2.4 Vertical Centrifugal Can-Type Pumps

Vertical centrifugal can-type pumps are the predominant type of pump used in condensate, heater drain, core spray (BWR plants), and RHR (BWR plants) pump applications. These pumps are typically selected from two types: vertical turbine or vertical volute. Both types of vertical pumps are usually motor-driven, wet-suction, multi-stage pumps, which are mounted in a barrel casing or can, with either single-suction or double-suction first-stage impellers (as illustrated in Figures 2-29 and 2-30).

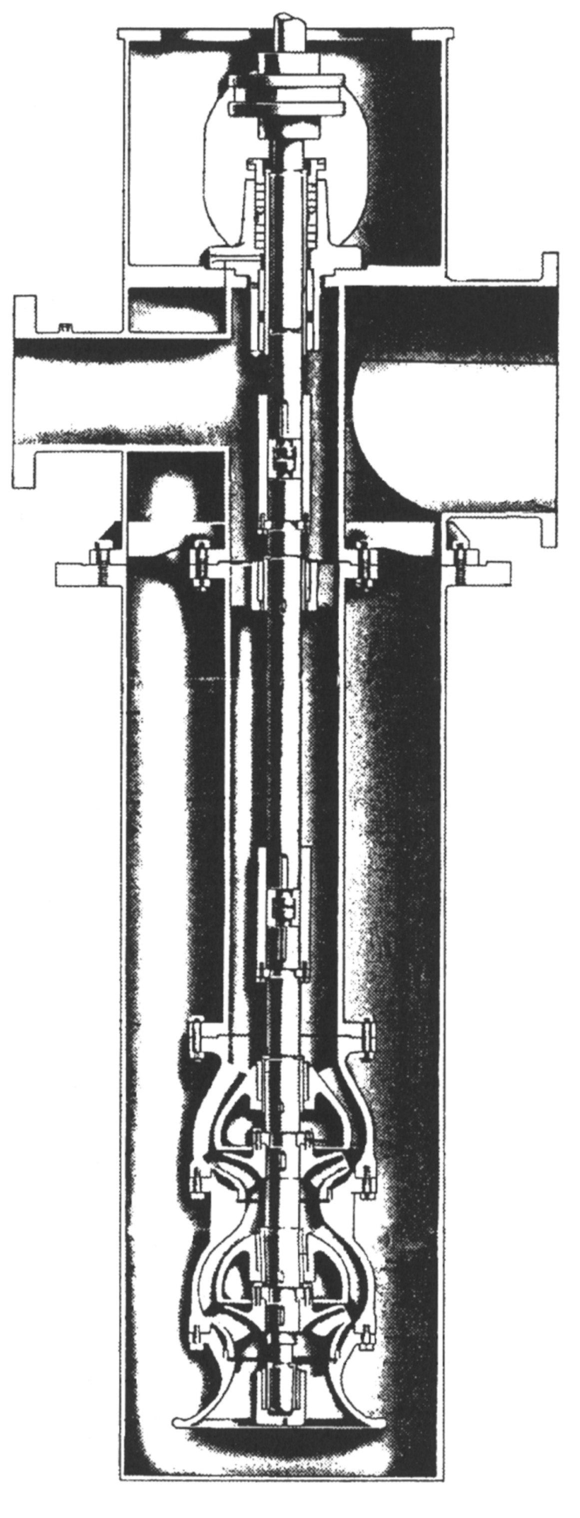


Figure 2-29
Condensate Pump (Vertical, Multi-Stage, Wet-Suction, Can-Type, with Single-Suction First-Stage Impeller)

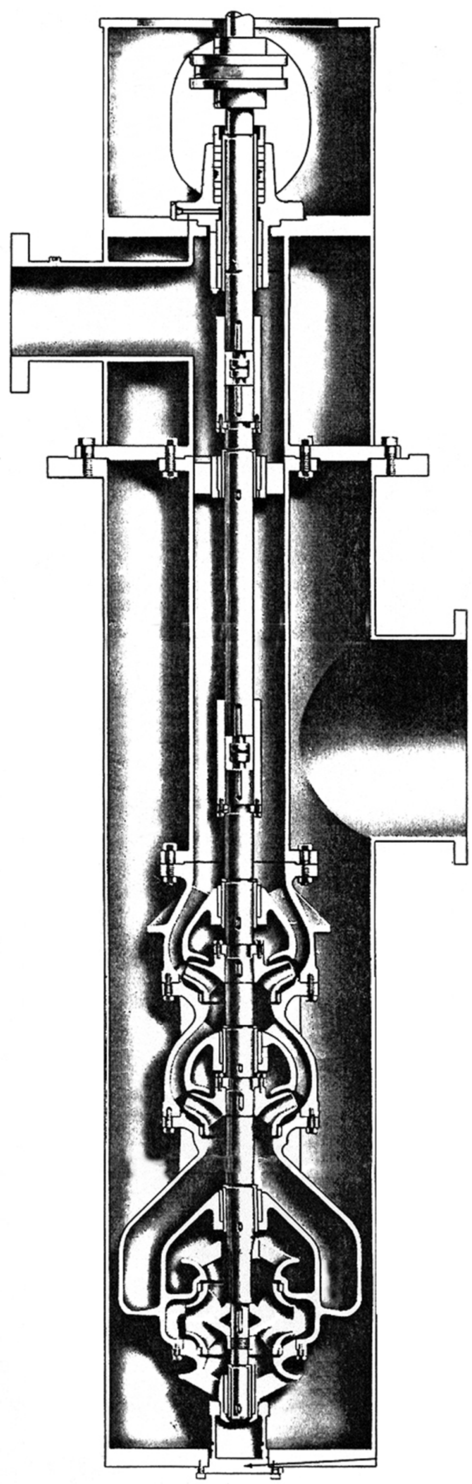


Figure 2-30
Condensate Pump (Vertical, Multi-Stage, Wet-Suction, Can-Type, with Double-Suction First-Stage Impeller)

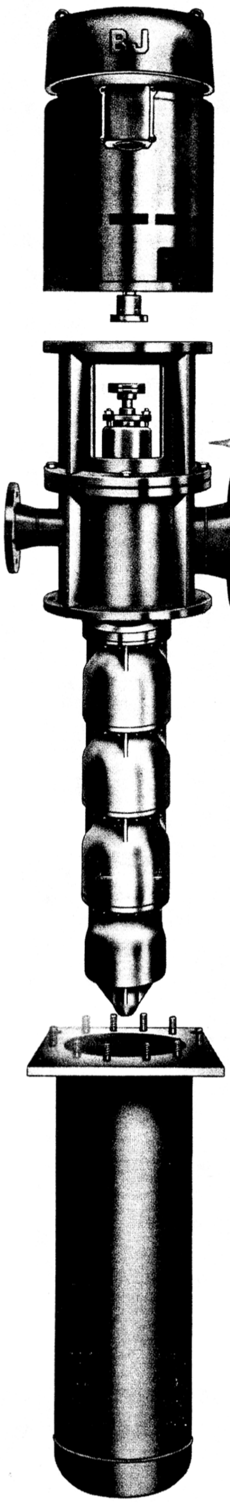
2.4.1 Byron Jackson—VMT Pumps

General Description

The VMT is a vertical self-contained pumping unit, primarily designed for pumping fluids at or near their boiling point. Following this description, see Figures 2-31 and 2-32 for an illustration of the VMT pump and other related information; see Table 2-1 for a listing of materials. The VMT pump was designed for conditions where the available net positive suction head (NPSH) was limited. The pumping unit is mounted in a barrel from which it takes suction. The base of the pump is at ground level, while the length of the barrel is determined by the NPSH or pump requirements.

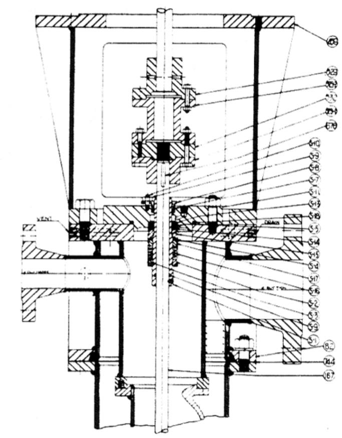
The unit consists of a:

- Complete pump bowl assembly housed within a fabricated steel barrel
- Fabricated steel nozzle head
- Fabricated steel motor barrel flanged at the top to receive the driver
- Flanged, rigid-type, adjustable coupling on the upper end of the pump shaft for direct connection to the driver shaft



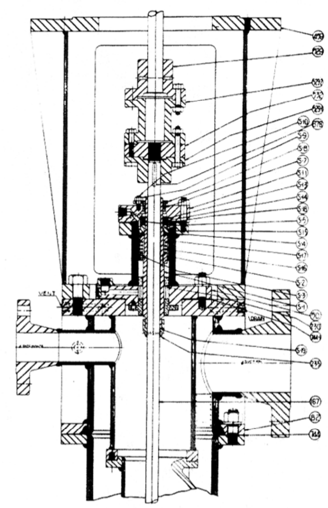
Style "IS"

1. For discharge pressure not exceeding 300 lb. psi
2. For pumping temperatures not exceeding:
   - A. 180°F for water
   - B. All other liquids not exceeding 250°F



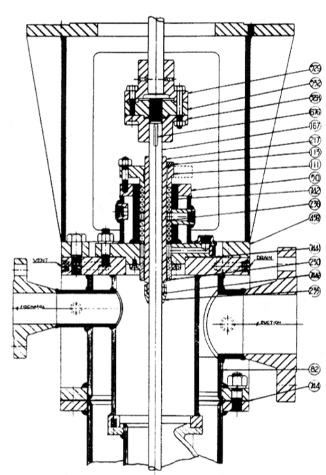
Style "ES"

1. For discharge pressure not exceeding 300 lb. psi
2. For pumping temperatures not exceeding:
   - A. 180°F for water
   - B. All other liquids not exceeding 250°F
3. For water pumping temperatures from 180–250°F
   - A. Style "ES", if cold condensate injection seal is available
   - B. Style "ES" with water-jacketed stuffingbox if cold condensate is not available



Style "EP"

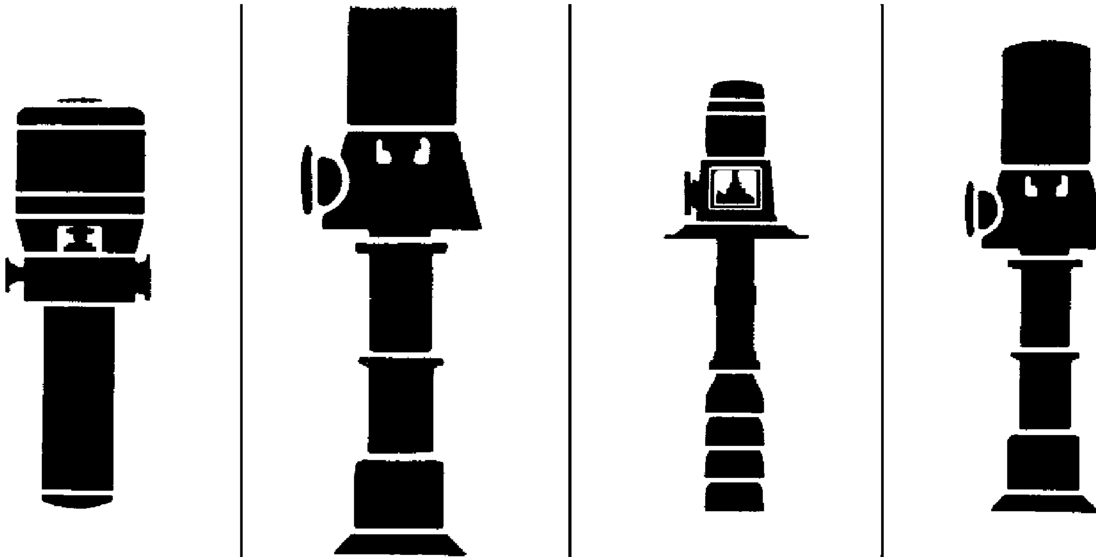
The conventional packing type unit when pumping temperatures exceed 250°F.



Disassembled view of the VMT pump showing the three simple elements that go into its makeup.

The method of shaft sealing depends on temperature and pressure. These arrangements are designed by "IS", "ES", and "EP" as shown above.

Figure 2-31
VMT Pumps—General Assembly and Construction



Vertical Double Case	Vertical Circulating	Vertical Turbine	HQ Circulators
VLT, VHT, VMT, Hydropress	KX, RX, PMR, PHR, HS, Type D, FCDW	VTP, SUMPMASTER	Sizes 15 in., 17 in., and 20 in.
Loading, booster, boiler feed, condensate, pipeline, waterflood.	Cooling tower, dewatering, condenser circulating, river intake, irrigation, molten salt. Thermal dilution.	Water wells, irrigation, booster, sump, circulating, industrial waste, source wells.	Industrial, cooling, processing and booster. Large wells, municpal.
Barrel mounted, high pressure and low NPSH models. Compact. Single or multi-stage. Jacketed, packing or mechanical seal models. Vertical design saves space.	Single or multi-stage. Oil or water lube. Above or underground discharge. Self-priming. Simple installation, small space without special dry pit. High efficiency. Job engineered designs.	Bowl and impeller design and materials to suit any need. Oil or self lube. Pit or surface installation. Extra long bypass bearing and seal combination. Effective sand cap. Closed or semi-open impellers.	Pre-engineered designs, pit or surface installation. Space saving. Vertical solid or hollow shaft, oil or water lube. Multi staging.
		Fully stocked components – Maximum Modular and Part Interchangeability.	
20,000–50,000 gpm	To 130,000 gpm standard (higher on specials)	To 10,000 gpm	To 12,000 gpm
55–2000 feet	To 500 feet or as required	To 800 feet	To 150 per stage

Figure 2-32
Byron Jackson Family of Vertical Pumps

Table 2-1
Standard Materials of Construction

The following specifications apply to standard construction.

Part Name	Material
Pump Case	Cast Iron Class 30
Impellers	Bronze
Shaft	11.5–13 Chrome Steel
Shaft Sleeve	420 H.T. (heat treat)
Shaft Sleeve Nut	Bronze
Bottom Case Bearing	Bronze
Series Case Bearing	Bronze
Case Bolting	SAE 1020
Stuffing Box	Fabricated Steel
Throttle Bushing	Hard Bronze
Cage Ring	Bronze
Gland	Hard Bronze
Outer Barrel	Fabricated Steel
Discharge Head	Fabricated Steel
Discharge Head Gasket	1/16 inch Durabla
Coupling	Steel

Shaft Sealing

The method of shaft sealing depends on temperature and pressure:

- For pumping temperatures not exceeding 180°F maximum for water and 250°F maximum for all other liquids, and with discharge pressures not exceeding 300 psig, an internal mechanical seal is employed. This arrangement is designated as Style I. S. (internal seal).
- When the discharge pressure exceeds 300 psig, a breakdown bushing is required and it necessitates the use of a bolted-on stuffing box. This is designated as Style E.S. (external seal). The temperature limitation for this arrangement, when pumping water, is a maximum of 180°F. For all other liquids, the maximum is 250°F. For water temperatures from 180°F to 250°F, Style E. S. can be used with cold condensate injection to the seal. When cold condensate injection is not available, a water-jacketed stuffing box must be employed.
- The conventional packed-type unit can be furnished when pumping temperatures exceed 250°F. This arrangement is designated as Style E. P. (external packing).

Pump Element

The pumping element includes a complete pump bowl assembly with renewable sleeve bearings, (single or multi-stage, depending entirely on the specified conditions of capacity, total dynamic head, and operating speed), semi-enclosed type impellers, and pump shaft.

Pump Barrel

The barrel, which houses the pumping element, is made of fabricated steel, flanged at the top to receive the nozzle head. This flange is square and serves as the base plate to carry the weight of the complete unit when installed.

Discharge Head

The discharge head is made of fabricated steel of the same diameter as the pump barrel. It is provided with a round flange on the bottom to bolt to the pump barrel, and a flange at the top to close off the suction chamber and to receive either the internal seal or the bolted-on stuffing box. The motor is bolted on top of the discharge head. Suction and discharge nozzles are flanged to 300 lb. A.S.A. raised face flanges, located 180° apart. A pipe extension, flanged and machined for bolting to the pump bowl assembly, is also provided.

Stuffing Box Assembly

Stuffing boxes are furnished for discharge heads incorporating Style E. S. and Style E. P. shaft sealing arrangements. Normally the stuffing box is not water jacketed when a mechanical seal is furnished. However, in a case where pumping water temperatures range from 180°F to 250°F, and cold condensate seal injection is not available, then water jackets are used. With packed type units, the stuffing box is water-jacketed and is provided with in and out connections to the lantern ring. When the Style I. S. is furnished, the stuffing box is not required.

Coupling

The couplings furnished with Style I. S. or Style E. S. discharge heads are flanged, rigid types with an adjustable spacer piece. The spacer allows for the setting of pump lift or running position, and to allow for removal of the seal or seal parts without disconnecting the driver. The spacer piece was eliminated when packed type units were furnished.

Motor

The motor must, in every case, be a vertical solid shaft, designed to carry the total axial thrust but locked to protect against potential reverse thrust.

2.4.2 Ingersoll-Rand, Model APKD Pumps

The Ingersoll-Rand model APKD pumps are specifically designed for extended operation on condenser condensate, heater drain service, or for handling any clean liquid where NPSH is limited or the installation requirements are best suited to a vertical pump. APKD pumps are vertical multi-stage pumping elements mounted in a can (shell). The can forms the intake well. The suction nozzle can be located above or below the pump mounting, depending on the available NPSH. The discharge nozzle is located above the mounting plate. See Figure 2-33 for a cut-away of this pump and Table 2-2 for a listing of materials.

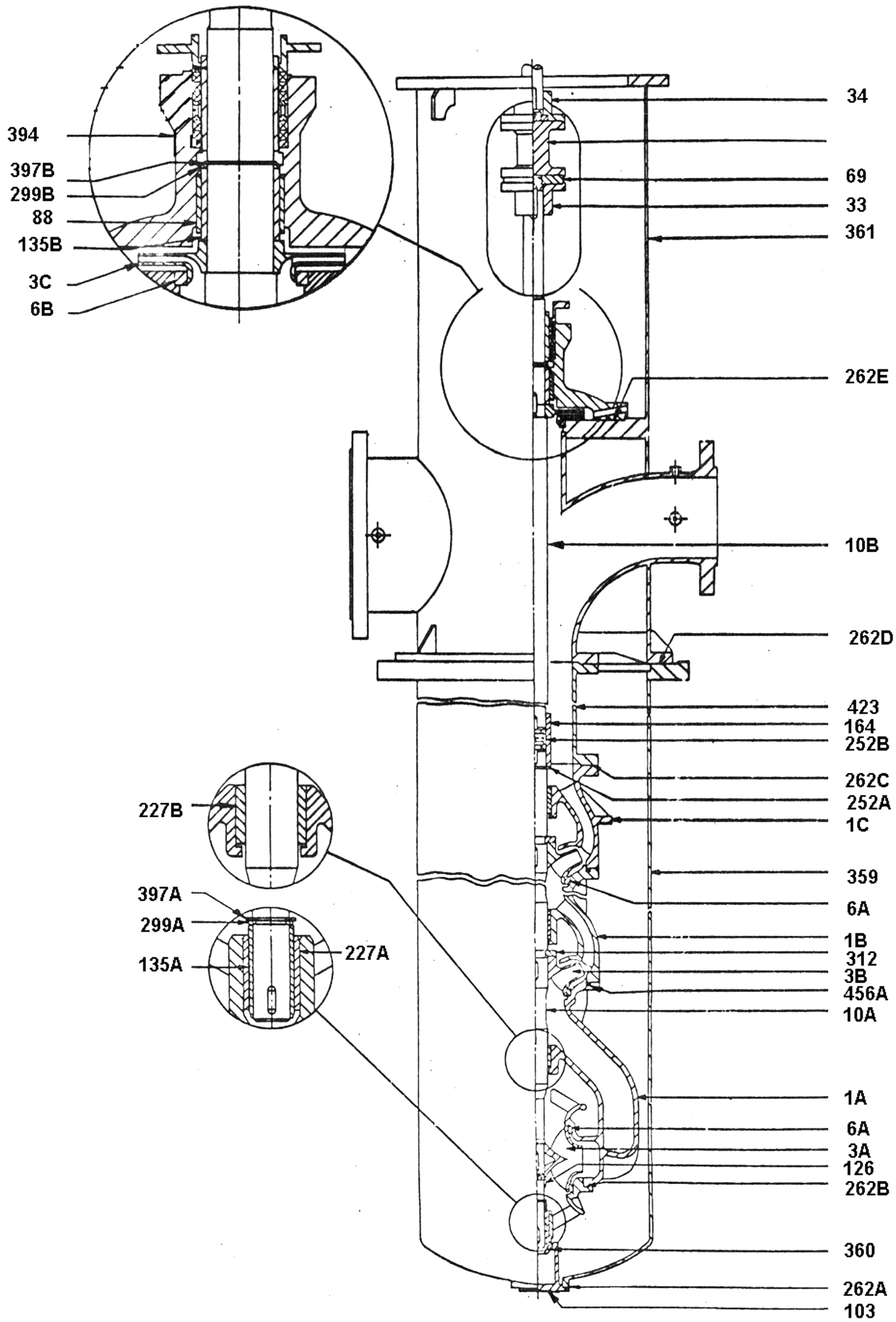


Figure 2-33
Typical APKD Pump Cross-Sectional Drawing

Table 2-2
APKD Materials for Condensate Applications—Maximum Temperature 200°F

<table>
<tr><th></th><th></th><th colspan="3">Material Column</th></tr>
<tr><th>Part No.</th><th>Part Name</th><th>B
pH Range 6.5 to 9.0</th><th>CF
9.0 to 11.0</th><th>KN
6.5 to 11.0</th></tr>
<tr><td>1</td><td>Casing</td><td colspan="3">ASTM A48 class 30B – gray cast iron</td></tr>
<tr><td>3</td><td>Impeller</td><td>ASTM B143 Alloy 922</td><td>ASTM A48 Class 25A, gray iron</td><td>12 Chrome CA-15 Casting</td></tr>
<tr><td>6</td><td>Casing Ring</td><td colspan="2">ASTM A48 class 30B – Gray cast iron</td><td>410 Stainless Steel (SS)</td></tr>
<tr><td>8</td><td>Shaft Sleeve</td><td colspan="3">AISI Type 410 Stainless Steel – Condition K</td></tr>
<tr><td>10A</td><td>Shaft Pumping Element</td><td colspan="3">AISI Type 410 Stainless Steel – Condition B</td></tr>
<tr><td>10B</td><td>Intermediate (when used)</td><td colspan="3">Carbon Steel Bars and Forgings</td></tr>
<tr><td>10C</td><td>Upper</td><td colspan="3">Carbon Steel Bars and Forgings</td></tr>
<tr><td>14</td><td>Seal Cage</td><td colspan="3">ASTM A48 Class 30B – gray cast iron</td></tr>
<tr><td>16</td><td>Gland</td><td colspan="3">ASTM A47, Grade 32510, Malleable Iron</td></tr>
<tr><td>64</td><td>Packing</td><td colspan="2">Cottonite</td><td>J. Crane</td></tr>
<tr><td>88</td><td>Stuffing Box Bushing</td><td colspan="3">AISI Type 410 Stainless Steel – Condition K</td></tr>
<tr><td>126</td><td>Shaft Nut at Suction Head</td><td colspan="3">AISI Type 410 Stainless Steel – Condition B</td></tr>
<tr><td>127</td><td>Shaft Sleeve Nut</td><td>Copper Alloy 932</td><td colspan="2">Carbon Steel Bars and Forgings</td></tr>
<tr><td>135</td><td>Journal Sleeve</td><td colspan="3">AISI Type 410 Stainless Steel – Condition K</td></tr>
<tr><td>164A</td><td>Shaft Column Coupling</td><td colspan="3">Carbon Steel Bars and Forgings</td></tr>
<tr><td>227</td><td>Bearings</td><td colspan="3">Nickel-Filled Carbon</td></tr>
<tr><td>252</td><td>Split Ring at Impeller</td><td colspan="3">AISI Type 410 Stainless Steel</td></tr>
<tr><td>262</td><td>Gaskets – Flat</td><td colspan="3">Hydroil</td></tr>
<tr><td>264</td><td>Stuffing Box Extension</td><td colspan="3">ASTM A48 Class 30B – Gray cast iron</td></tr>
<tr><td>312</td><td>Lock Collar</td><td colspan="3">AISI Type 410 Stainless Steel</td></tr>
<tr><td>359</td><td>Shell</td><td colspan="3">766 or 667</td></tr>
<tr><td>360</td><td>Suction Head</td><td colspan="3">ASTM A48 Class 30B – Gray cast iron</td></tr>
<tr><td>361</td><td>Discharge Head</td><td colspan="3">ASTM A283, Grade C or ASTM A36 Carbon Steel</td></tr>
<tr><td>423</td><td>Outer Column (when used)</td><td colspan="3">ASTM A283, Grade C or ASTM A36 Carbon Steel</td></tr>
<tr><td>456</td><td>O-Ring Gaskets</td><td colspan="2">BUNA-N</td><td>EPR</td></tr>
</table>

Table 2-2 (cont.)
APKD Materials for Condensate Applications—Maximum Temperature 200°F

Part No.	Bolting and Miscellaneous Parts	B	CF	KN
	Impeller Lock Collar	ANSI B18.3 Alloy Steel (N)		
	Pressure Bolting	ASTM A193, Grade B7, Alloy Steel (B, N)		
	Pressure Nuts	ASTM A194, Grade 2H, Carbon Steel Nuts (N)		
	Keys	ASTM A582 Type 416 SS Condition H		
48	Intermediate Piece	ASTM A48 Class 30B – Gray cast iron		
164B	Shaft Coupling	AISI Type 410 Stainless Steel		
262	Split Ring at Coupling	AISI Type 410 Stainless Steel		
172	Motor Support	Carbon Steel (S)		
471	Sole Plate	Carbon Steel (S)		

Notes:

- Materials in columns CF and KN denote upgrading from standard column B (carbon steel bars and forgings); intermediate and upper shafts require journal sleeve.
- When shafting is all stainless, the upper and intermediate shaft will be AISI Type 410 Stainless Steel (B) and the shaft coupling will be AISI Type 410 Stainless Steel (B, F, S).
- Where low sulphur is a requirement, AISI Type 316 will be used in place of ASTM A582 Type 416.

Discharge Head

The discharge head of the APKD is fabricated using full penetration butt welds on the pressure retaining parts in full compliance with ASME Section VIII. A single-piece forged discharge elbow is standard on the 29APKD and larger APKD sizes. On every discharge head, the fundamental frequency and reed critical frequency have been analyzed to ensure compatibility between the discharge head and motor.

Suction Can

The suction can is sized to minimize can velocities. The low velocity at the suction nozzle eliminates flashing at the can entrance. On long settings, a self-seating snubber pins the pumping element to eliminate column movement and prevent excessive wear on bearings during low flow operations or system upset.

First-Stage Design – APKD

The first stage of an APKD is a dual-volute, double-suction design to reduce the NPSH required by the first-stage impeller. This design incorporates a shorter more rigid pump without compromising hydraulic performance. To further improve NPSH characteristics, the diameter of the shaft at the first stage was reduced. Low peripheral speeds on the impeller inlet ensure stable operation over the entire operating range and maintain suction specific speed (N_{ss}) within Hydraulic Institute standards. A line from the discharge of the first-stage bowl to the suction head bearing eliminates cavitation damage in the bearing.

Intermediate-Stage Casings

The intermediate-stage casings are one-piece castings. Specifications required visual inspections for surface defects and caliper checks to ensure uniform wall thickness. More than 1/8 inch corrosion allowance is provided in the pressure walls. The cast iron casings were tested in excess of 2,000 psig. Heavy bolting (16 alloy steel studs in high-pressure applications) and gasket or O-ring joints prevent premature failure of the flange surface due to *wire drawing*. All stage bolting is mechanically locked to prevent loosening. Casing material can be upgraded to stainless steel.

Rabbet Fits

Precision, machined, rabbet fits are used throughout the casing and discharge column to ensure correct alignment of the rotor assembly.

Impellers

The impellers are a closed shroud design with the first stage being double suction. Impellers are positively locked to the shaft with lock collars, which are set-screwed to the back of the impeller hub. Full keyways uniformly transmit the torque. The standard impeller material is cast iron. The first-stage impeller is subject to cavitation damage and is commonly upgraded to stainless steel because of its resistance to cavitation and its ability to be weld repaired.

Shafting

A conservative shaft stress of less than 7500 psi is used to prevent *shaft whip*, common in vertical turbine pumps. The shaft is machined, ground, and polished to ensure trueness. Run-out is checked to tight tolerances and recorded after final machining. Ultrasonic testing is performed if requested.

Shaft Sleeves

The shaft sleeves are replaceable, hardened, stainless steel, and are keyed to the pump shaft. They can be located under carbon bearings to provide a compatible bearing surface. Not all APKD pumps were designed with shaft sleeves.

Shaft Couplings

Line shaft couplings are a solid-sleeve type. Torque is transmitted through full keyways and thrust transmitted through split rings; no threaded parts are used to transmit thrust or torque.

Radial Bearings

The radial bearings are nickel-filled carbon with bronze available as an alternate. Tight tolerances held on diameters ensure stable rotor operation.

Axial Thrust

The axial thrust is handled in upper motor bearings. It is not necessary to balance the pump stages with back rings and balance holes unless the down thrust capacity of the standard motor Kingsbury thrust bearing (KTB) is exceeded. Back rings tend to *over-balance,* creating rotor instabilities and leading to sudden up thrust on startup.

Stuffing Box

The stuffing box is a multi-element type. The stuffing box includes both a bearing (a bearing might not be present in all designs) and a separate, hardened stainless steel throttle bushing to bleed pressure back to the suction can. The shaft is completely protected through the stuffing box by a keyed stainless steel sleeve. The sleeve is serrated under the throttle bushing to minimize possible contact area and improve pressure breakdown. See Figure 2-34 for a typical stuffing box configuration. Typical stuffing box injection water specifications are:

- Packing: 10–15 psig over suction pressure
- Mechanical seal: (1/7 x design pressure) psig + 10–15 psig over suction pressure
- Injection flow: 1–2 gpm

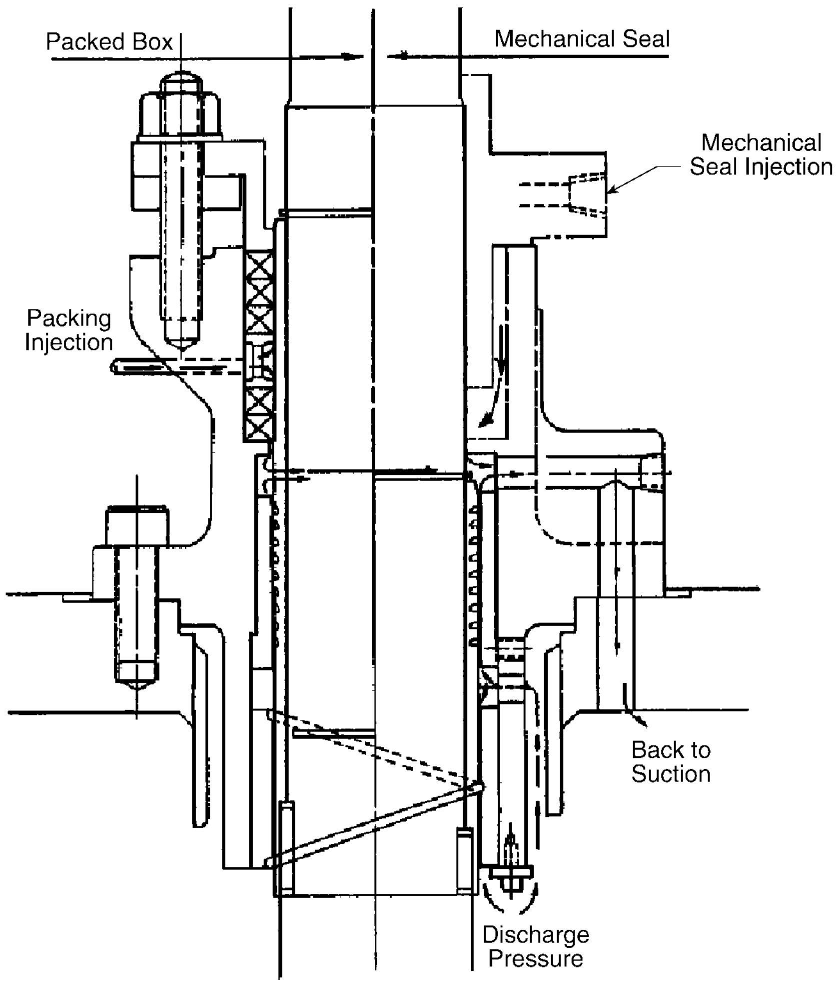


Figure 2-34
APKD Stuffing Box Arrangement

Motor Drive Coupling

Components of the motor drive coupling are forged for maximum strength and dynamic balance. The concentricity of bores and parallelism of machine faces are checked prior to final assembly to ensure proper alignment of the pump and motor shaft.

Pump Support

The pump is secured by four bolts, one in each corner of the mounting plate, which is an integral part of the shell. A foundation ring (curb ring) is available as required.

2.4.2.1 APKD Heater Drain Pumps

The APKD vertical can heater drain pumps are similar to the APKD condensate pumps with the following design additions to make them specifically suited for extended heater drain service.

Construction

For temperatures above 300°F, the multi-element stuffing box is water cooled with injection connections. All vertical can pumps on heater drain service use a self-seating snubber, located at the bottom of the shell, to eliminate column movement during low flow or system upsets; this ensures extended bearing life. See Table 2-3 for typical materials used in this application.

Table 2-3
APKD Materials for Heater Drain Application

Part No.	Part Name	Material pH Range 6.0 to 14.0
1	Casing	CA-15
3	Impeller	CA-15
6	Casing Ring	AISI Type 410 Condition K
8	Shaft Sleeve	AISI Type 410 Condition K
10A	Shaft – Pumping Element	AISI Type 410 SS (B)
10B	Intermediate (when used)	AISI Type 410 SS (B)
14	Seal Cage	AISI Type 410 SS (B) or CA-15
16	Gland	ASTM A216, Grade WCB, CS
64	Packing	J. Crane 1625 GF
88	Stuffing Box Bushing	AISI Type 410 Condition H
126	Shaft Nut at First-Stage Impeller	AISI Type 410 Condition H
127	Shaft Sleeve Nut	AISI Type 410 Condition H
135	Journal Sleeve	AISI Type 410 Condition K
138	Bearings	Nickel Filled Carbon

Table 2-3 (cont.)
APKD Materials for Heater Drain Application

Part No.	Part Name	Material pH Range 6.0 to 14.0
164A	Shaft Coupling	AISI Type 410
262	Gaskets	PE800-Asbestos
264	Stuffing Box Extension	CA-15
312	Impeller Locking Collar	AISI Type 410
359	Shell	ASTM A36 or ASTM A283, Grade C Carbon Steel
360	Suction Head	CA-15
423	Outer Column (when used)	ASTM A36 or ASTM A283, Grade C Carbon Steel
Bolting and Miscellaneous Parts		
	External Pressure Bolting	ASTMA193, Grade B7, Alloy Steel (B,N)
	Nuts	ASTM, Grade 2H, Carbon Steel Nuts (N)
	Internal Pressure Bolting	ASTM A193 Gr. B6X
	Nuts	AISI Type 416 SS
	Set Screws	Stainless Steel
	Keys	AISI Type 416 SS
	Socket Head Cap Screws	AISI Type 410 SS
	(Impeller Lock Collar)	
Special Parts for Two-Piece Pumping Element (APKD only)		
48	Intermediate Piece	CA-15
164B	Shaft Coupling (Pumping Element)	AISI Type 410
252	Split Ring at Coupling	AISI Type 410
Optional Parts Special Construction (When Used)		
172	Motor Support	Carbon Steel
471	Sole Plate	Carbon Steel

Performance

The hydraulics of the heater drain pumps are identical to that of the condensate pumps. A variety of impeller types are available to suit many system requirements and conditions. Head rises in excess of 70% are possible by properly locating the design flow. The heater drain pumps are designed to withstand the effects of occasional cavitation or flashing, but the pump cannot be guaranteed to operate in this condition without damage. The drain system should be designed to preclude flashing under all operating conditions.

2.4.3 Ingersoll-Rand Model APH/APM/APMA/APS Pump

The APH/APM/APMA pumps are vertical mixed-flow pumps specifically designed for extended operation on condenser cooling water service and flood control, or where large capacities at relatively low heads are required and installation requirements are best suited to a vertical wet-pit pump.

APS pumps are axial flow pump designs. The APS pumps are used as large-capacity low-head circulating pumps, or for dilution services. The heads on these pumps are generally limited to less than 20 foot total developed head.

Construction

These pumps are vertically mounted wet-pit type pumps. They are designed to meet a customer’s specified material and construction requirements. Non-pullout and pullout pumping elements are available to fit a customer’s installation requirements. Open shaft or enclosed shaft with inner-column construction is available. See Table 2-4 for typical materials of construction.

Table 2-4
Materials for Vertical Wet-Pit Pumps APH/APM/APMA/APS

Part No.	Part Name	Material Column				
		B (1)	CB (2)	CB	CB	CB
			IRON/SS	NIRESIST/SS	SS	AL BRZ
1	**Casing**	ASTM A48, Class 30B, Gray Iron	AISI Type 316L SS	AISI Type 316L SS	AISI Type 316L SS	ASTM 8 169 GR 613
3	**Impeller**	Alloy 922, Leaded Tin Bronze	ASTM A351, Grade CF-3M, SS			Copper Alloy
8	**Shaft Sleeve**	AISI Type 410, SS	AISI Type 316 SS			
10	**Shaft – Pump Element**	Carbon Steel	AISI Type 316, SS (B)	AISI Type 316, SS (B)	AISI Type 316, SS (B)	AISI Type 316, SS (B)
10A	**Shaft – Column and Upper**	Carbon Steel	AISI Type 316, SS (B)	AISI Type 316, SS (B)	AISI Type 316, SS (B)	AISI Type 316, SS (B)
15	**Suction Bell**	ASTM A48, Class 30B, Gray Iron	ASTM A48, Class 30B, Gray Iron	ASTM A436, Type 1, Gray Iron (A)	AISI Type 316L SS	ASTM 8 169 GR 613
16	**Gland**	Copper Alloy 932 (A)			A351 Grade CF-3M, SS	Copper Alloy 932 (A)
89	**Shroud**	407/241	AISI Type 316L SS	AISI Type 316L SS	AISI Type 316L SS	ASTM 8 169 GR 613
110	**Bearing Support**	ASTM A48, Class 30B, Gray Iron	-	-	-	-
135	**Journal Sleeve**	AISI Type 410, SS	AISI Type 316 SS			
138	**Bearings**	BRONZE-BACKED CUTLASS RUBBER				
164	**Shaft Coupling**	AISI Type 410, SS	AISI Type 316 SS			
172	**Motor Support**	CARBON STEEL				
176	**Pump Support**	AISI 1025	AISI 1025	AISI 1025	AISI 1025	AISI 1025
253	**Increaser**	Carbon Steel	ASTM A48, Class 30B, Gray Iron	ASTM A436, Type 1, Gray Iron (A)	AISI Type 316L SS	ASTM 8 169 GR 613
264	**Stuffing Box Extension**	ASTM A48, Class 30B, Gray Iron	ASTM A48, Class 30B, Gray Iron	ASTM A48, Class 30B, Gray Iron	ASTM A351, Grade CF-3M, SS	Copper Alloy
361	**Discharge Head**	Carbon Steel	ASTM A48, Class 30B, Gray Iron	ASTM A436, Type 1, Gray Iron (A)	AISI Type 316L SS	ASTM 8 169 GR 613
410	**Inner Column Guide**	Carbon Steel	AISI Type 316L SS	AISI Type 316L SS	AISI Type 316L SS	ASTM 8 169 GR 613
421	**Discharge Head Liner**	Carbon Steel	ASTM A48, Class 30B, Gray Iron	ASTM A436, Type 1, Gray Iron (A)	AISI Type 316L SS	ASTM B 169 GR 613
423	**Outer Column**	Carbon Steel	ASTM A48, Class 30B, Gray Iron	ASTM A436, Type 1, Gray Iron (A)	AISI Type 316L SS	ASTM 8 169 GR 613
424	**Inner Column (3)**	Carbon Steel	AISI Type 316L SS	AISI Type 316L SS	AISI Type 316L SS	ASTM 8 169 GR 613
471	**Sole Plate**	AISI 1025	AISI 1025	AISI 1025	AISI 1026	AISI 1025
	Bolts	ASTM A307 Grade B, CS	AISI Type 316 SS	AISI Type 316 SS	AISI Type 316 SS	AISI Type 316 SS
	Keys	SAE 1020 Carbon Steel	AISI Type 316 SS	AISI Type 316 SS	AISI Type 316 SS	AISI Type 316 SS
	Baffle Plate	Carbon Steel	AISI Type 316 SS	AISI Type 316 SS	AISI Type 316 SS	ASTM 8 169 GR 613

(1) Material column B is for fresh water service where pH levels and chloride content are normal.
(2) Material columns CB are for brackish water and saltwater service.
(3) An inner column is used when shaft span requires a column bearing.

Note:
Large vertical condenser circulating water pumps, being of high specific speed design, are susceptible to cavitation damage unless proper selection of materials is made. To minimize cavitation damage, materials are changed to lessen the effects of fluid separation on impeller blades. Column B pumps that are Class III or in cooling tower (corrosive) service must have stainless steel impellers.

Casing

The casings are cast or fabricated to meet customer material or construction specifications. When cast, it is a one-piece precision casting with heavy walls and vanes. Foundry procedures that involve hot shake out and stress relieving prevent subsurface cavities and eliminate shrinkage cracks.

Impellers

The impellers are an open design and are cast or fabricated to meet customer material or construction specifications. The open impeller and shroud offer improved efficiency and easily renewable clearances. Cast impellers are from a special low-yield one-piece casting formed in precision molds to ensure subsurface quality and proper vane location, shape, and finish.

Each impeller is checked to confirm required tolerances, and vane profile is checked with templates. Dynamic balancing of the finish-machined impeller ensures low vibration levels.

Impeller surfaces are 100% visually inspected for surface defects and repaired by qualified welders. The impellers are locked in position by split rings, which are bolted to the impeller hub on the inlet side.

Impeller Shroud

The impeller shroud is a one-piece fabrication, bolted to the casing end bell on a non-pull-out design, and bolted to the inner casing on the pullout design. On a pullout design, the shroud is removed with the pumping element. Special consideration is given to fabrication procedures that remove locked-in stresses and provide the necessary finish to permit uniform running clearances with the impeller for extended high-efficiency performance.

Suction Bell

The suction bell is a heavy wall one-piece casting or fabrication with optimum shape and finish for efficient velocity increase and vortex suppression. The suction bell has straightening vanes to minimize flow disturbances at the impeller eye and to ensure high design point efficiency.

Discharge Head

The discharge head can be cast or fabricated to meet customer material or construction specifications. On a pullout pump, the discharge head involves two pieces. The discharge elbow and the discharge head liner, integral with the pullout cover, efficiently change the direction of the flow from the column to the discharge nozzle. On a non-pull-out pump, the discharge head is a one-piece casting or fabrication, and turning vanes or a long radius elbow is employed to accomplish the change of flow direction. On all models, the discharge head is conservatively designed to accept the continuous reaction force resulting from an unrestrained expansion joint.

Outer Column

The outer column is fabricated or cast to meet customer material or construction specifications. The outer column flanges feature precision rabbet fits to ensure proper alignment of each pump section. Where necessary, support feet for blocking are provided at proper locations to simplify assembly and disassembly.

Inner Column

The inner column is fabricated and flanged with rabbet fits to ensure proper alignment. The inner column is internally pressurized to prevent loss of lubrication during operation.

Shafting

The shaft is precision machined for trueness and balance, which will minimize shaft vibration and maximize bearing life. Shaft sections are held together with keyed sleeve couplings that employ a split lock collar. Torque is transmitted by the keys and axial thrust is transmitted by the lock collar. To facilitate assembly/disassembly, no threaded parts are used in critical areas.

Bearings

The bearings are cutless rubber with metal backing and will be lubricated prior to and during operation. Two bearings are installed in the casing, one adjacent to the impeller and one at the top of the casing to carry the radial impeller loading. Column bearings and spiders are provided as necessary to maintain a stiff rotor construction. A bearing is located adjacent to the stuffing box extension to support the shaft in the discharge head.

Stuffing Box

The stuffing box is a one-piece casting and is packed due to the service and pressures. Prior to startup, and during operation, injection is supplied to the stuffing box and bearings.

Motor Coupling

The motor coupling is a rigid adjustable coupling, which provides the means for maintaining the design clearance between the impeller and the shroud, thus ensuring maximum efficiency. Because the APS pump shroud is straight, there is no need for an adjustable coupling and the coupling is a rigid non-adjustable type. Pump support is accomplished by a heavy-duty steel mounting plate to be bolted to the customer's foundation. A sole plate (curb ring) is available as required.

APM/APMA/APS Performance Curves

Most of the APM/APMA/APS performance curves have a discontinuity or hook. This discontinuity is not normally detrimental to startup, but prolonged operation in the discontinuity region of the pump curve should be avoided.

Key Technical Point

Most APM/APMA/APS performance curves have a discontinuity or hook. Typically minimum operating flow is 85% or greater than the pump best efficiency point (BEP) flow.

APH/APM/APMA Impellers

The APH/APM/APMA impellers are mixed-flow types, and the hydraulic performance of each pattern is varied by cutting its vane length or diameter as shown in Figure 2-35. Full vane length will provide the maximum capacity and head.

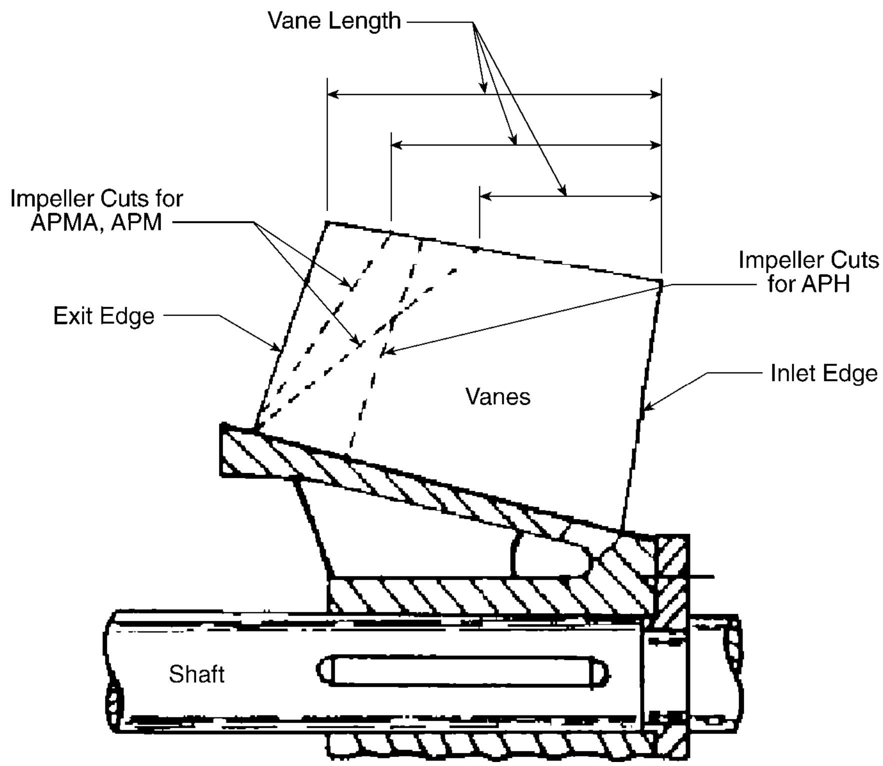


Figure 2-35
APH/APM/APMA Impellers

APS Impellers

The APS impeller is an axial flow type. Hydraulic performance is varied by changing the number of impeller vanes and the impeller vane angle. For axial flow pumps, impeller vane length and diameter are not cut (see Figure 2-36).

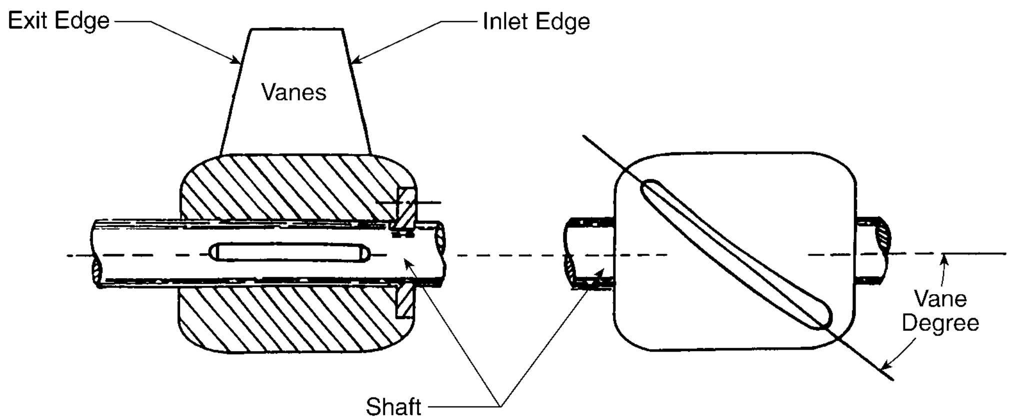


Figure 2-36
APS Impeller

2.5 Screen Wash Pump Upgrade for Grass Fouling

Thick grass can bridge suction bell openings (between the bottom bearing ribs) and eventually starve the pump for water. Pumps that have previously been fitted with the conventional grass cutting impellers on the first stage (sharpened inlets) help, but this upgrade can be inadequate to address the frequent “bridging” issue on the bell.

Pumps can be modified with an external grass cutter that fits the contour of the suction bell and addresses loss of product bearing lube due to extreme grass fouling. A three-vane cutter was contoured for grass cutting without vane pass vibration or fouling potential.

This upgrade consists of a redesigned suction bell that would allow an external cutter to be installed upstream of the first-stage impeller. The cutter has three vanes that will cut the grass against the suction bell and prevent the suction of the pump from being clogged with grass. This has been installed on all screen wash pumps at Salem Nuclear Plant.

3
VERTICAL PUMP VIBRATION MONITORING, DIAGNOSTICS, AND TROUBLESHOOTING

Vertical pumps offer unique vibration monitoring, diagnostics, and prediction challenges. The following information provides guidance and process information for pump vibration diagnostics, which is required for the installation, maintenance, and problem solving situations often encountered.

This framework includes some recommended processes to pursue field problems typically encountered and is a combination of technical material and definitions, flow charts for common vibration situations, and tables and charts that provide key parameters and characteristics needed for the resolution of vibration-related issues.

3.1 Technical Definitions and Discussion

3.1.1 General Concepts

Key Human Performance Point

Vibration testing is the science of using appropriate testing techniques, depending upon the results or outcome of previous data or information. This creates difficulty in establishing a standard test plan for a specific problem. The use of various techniques and technologies allows the evaluator to quickly focus on the key elements of a problem and bypass other non-vital information.

Vibration testing is the art and science of measuring and quantifying a symptom of a problem with instrumentation that provides very detailed and specific information. This testing and data collection typically includes explanation and details that are obtained through conversations with maintenance, engineering, operations, and vendor staffs, but not specifically observed from vibration data.

Several methodologies are adopted to perform vibration testing on pumps and associated components. Some of the more typical methodologies include:

- Trending of vibration amplitudes over time at distinct frequencies to monitor changes in equipment condition
- Evaluating the frequency and phase relationship of high-amplitude output to provide insight regarding the cause of distress
- Measuring the vibration amplitudes throughout the speed range

High pump vibration is typically a result of one or more of the following problems or conditions:

- Misalignment
- Unbalance
- Structural resonance
- Excitation of rotor critical speed
- Inlet flow disturbances
- Increased impeller-to-shroud axial clearance (increased recirculation)
- Impeller inlet separation/cavitation
- Shock loading of exiting impeller flow onto stationary diffuser/volute vanes
- Increased pump radial bearing clearance
- Bearing looseness
- High axial thrust
- Worn/damaged motor bearings
- Component rubbing
- Cracked or bowed shaft
- Broken or eroded components (for example, missing impeller vane)
- Loose foundation or column joints
- Foreign material lodged in pump impeller
- Flow cavitation
- Air intrusion or ingestion
- Improper flow requirements

Key Technical Point

The most common vibration problems associated with vertical pumps are unbalance and misalignment.

These problems can result in reduced pump performance and/or catastrophic failure. Early detection is important to facilitate maintenance planning and to avoid lost revenues due to plant load reductions. In extreme cases, these problems can result in forced outages. In some instances, early detection and analysis can lead to remedying the cause of vibration without dismantling the unit.

Early detection can best be achieved through the use of a vibration monitoring system and/or frequent measurements and review by experienced staff.

3.1.2 Likely Failure Modes

Vibration problems in pumps can be related to either rotating or stationary components. The two most common rotor-related problems are misalignment and unbalance.

Misalignment can manifest itself within the motor, between the pump and motor, or among the pump column sections. Misalignment can be parallel, angular, or a combination of the two. There are several variables that affect alignment during normal operation, including thermal growth during startup and fluid dynamic forces.

Unbalance can appear as overall system unbalance or be limited to pump and/or motor components. Even if the pump is "perfectly" balanced prior to shipment, the rotor balance can change during operation.

Other rotor-related problems that affect equipment vibration amplitudes and reliable pump operation include:

- Axial position change (excessive thrust)
- Rubbing between rotating and stationary parts
- Operation near or at rotor critical speed
- Shaft cracks and breaks
- Shaft bending or bowing
- Loose components
- Instabilities due to pump output changes

Table 3-1 provides some common pump problems with respect to frequency content and/or inspection forensics.

Table 3-1
Common Pump Problems

	Wear Pattern				
Condition	**Localized Stationary**	**Localized Rotating**	**Circumferential Stationary**	**Circumferential Rotating**	**Vibration Frequency**
Static bow		X	X		1X
Dynamic bow in the shaft (operating at any mode of a shaft critical speed)			X	X	1X
Loss of flush water flow or pressure	X	X			
Non-concentric stationary component	X			X	1X

Table 3-1 (cont.)
Common Pump Problems

	Wear Pattern				
Condition	**Localized Stationary**	**Localized Rotating**	**Circumferential Stationary**	**Circumferential Rotating**	**Vibration Frequency**
High vibration caused by the following:					
Pump unbalance				X	1X
Cavitation/separation					(0 to 1X)
Hydraulic instability					(0 to 1X)
Component looseness			X	X	(1/2X, 1X, 1 1/2 X, 2X)
Loose bearing					(1X, 2X, 3X)
Cracked shaft		X			(1X, 2X)

3.1.3 Basic Vibration Principles

All vibration is initially caused by a force. The result of this force is motion. Motion can be described by these three basic parameters:

- Amplitude
- Frequency
- Phase angle

Amplitude

Vibration amplitude is a measurement of vibration magnitude. The level of magnitude is an indication of the vibration severity. A normal operating machine will generally have a stable amplitude reading at an acceptably low level. Any change in amplitude signifies a change in machine condition.

The amplitude can be expressed in terms of displacement, velocity, or acceleration. When measuring displacement, the amplitude of motion can be absolute, or relative to some frame of reference.

Displacement is also a useful tool when measured relative to some frame of reference. Proximity probes located at or near bearings give the relative displacement of the shaft to the bearings. Knowing the design clearance, relative shaft displacement can become a very valuable tool for determination of machine integrity.

Velocity is the more common method of expressing vibration amplitude today. Its popularity stems from the fact that velocity is an excellent indication of vibration severity at any frequency because severity follows constant velocity lines over the useful frequency range.

Acceleration is a more functional means of expressing vibration when there is a desire to quantify the acting forces (F = ma), particularly at a higher frequency where displacements might be quite small.

Key Technical Point

Amplitude can be expressed in many forms such as displacement, velocity, and acceleration.

Amplitudes are normally compared to vibration severity (allowable vibration limits) to determine the acceptability of various levels of vibration. Table 3-2 provides an example of a company standard for vibration severity. Figures 3-1 and 3-2 provide some vibration severity criteria obtained from industry experience and from a recognized consultant.

Table 3-2
Company-Specific Acceptance Bearing Housing Standard

Acceptable Vibration Amplitude Limits		
Frequency Range	**Units**	**Discrete Spectrum Frequency**
Sub-harmonic (<1X rpm)	in./sec pk (mm/sec pk)	0.04 (1.02)
1X rpm	in./sec pk (mm/sec pk)	0.15 (3.85)
(1.1X–10X rpm)	in./sec pk (mm/sec pk)	0.10 (2.56)
Overall velocity (0–2000 Hz)	in./sec pk (mm/sec pk)	0.20 (5.13)

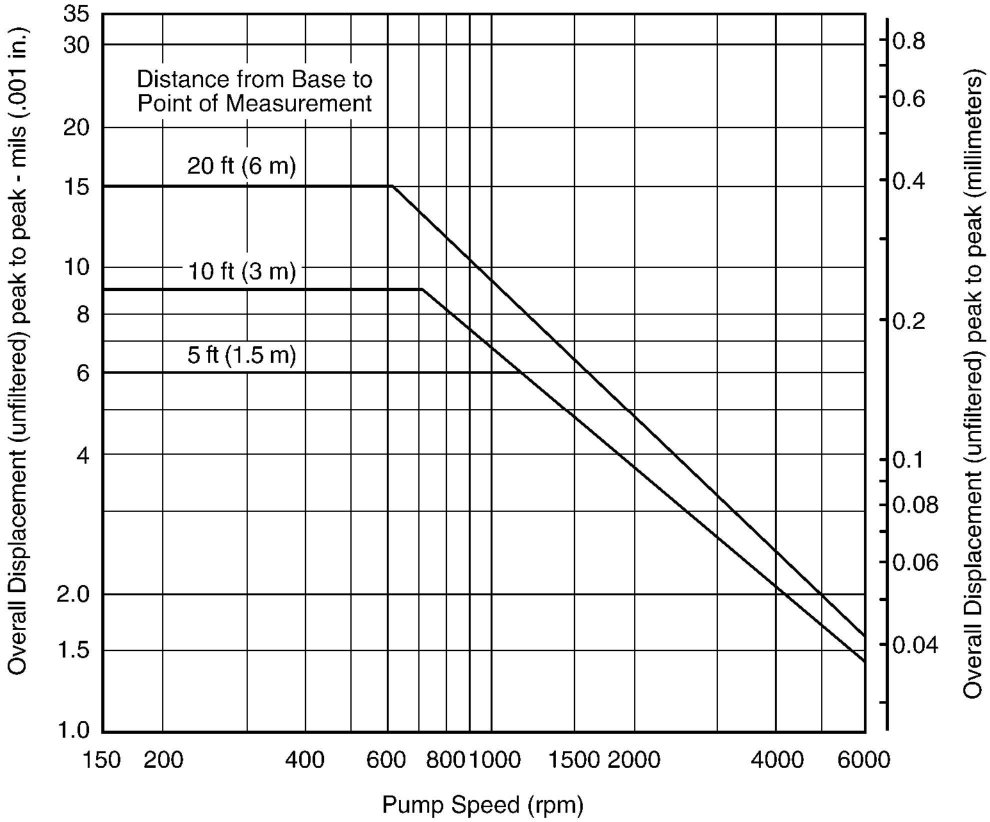


Figure 3-1
Hydraulic Institute Vibration Severity Based on Height, Speed, and Overall Amplitudes

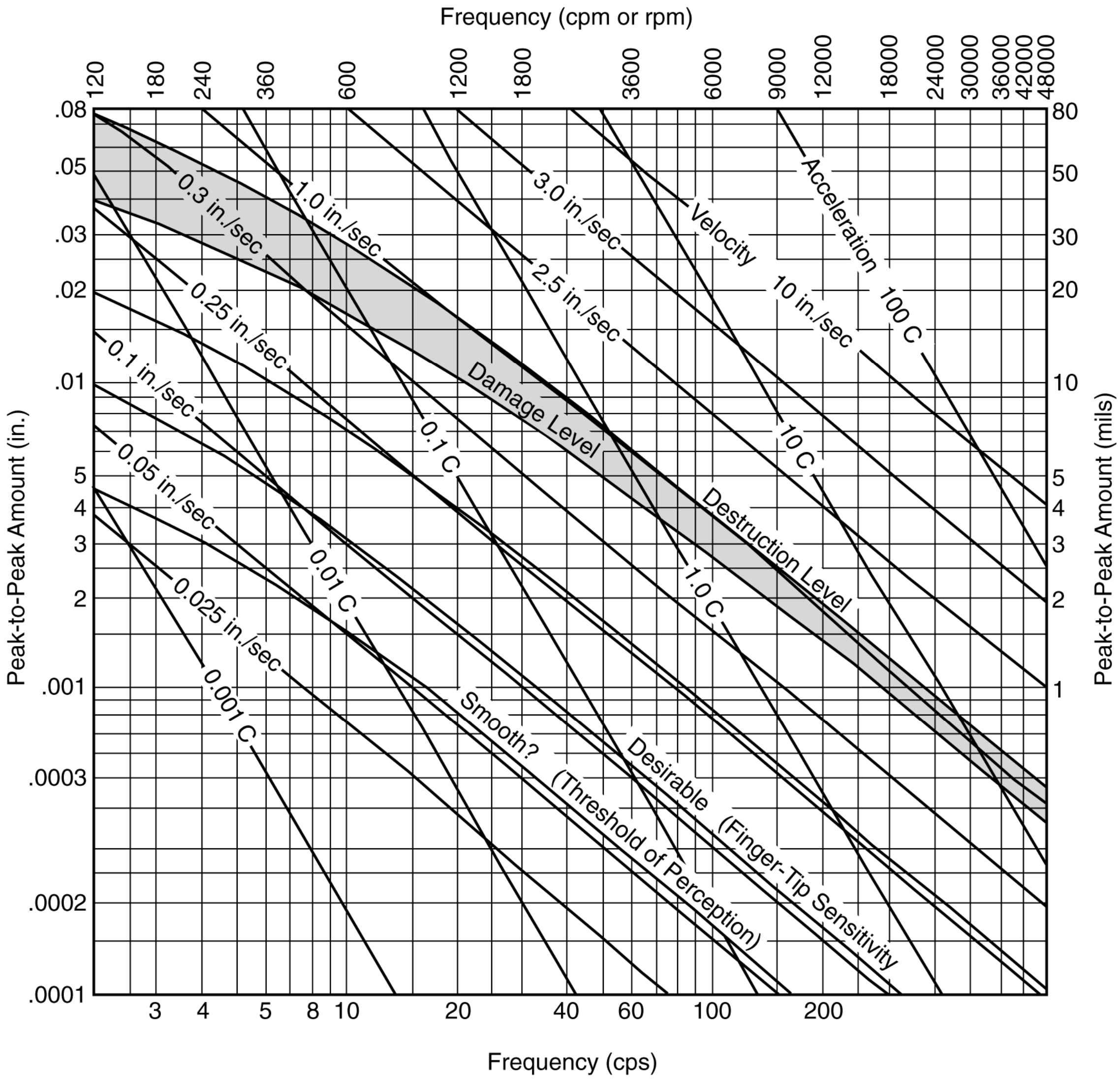


Figure 3-2
McKay Severity Chart (Shaft Relative Vibration)

Frequency

Frequency quantifies the timing relationship of a vibration signal. The period of motion is the number of seconds it takes to complete one cycle. The frequency of motion is the inverse of the period (cycles/sec). Frequency is not directly related to vibration severity; however, frequency is very useful in determining the cause of vibration. The frequency of vibration often identifies the mechanism that is creating the abnormal forces on a machine.

In rotating machines, frequency is most commonly expressed in multiples of the equipment rotating speed because machine vibrations usually occur at direct multiples or submultiples of this speed. Examples are:

1 x: Frequency of vibration is the same as the equipment rotating speed.

2 x: Frequency of vibration is twice the equipment rotating speed.

1/2 x: Frequency of vibration is one-half the equipment rotating speed.

Z x X: Frequency of vibration is the equipment vane-pass frequency where Z is the number of impeller vanes and X is a term to express the equipment rotating speed.

Key Technical Point

Frequency is the number of occurrences with respect to time. The frequency provides insight into the source of the vibration.

Phase Angle

The phase angle is actually a relative time measurement that is converted to an angle of rotation based on some reference point and frequency. The reference point can be an event, such as a reference on the shaft surface, or another vibration signal source.

For rotating shafts, the phase normally corresponds to the high spot on the rotor as it passes the transducer, or the high point of the sine wave for any specific frequency.

Phase angle versus frequency, when plotted coincidentally with amplitude versus frequency, is a good means of determining component natural frequencies. For a “text book” natural frequency, there will be a phase angle shift of at least 90° and a corresponding amplitude increase, depending upon the amount of system damping.

Phase angle is essential in component balancing and other advanced diagnostic techniques such as modal testing, operating surveys, order track analysis, and orbit analysis.

Key Technical Point

Phase angle provides the timing relationship of the vibration amplitude of any two points.

3.2 Standard Monitoring Guidelines

Standard monitoring equipment and locations are provided in Tables 3-3 and 3-4, and Figure 3-3. The standard recommendations in Table 3-3 reflect the ideal setup for detection and diagnosis of equipment problems. A program that meets the standard equipment and location requirements provides maximum insight into the equipment issues or problems so that early detection and problem resolutions can be implemented. The minimum monitoring recommendations provided in Table 3-4 allow a more limited view of the component condition and, as such, might not provide sufficient warning to prevent costly or catastrophic failures, including equipment destruction and forced plant outages.

Additionally, the approach to gathering data before and after equipment replacement and maintenance significantly affects the ability to diagnose problems and identify installation issues. The flow charts in Section 3.3 identify the recommended and minimum approaches.

The success of a vibration monitoring and diagnostic program is highly dependent on the quality of the equipment installed and the approach used to obtain diagnostic and routine data. Deviation from the recommended monitoring table and diagnostic approach could limit the effectiveness of a comprehensive program.

Table 3-3
Recommended Vibration Monitoring Equipment and Locations

Monitoring Recommendations		
Probe Type	**Location**	**Measurement**
Proximity X-Y (90°)	X-Y above coupling looking at motor shaft and X-Y below coupling looking at pump shaft	-Shaft position and motion relative to bearing clearance -Shaft relative radial vibration -Shaft mode shapes -Pump/Motor alignment -Rotor unbalance -Shaft runout
Phase Reference Transducer	On motor shaft (above coupling)	-Shaft speed (rpm) -Synchronous reference (1/rev)
Proximity Probe, Axial (KTB Bearings)	At motor thrust collar or end of shaft	Thrust reversal with low flow operation
Accelerometer	X-Y mounted on motor housing at each motor bearing location; axial direction on motor housing (usually top of motor)	-Quantify the structural vibration and/or force transmitted to housing -Structural natural frequencies -Provides shaft overall displacement when used with shaft relative probes
Proximity X-Y (90°) (Optional for added protection and diagnostic information)	X-Y probes along the pump column looking at pump shaft. Feasible when detailed for extreme environment.	Provides additional shaft relative motion at depths below pump head. Some limitations on sensor equipment reliability under hydrostatic and elevated temperatures.

Table 3-4
Minimum Recommended Vibration Monitoring Equipment and Locations

Minimum Monitoring Recommendations		
Probe Type	**Location**	**Measurement**
Proximity X-Y (90°)	X-Y below coupling looking at pump shaft	-Shaft position and motion -Shaft radial vibration -Shaft mode shapes -Rotor unbalance -Shaft run-out
Phase Reference Transducer	On motor shaft (above coupling)	-Shaft speed (rpm) -Direction of rotation -Synchronous reference (1/rev)
Accelerometer	X-Y mounted on top of motor housing; axial direction on motor housing (usually top of motor)	-Quantify the structural vibration and/or force transmitted to housing -Structural natural frequencies -Provides shaft overall displacement when used with shaft relative probes

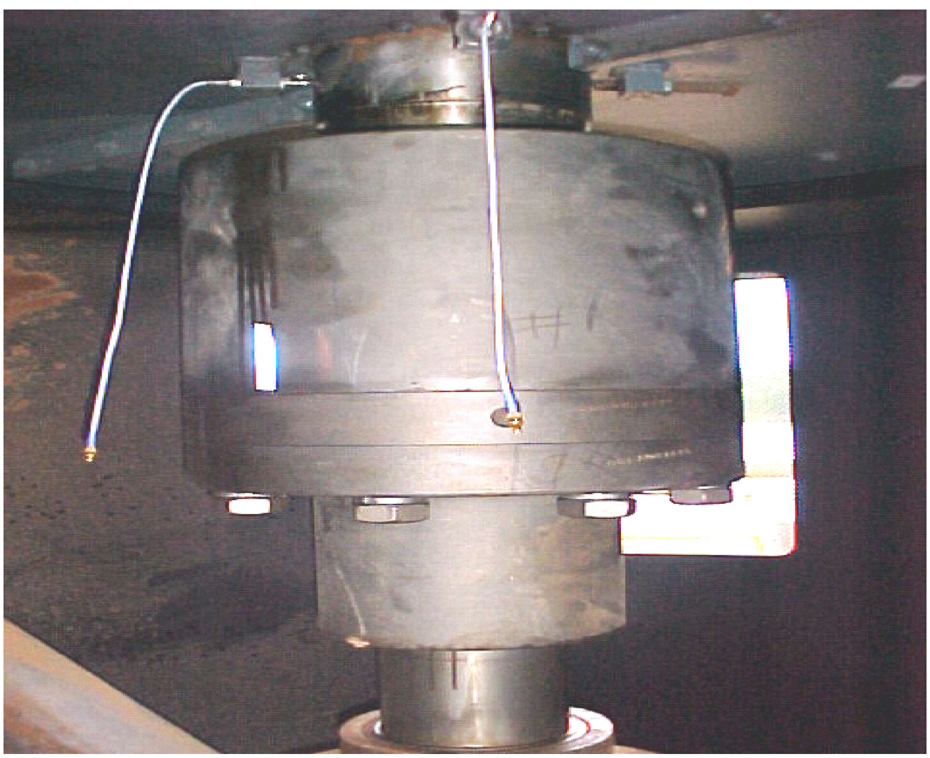


Figure 3-3
Vertical Shaft Coupling with Proximity Probes Above Coupling

Figure 3-4 provides an example of the shaft radial centerline position for various operating conditions.

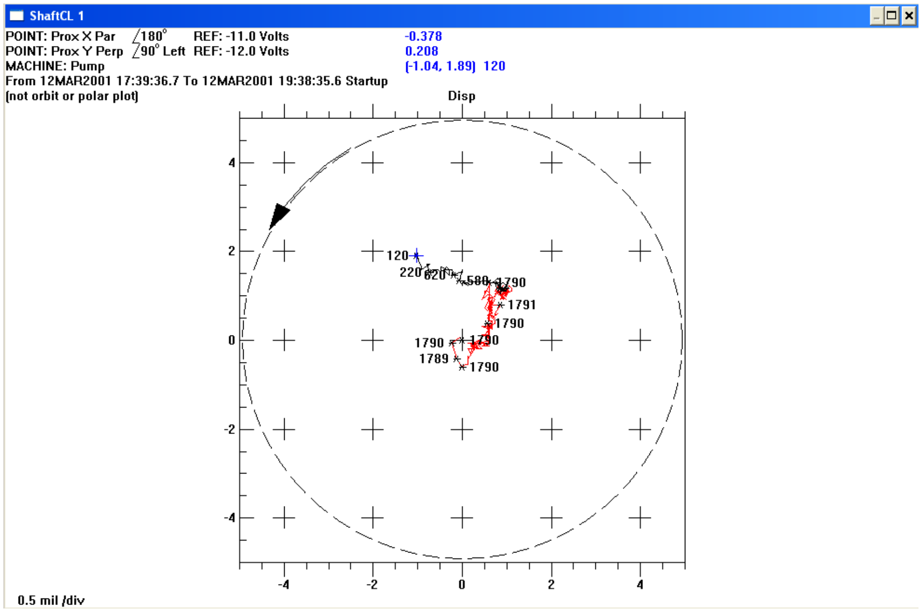


Figure 3-4
Shaft Centerline Plot of Shaft Position from Startup to Full Speed of 1800 rpm

3.3 Vibration Diagnostics and Troubleshooting

Analysis of data from routine vibration collection, lube oil samples, motor current collection, etc., have indicated a problem or issue exists in a pump or rotating piece of equipment. The next step in the process would be problem diagnosis to identify the root cause(s), implement effective solutions, and verify successful results. Figures 3-5 through 3-11 in the following sections provide guidance for troubleshooting an identified pump problem. Figures 3-12 and 3-13 also provide decision trees to help in guiding the troubleshooting process during normal maintenance activities.

Key Technical Point

In most plants, vibration test personnel collect measurements to compare to formal or informal acceptance levels. If the vibration meets the acceptance criteria, then the pump is placed in service. If the pump does not perform satisfactorily, then the process of vibration diagnostics begins.

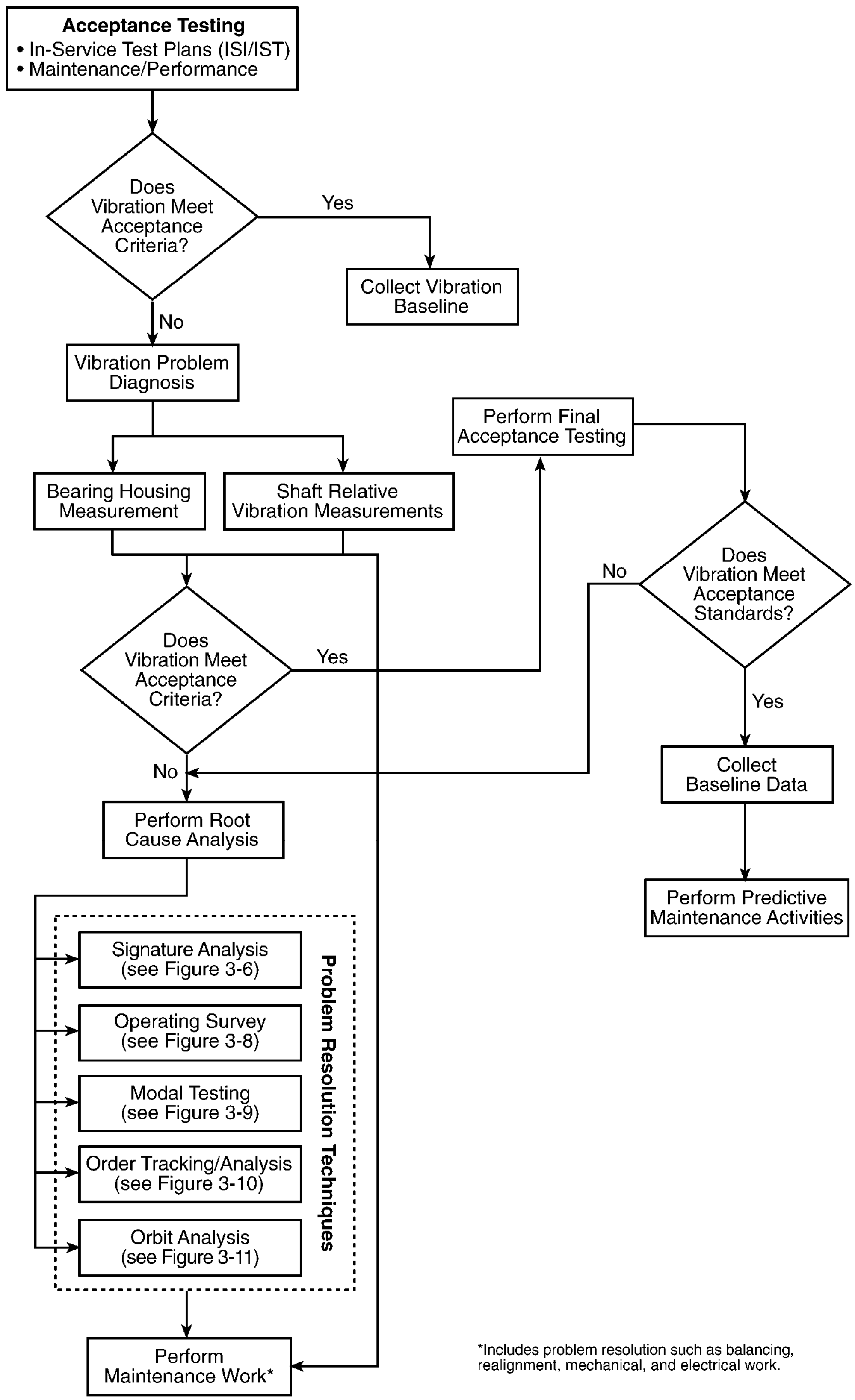


Figure 3-5
Troubleshooting Approach for Pump Problems

3.3.1 Problem Resolution Technique – Signature Analysis

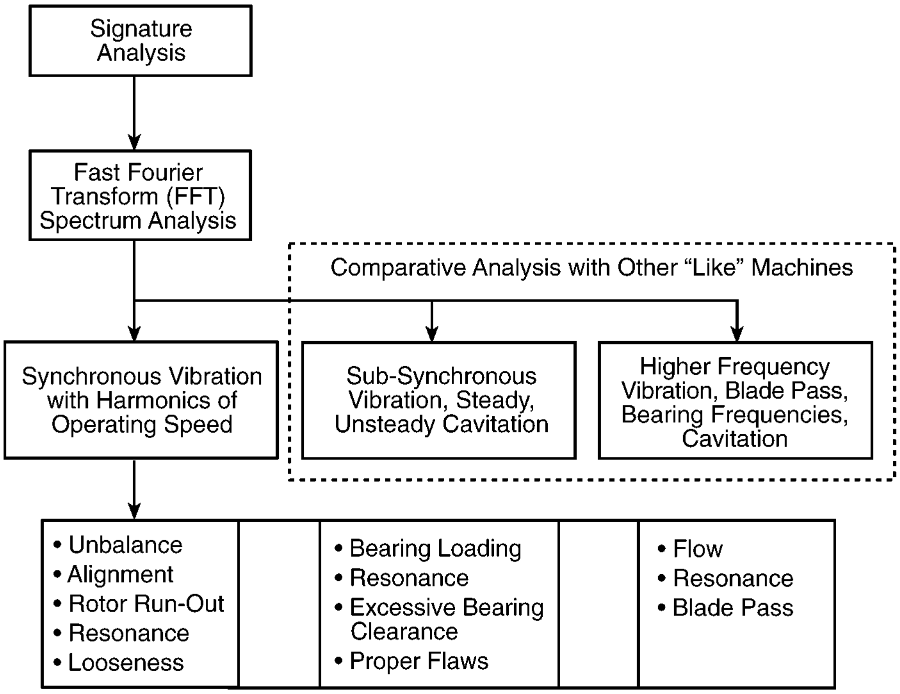


Figure 3-6
Signature Analysis Process Technique

Figure 3-7 provides a typical spectrum plot of an accelerometer attached to the top of a vertical pump.

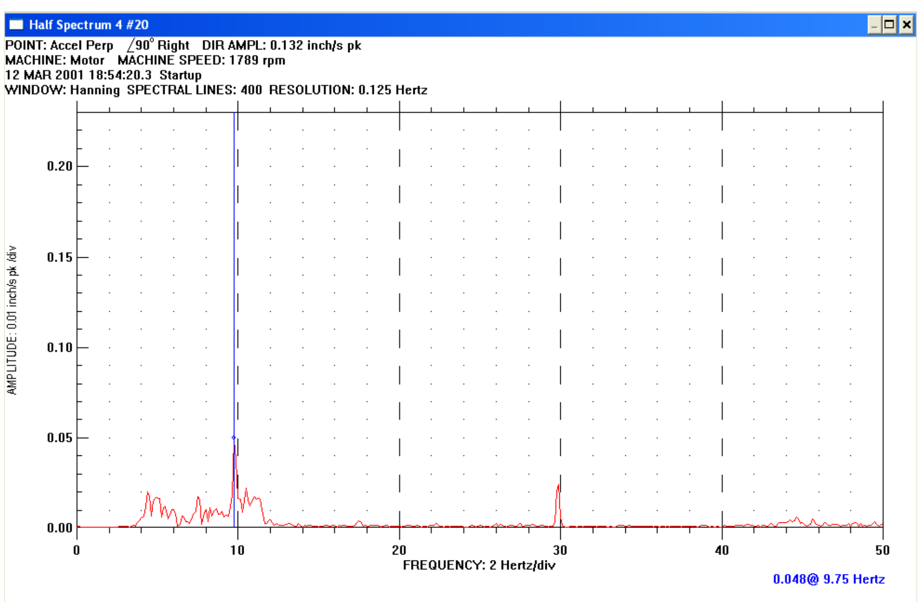


Figure 3-7
Frequency Spectrum Plot of Accelerometer Mounted on Top of Motor Parallel to Discharge Piping

3.3.2 Problem Resolution Technique – Operating Survey Analysis

This technique is similar to the modal analysis technique. This technique provides the relative motion of several data points at a particular frequency. This technique will indicate the motion of the equipment under normal and abnormal conditions; therefore, an abnormal or illogical shape will reveal the problem. This process fundamentally requires the existence of a steady amplitude and phase at the particular frequency of interest.

Figure 3-8 provides a typical layout of a vertical pump, when the operating survey technique is used. The stations provided are indicative of the elevations used to collect the vibration measurements. These measurements might typically involve four locations around the circumference of the pump column or base plates. This technique will identify flexibility or looseness at particular measurement locations and help the analyst with problem identification.

From spectrum analysis, determine the significant frequencies of concern and/or severity.

↓

At the frequency of interest, collect the vibration amplitude and phase angle at any data point on the structure in 3 dimensions.

↓

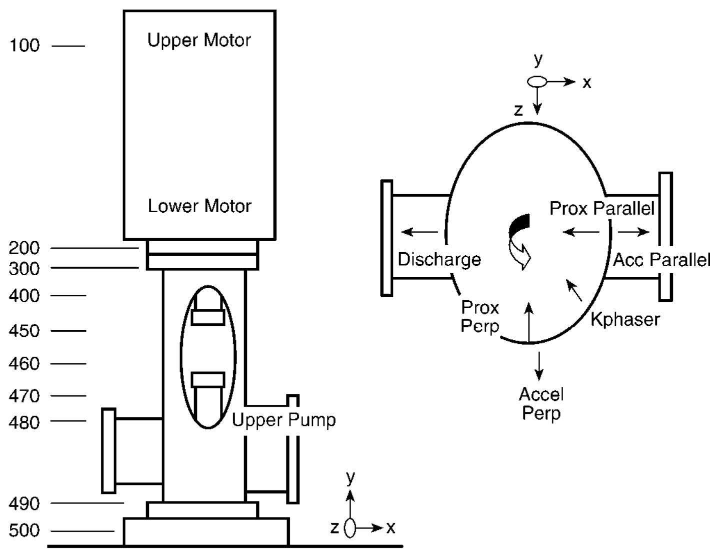
Draw and describe the structure in a simple stick figure, to represent the major points of motion on the structure.

↓

Overlay the vibration displacements at the respective data points on the structure to represent the amplitude of motion and the relative phase between the various data points.

↓

Look in particular across mechanical joints, base plates, and interfaces between pump/motor, pump/base, and so on for differential motion and/or phase angle. (This would indicate the potential source of the problem.)



Typical Operating Survey Layout

Figure 3-8
Operating Survey Analysis Technique

3.3.3 Problem Resolution Technique – Modal Analysis/Impact Testing

This technique allows the analyst to:

- Identify natural frequencies
- Quantify and describe the motion of the natural frequency
- Impact equipment/structure with a hammer

This technique provides the natural frequency of the pump structure or component (see Figure 3-9). Natural frequencies exist on all equipment and structures. The ideal condition is to not have a natural frequency close to the operating speed, harmonic, or discrete frequency of vibration.

The use of the impact test response function is to identify the natural frequencies that are harmful or potentially of concern in amplifying the vibration condition of a pump component. When performed for a medley of data points at any particular frequency, the amplitude and phase angles can also provide a modal shape of the natural frequency, similar to the operating survey.

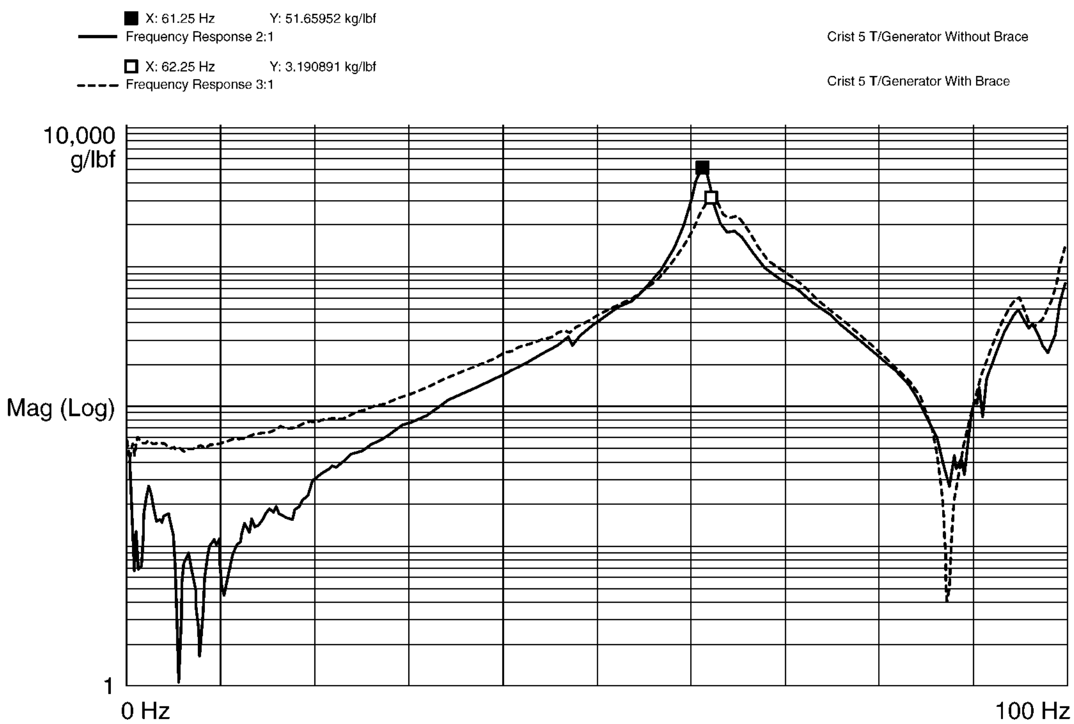


Figure 3-9
Modal Analysis/Impact Testing

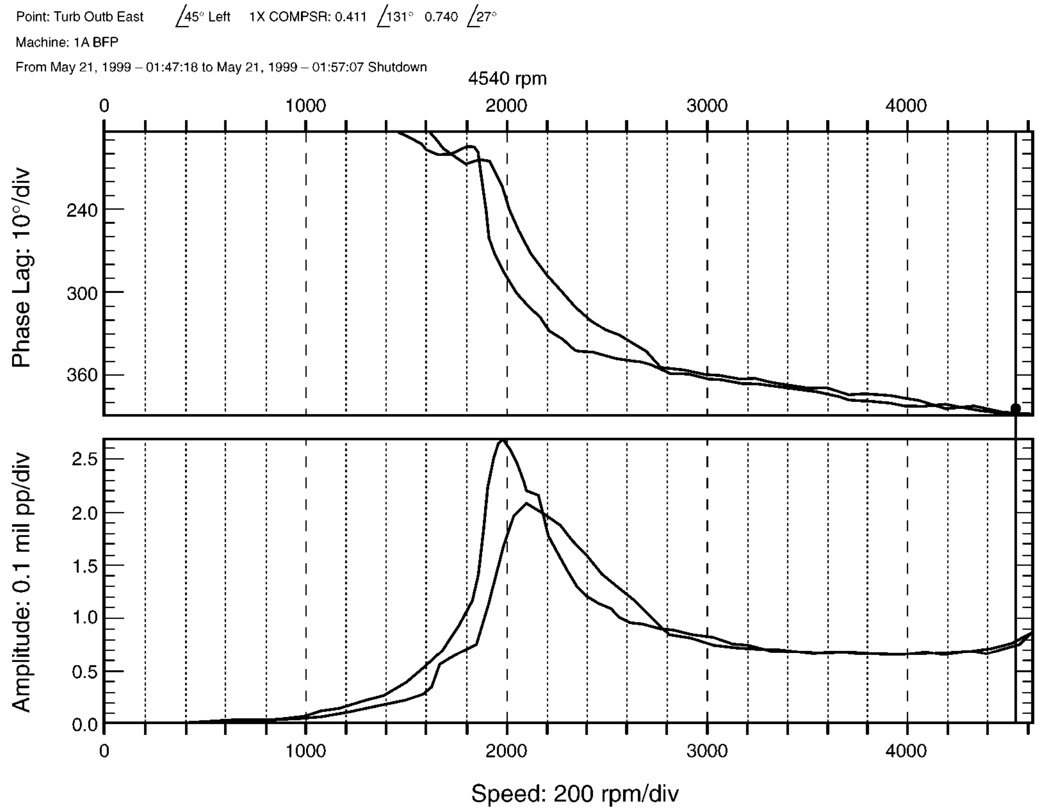
3.3.4 Problem Resolution Technique – Order Tracking and Analysis

The vibration at the operating speed is called synchronous or 1X vibration. Harmonics of the synchronous (1X) vibration are called 2X, 3X, and so on. These harmonics are also referred to as orders. Thus, the 2nd harmonic is also the 2nd order.

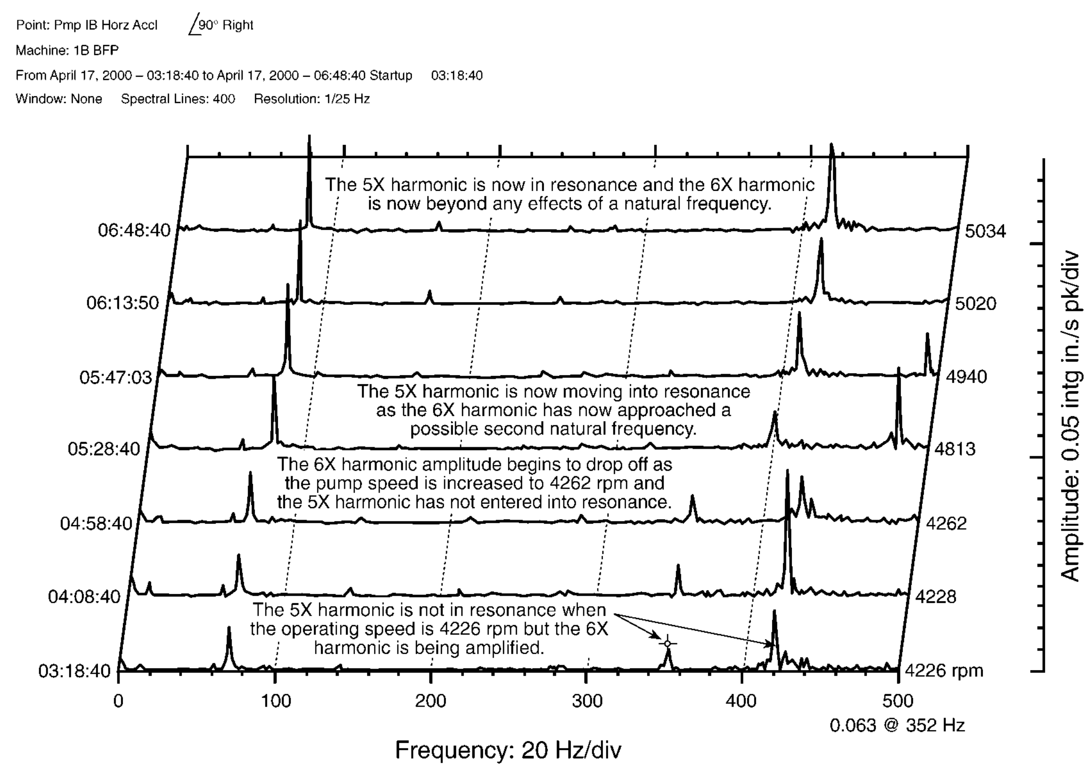
Order tracking records the amplitude of the harmonics or orders with speed change. Order track plots show how the harmonics vary with speed. These plots are particularly useful in determining the natural frequencies of the system in those speed ranges. These plots are also referred to as Bode plots.

The natural frequencies show up in a Bode plot as peaks in the amplitude plot with a corresponding phase shift. The Bode plot in Figure 3-10a shows a shaft critical at about 2000 rpm.

The waterfall plot (Figure 3-10b) shows the vane pass (5th order) vibration as it approaches a bearing housing natural frequency at 4900 rpm.



(a)



(b)

Figure 3-10
Order Tracking and Analysis

3.3.5 Problem Resolution Technique – Orbit Analysis

The orbit plot is the plotting of the vibration from two orthogonal probes that are located in the same radial plane. This plot provides the displacement path of the shaft surface as shown in Figure 3-11. Under normal circumstances, this orbit should be an oval-shaped circle. Each line from the orbit plot represents a single 360° rotation. Each orbit rotation should retrace the previous orbit, if the vibration is steady and repeatable, and no flat spots or discontinuities in the trace should be found.

The study and comparison of the orbit plot provides insight into the shaft vibration levels and the condition of the bearings, shaft surfaces, and relative motion of the shaft with respect to its bearings.

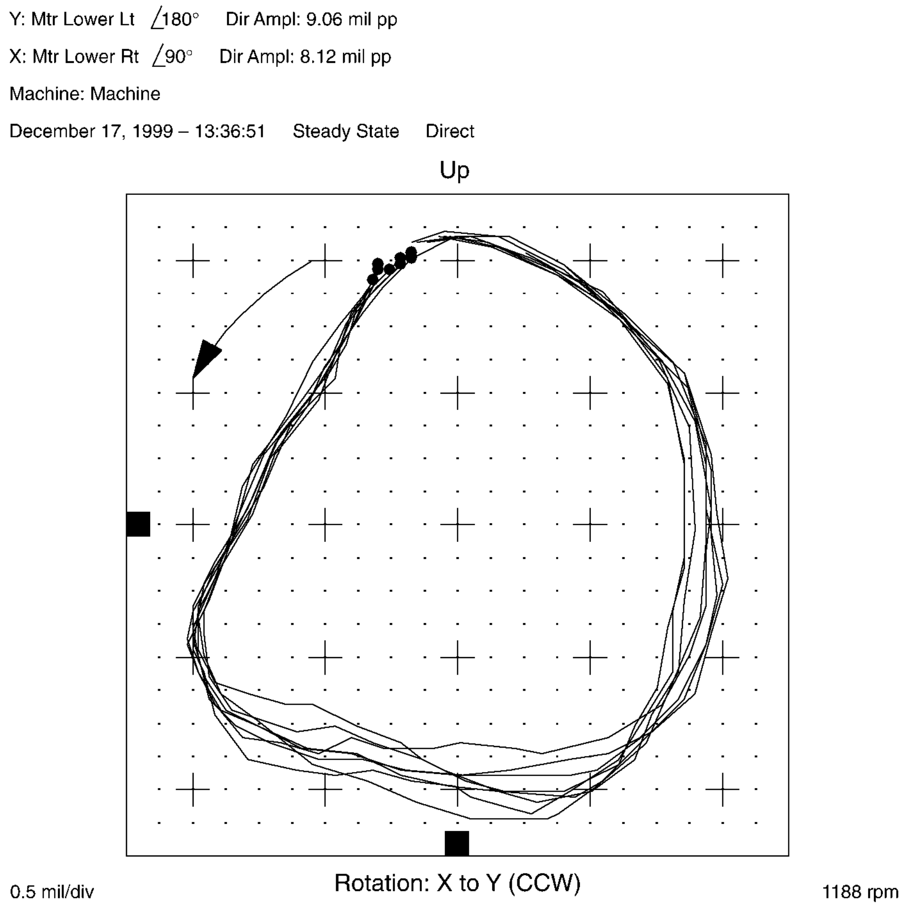
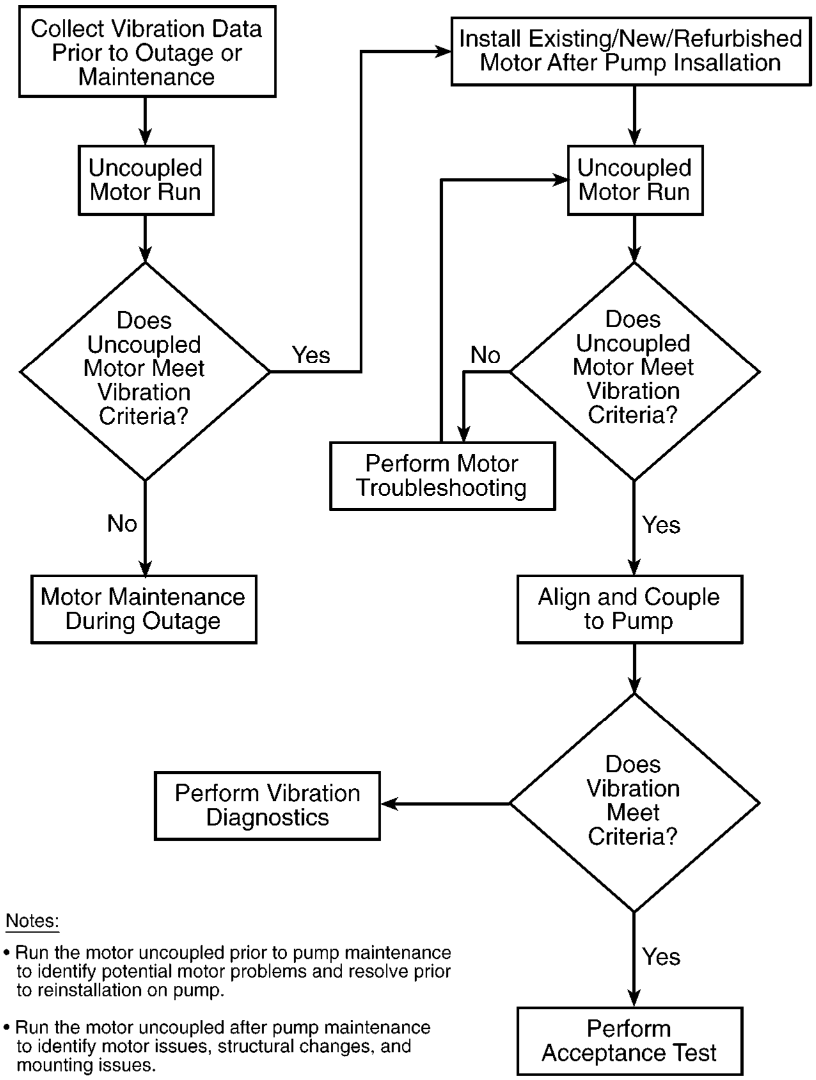


Figure 3-11
Orbit Analysis



Notes:

- Run the motor uncoupled prior to pump maintenance to identify potential motor problems and resolve prior to reinstallation on pump.
- Run the motor uncoupled after pump maintenance to identify motor issues, structural changes, and mounting issues.

Figure 3-12
Recommended Diagnostic Approach

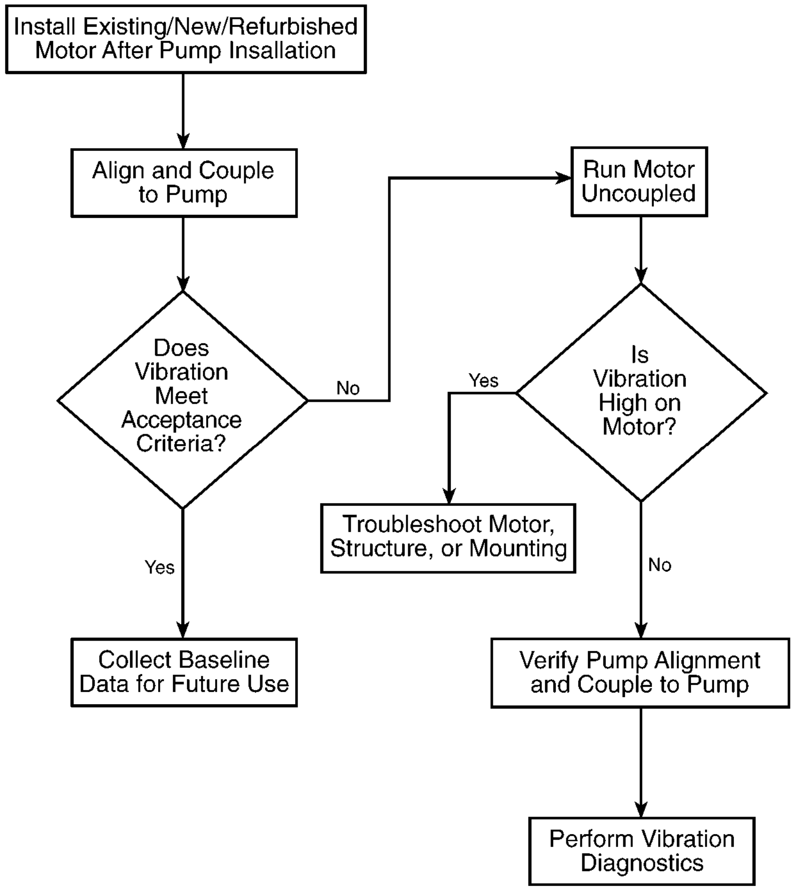


Figure 3-13
Minimum Pump/Motor Diagnostic Approach

3.4 Problem Resolution Recommendations (Field Balance Process)

During the diagnosis procedure, it might become obvious that the cause of the vibration problem is related to unbalance of the assembly or sub-assembly. The best situation by far is to correct the unbalance at the location of the true unbalance condition. However, this might not be possible, either because of a lack of balance quality during assembly of the rotating components, or because of the inability to balance the machine in the production bearings and/or structure at normal operating speeds and conditions.

Under these scenarios, field balancing procedures are provided to aid in an approach for the field correction of unbalance under various conditions. The following general process (Figure 3-14) provides insight into the decisions that are commonly required to begin this balance iteration.

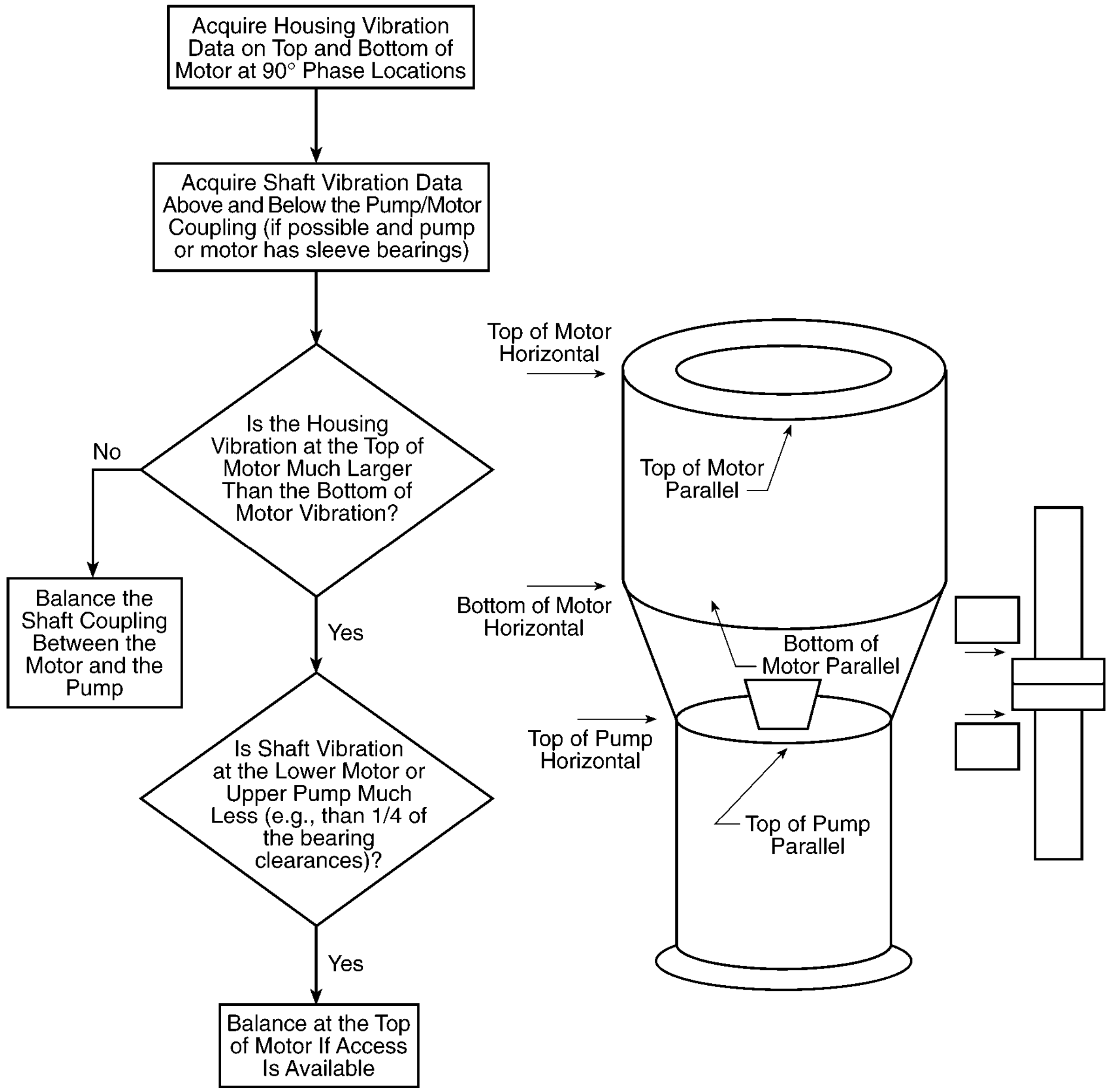


Figure 3-14
Field Balance Process and Procedure for Vertical Pumps

4
VERTICAL PUMP MAINTENANCE AND REPAIR GUIDELINES

4.1 Overview

The maintenance and repair of vertical pumps should focus around alignment, fits, balance, and running clearances. It is necessary to understand the basic relationship between these aspects of the repair, so that one can ensure that the pump will operate reliably after repair. It is also these relationships that will help troubleshoot an unreliable pump, or a pump that once provided five or six years of service and now requires repair every one to two years. The service life expectation of a vertical pump should be six years or greater.

There are two aspects of vertical pump alignment. The first is the alignment of the pump internal stationary components, and the second is the pump-to-motor alignment. The stationary components establish the position of the bearings and wear rings in the pump. The alignment registers of a stationary component position the mating stationary component(s) relative to its bearings and wear rings and vice versa. It is imperative that the alignment registers have the same centerline as its bearings and wear rings. In a properly repaired pump, the assembled stationary components (for example, suction head, case(s)/bowl(s), column(s), discharge head) have the bearings and wear rings on the same centerline. It is not practical to line bore a fully assembled vertical pump, so the components must be repaired individually and assembled. The alignment of the internal stationary parts is controlled during the repair process.

The second alignment issue is the pump to motor alignment. Again, a properly aligned pump and motor have the same bearing centerlines. It is assumed by most, if not all alignment procedures for vertical pumps, that the pump stuffing box represents the center of the pump. The motor shaft is positioned in the center of its bearings and the motor is moved until it aligns with the centerline of the pump (stuffing box). Several methods are discussed in Section 4.4 and are based on the stuffing box centerline being the referenced point for centering the motor to the pump.

Key Human Performance Point

A key component in the repair process is making *the stuffing box centerline, the centerline of the pump* assumption valid.

Forces causing unbalance during pump operation are minimized or eliminated with a proper dynamic balance of the pump rotating components. In vertical pumps, the rotor weight radial force does not exist to load the bearings. As such, pump synchronous vibration usually dominates the vibration frequency spectrum. The shaft has the freedom to move within the bearing clearance, and the bearings perform more as bumpers than as rotor dynamic components in the system. Thus, a precision balance of the vertical pump rotating components is important to low synchronous vibration and long-term pump service life.

Key Technical Point

A precision balance of the vertical pump rotating components is important to low synchronous vibration and long-term pump service life.

The fit between mating components impacts the ability of the components to actually align on the same centerline. A component machined with centerline perpendicularity and concentricity of ±0.002 inch is connected with another component having centerline perpendicularity and concentricity of ±0.002 inch. The maximum amount of offset between the centerlines of the two components would be 0.004 inch, for a line-to-line of slightly interference fit between the two components. However, if the fit between the two components is 0.004 inch loose, the offset could increase to 0.008 inch. For a multi-stage pump application, such as the condensate and heater drain pumps where the bearing clearances are nominally 0.008 inch, by the time the fourth-stage bowl or case is installed, the first- and fourth-stage bearings could be making direct contact with the shaft. If not in contact, you have varying degrees of fluid film thickness in the bearing causing accelerated wear and opening of clearances, which lead to higher vibration, possibly bearing instability (that is, oil whirl), and reduced pump service life.

Running clearances between the pump wear rings, bearings, and pressure breakdown bushings define the pump efficiency, rotor dynamic characteristics, and mechanical seal or packing performance. These components are usually always corrected during a repair and might be the only criteria imposed during a repair, particularly if the repair is performed on-site. In the pump business, this would be termed a *ring and bushing* job.

Key O&M Cost Point

A *ring and bushing* job is the least expensive repair cost but can potentially be the most costly total cost when the service life of the pump is reduced, or the pump catastrophically fails on startup or in service and results in a plant trip.

Experience has proven that the perpendicularity, concentricity, and fit between components change during the service life of a pump. The fit between components will increase usually due to corrosion and erosion. The perpendicularity and concentricity of a component's centerline can change over time as the component goes through a natural stress relieve process. Therefore, almost without exception, it is imperative that these items be checked and corrected as required during the repair.

4.2 Repair Specifications and Repair Criteria

One has the option to depend on others to make sure a pump has been repaired properly or take a proactive approach and develop requirements or expectations for the repair in the form of a Repair Specification. The development of repair specifications can be a daunting task, but with the guidance in this section and the sample specifications provided in the appendices, it should be manageable. A repair specification provides the basic expectations or requirements for a repair, but there are many more benefits to developing and using repair specifications. A repair specification can be used to:

- Develop the repair work scope
- Obtain competitive bids from pump vendors, including on-site maintenance departments
- Evaluate vendor bids
- Evaluate the quality of the repair

The goal is to provide a mechanism or process, before a pump has been removed from service, by which competitive bids from vendors (including the on-site maintenance department) can be obtained and evaluated. Based on years of developing specifications and monitoring the repairs in progress, it is possible to write a repair specification that defines a base work scope, and to be able to handle the unknowns that are uncovered during the repair process.

Key O&M Cost Point

Based on years of developing specifications and monitoring the repairs in progress, it is possible to write a repair specification that defines a base work scope, and to be able to handle the unknowns that are uncovered during the repair process.

Once the bid process has been completed and the vendor has been chosen, the next step is to pull the pump and proceed with the disassembly and inspection. Based on the inspection results, the repair work scope can be finalized and a detailed job schedule developed.

The specification is for both the repair vendor and the equipment owner. It should clearly identify to the repair vendor what the component is that requires repair, and the requirements or expectations for that repair. It is also a handy reference for the equipment owner during shop surveillance, reviewing as-found data, work scope development, and evaluating completeness or thoroughness of the vendor's work. If it is information that you or the vendor might need, include it in the specification. If there is something learned during a repair, revise the specification to capture that information. If it is information that you or the vendor might need, include it in the specification.

Key O&M Cost Point

If there is something learned during a repair, revise the specification to capture that information.

See Appendix H for an example of a repair specification for APM pumps.

4.3 Pump Maintenance and Repairs

An outline or template for the development of a detailed repair specification has been provided in this section. The details will change based on the actual design, number of stages, type of bearings, running clearances, etc. of the pump scheduled for removal, inspection, and repair.

Visual Inspection

- Catalog all loose parts
- List any shipping damage or other visible damage before disassembly
- Visually inspect all components upon receipt and during disassembly for any abnormalities, defects, or unusual wear

Disassembly

- Record total rotor end float or vertical lift
- Record pump shaft stick up, relative to motor support plate
- Disassemble the pump and record rotor end float as stationary components are removed

Key Human Performance Point

It is important to verify shaft stick up during pump assembly, to ensure that no interference issues will arise between the pump and the motor shaft once installed at the plant.

Cleaning

- Steam clean, bead blast, and/or grit blast stationary components as required or appropriate to remove rust or scale and to prepare critical surfaces for nondestructive examination (NDE) and dimensional inspection.
- Bead blast the rotating components as required, except the shaft, to prepare critical surfaces for NDE and dimensional inspection.
- Clean the shaft using a stainless wire brush, or blast with aluminum oxide, walnut shells, or similar non-aggressive blasting media to prepare the surface for NDE and dimensional inspection.

Nondestructive Examinations

- Certified and qualified personnel shall perform *Magnetic* and *Liquid Penetrant* examinations

Dimensional Inspections

- The dimensional inspections outlined in the specification are not intended to provide a complete inspection scope, but should be completed as a requirement of the repair agreement.
- The customer reserves the right to add inspection requirements up to and including 100% dimensional inspection of parts.
- Inspections required during pump disassembly should also be performed and documented during pump assembly.

> **Key Technical Point**
>
> Provide any special requirements or restrictions (hold points) that might be required, such as during pump disassembly and for inspection cleanliness.

Dimensional Inspections and NDE

This section details the pump components and their inspection requirements. Each component requiring evaluation for repair should be identified and the inspection requirements provided.

Identify each stationary component, its alignment registers, and the components that attach and align with it. The registers must be on the same centerline, identify the components by Item or Part Number on a cross-sectional drawing, which should be provided as an attachment to the specification.

Discharge Head

- Record diameter of the stuffing box extension, last-stage casing, and the shell/can/sole plate alignment registers, in a minimum of two places 90° apart.
- If required, record concentricity and perpendicularity of alignment registers, mating surfaces, flanges, and the motor support plate relative to the centerline of the register for the stuffing box extension.

> **Key Technical Point**
>
> Record concentricity and perpendicularity during disassembly only if repairs are not warranted by the dimensional inspection results but physical damage (for example, bearing wear, seal failures) indicates a problem or potential issue with the component.

- Repair alignment registers, mating surfaces, flanges, and the motor support plate as required to satisfy concentricity, perpendicularity, and fit criteria.
- Paint with plant-approved coating.

Key Technical Point

An option for the above step is to include machining alignment registers on the underside of the motor support plate to aid in pump alignment at the plant. This option should only be performed, however, if the machining of all registers can be performed or verified in the vertical position and on the same setup.

- Stuffing Box Extension Record the diameter of the discharge head alignment register (Part No.), and the inside diameter (ID) of the stuffing box bushing (Part No.) and/or bearing (Part No.) in a minimum of two places, 90° apart.
- Remove the stuffing box extension bushing and/or bearing. Record the outside diameter (OD) of the stuffing box bushing and/or bearing, and the ID of the fit location in the stuffing box extension, at a minimum of two places, 90° apart.

Key Technical Point

Many times, the bearings, bushings, or case rings are damaged during disassembly, such that the ODs of the bearings, bushings, and case rings cannot be obtained. However, this should be recorded during assembly and verified to meet fit criteria.

- If required, record concentricity and perpendicularity of the bushing fit location, alignment registers, mating surfaces, and stuffing box packing area, relative to the centerline of the stuffing box.
- Repair bushing fit locations, alignment registers, mating surfaces, and the packing area, as required to satisfy concentricity, perpendicularity, and fit criteria.
- Replace the stuffing box extension bushing (Part No.).
- Paint to match discharge head.

Problems have been experienced with Ingersoll-Dresser Pump (IDP, formerly Ingersoll-Rand and Pacific Pumps) RHR pumps supplied by Westinghouse Electric Corporation—IDP models 8X20 WDF (close-coupled), 8X2OWD (close-coupled), and 8X18, 8X19, 8X20, 8X22, and 8X24 SPF (coupled) supplied by Westinghouse Electric Corporation. The term *close-coupled* refers to pump assemblies that have a motor shaft extending directly into the pump bowl. The term *coupled* refers to pumps that use a spacer coupling between the pump and motor.

Information on pump seizures was identified by Westinghouse in Technical Bulletin ESBU-TB-96-03-R0 dated 6/20/96. The stuffing box extension on this pump can distort during operation, leading to alignment issues that could develop into internal rubs.

Other items specific to these pumps are the installation of the shaft sleeve and verifying that the step in the shaft sleeve is within ±1/32 inch of the face of the stuffing box extension. This dimension can be affected by distortion of the stuffing box extension, length of the shaft extension, height of the motor mount, and height of the stuffing box extension. Exceeding the ±1/32-inch dimension may cause leakage past the rotating face O-ring that seals to the shaft sleeve just below the step.

Casing distortion has also been observed in these pumps. The pump casing ID should be 32.218 to 32.220 inches. The stuffing box extension OD should be 32.214 to 32.216 inches. The flatness of the stuffing box extension gasket surface may also distort. To correct these conditions, the casing should be machined to achieve the factory dimensions or to be within 0.002 inch of being round.

First-Stage Casing, Intermediate Casing(s), Last-Stage Casing

- Record the diameter of the alignment registers and the ID of case bearings (Part No.) and case rings (Part No.) in a minimum of two places, 90° apart.
- Remove casing bearings and rings. Record the OD of the bearings/rings and the ID of fit locations in the casing in a minimum of two places, 90° apart.
- If required, record concentricity and perpendicularity of alignment registers, mating surfaces, and casing ring fit location relative to the casing bearing centerline.
- Examine by NDE the casing guide vanes from the vane inlet to approximately 6 inches along the vane.

Key O&M Cost Point

The casing vane NDE recommendation is intended primarily for cast iron casings. Cast iron casings crack in this area due to stress cracks (sometimes microscopic) that are induced during manufacturing and that propagate over time. In most cases, the casings can be salvaged by grinding out the crack and reshaping the inlet vane. Cracks extending as much as 5 inches along the vane have been removed without affecting pump performance.

- Machine alignment registers, mating surfaces, and bearing/ring fit locations as required to meet concentricity, perpendicularity, and fit criteria.
- Replace casing bearings (Part No.) and casing wear rings (Part No.).

Suction Head

- Record the diameter of the alignment register, the ID of the suction head bearing (Part No.) and the case ring (part no.) in a minimum of two places, 90° apart.
- Remove suction head bearing and wear ring. Record ODs of the bearing and wear ring, and the ID of the fit locations in a minimum of two places, 90° apart.
- If required, record concentricity and perpendicularity of the alignment register, mating surfaces, and suction head ring fit location relative to the suction head bearing centerline.
- Machine align the registers, mating surfaces, and bearing/ring fit locations, meeting concentricity, perpendicularity, and fit criteria.
- Examine by NDE the suction head bearing supports (spider).

Key Technical Point

NDE of suction head is intended for cast iron suction heads. Cast iron suction head bearing supports crack due to stress cracks (sometimes microscopic) being induced during manufacturing that propagate over time. If cracks are found, grind to remove them and evaluate the strength of supports.

- Replace the suction head bearing (Part No.) and the suction head wear ring (Part No.).
- Inspect the suction head bearing lube line. Verify that it is free of debris or blockage and acceptable for continued use. Replace if required.
- Inspect the suction head bearing to verify that the lube line aligns with the hole in the bearing.

First-Stage Impeller, Second-Stage Through Last-Stage Impellers

- Perform 100% crack inspection of impellers using magnetic particle or liquid penetrant examination.
- Record the OD of impeller wear ring turns and impeller shaft bore in a minimum of two places, 90° apart.
- Weld repair impeller wear ring turns and machine them to meet running clearance criteria.

Key Technical Point

"Weld repair impeller wear ring turns" applies to impellers that do not have replaceable wear rings. One option, to weld repair the impeller wear ring turn area, would be to undercut the ring turn and add a replaceable wear ring.

- Repair the impeller bore as required to meet fit and concentricity tolerances.
- Individually balance each impeller to achieve a maximum residual unbalance of 1 W/N per balance plane.

Pump Shaft, Upper Shaft

- Perform a 100% crack inspection of shafts using magnetic particle or liquid penetrant examination.
- Record shaft total indicated run-out (TIR) at each bearing journal, impeller fit, coupling fit, and journal sleeve location.

Key Technical Point

Shaft straightening is not a preferred option, due to the memory of most shaft materials. If possible, undercut coat journals and fits, and precision grind to get these components on the same centerline. If unbalance results, it should be corrected during rotor balancing.

- Record the shaft OD at bearing journal(s), impeller(s), shaft coupling(s), and journal sleeve(s) locations.
- Remove the suction head bearing journal sleeve (Part No.) from the pump shaft and the stuffing box extension shaft sleeve (Part No.) from the upper shaft. Record the ID of the journal sleeve and the OD of the shaft fit location in a minimum of two places, 90° apart.
- Replace the suction head bearing journal sleeve (Part No.) and the stuffing box shaft sleeve (Part No.).
- If required, repair the bearing journal, the line shaft coupling fit, and the impeller fit locations of the pump shaft.
- If required, repair the stuffing box shaft sleeve and the line shaft coupling fit locations of the upper shaft.
- For the suction head bearing journal sleeve, install three setscrews, equally spaced around the circumference of the sleeve, to hold the sleeve into place.

Key Technical Point

APKD pump designs without setscrews to hold the suction head bearing journal sleeve in place have had failures. The setscrews prevent the sleeve from resting on the anti-rotation key and causing the key to shear. The sleeve continues to slide down on the shaft, making contact with the suction head. The original equipment manufacturer (OEM) offers a modified design that combines the impeller locknut and journal sleeve functions. The new design should work, but the use of the setscrews has proven to provide years of trouble-free service.

- Prepare and burnish the upper shaft near the drive coupling fit for proximity probes.

Key Technical Point

Burnishing the shaft for proximity probes would be a good idea even if permanent proximity probes are not installed at the facility, because troubleshooting might require the installation of temporary proximity probes.

Shaft Coupling

- Record the ID of the shaft coupling at shaft fit locations.
- If required, record the perpendicularity and concentricity of the shaft fit bores.
- If required, repair the perpendicularity and concentricity, and the shaft fit locations of the shaft coupling.
- Replace the two coupling keys with a single coupling key (see Figure 4-1).

Key Technical Point

A single, fitted coupling key is recommended so that the entire length of the coupling keyway is filled. The filled keyway reduces the unbalance inherent in the original design of the coupling with two keys. Also, the single key can aid the alignment of the adjoining shaft while the coupling is being installed.

- Mandrel balance the shaft coupling to 1 W/N maximum residual unbalance.

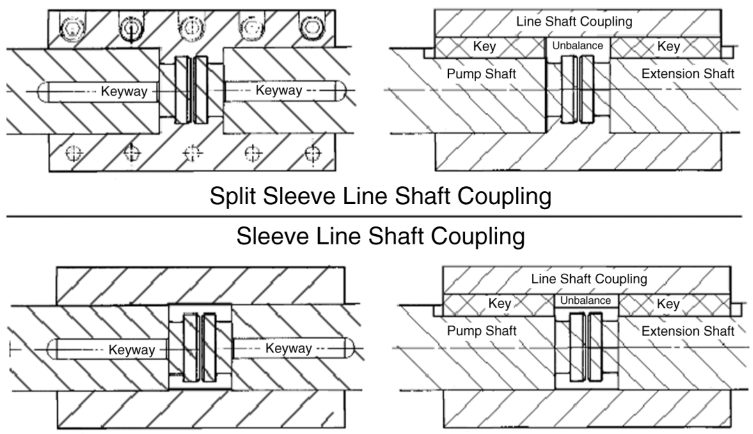


Figure 4-1
Axial Split Line Shaft Coupling with a One-Piece Key

Rotor Balance

- Install the second-stage through last-stage impellers on the pump shaft.
- Place the rotor in the balance stand with the impellers between the supports.
- Dynamically balance the rotor using the second-stage and last-stage impellers as the final balance planes to 4W/N of maximum residual imbalance.
- Install the first-stage impeller on to balanced rotor assembly.
- Balance the first-stage impeller as a single plane, over hung weight to 4 W/N maximum residual imbalance.

> **Key Technical Point**
> Balance criteria are normally considered a total residual unbalance allowable. Divide the criteria by two for the per plane balance tolerance.

Miscellaneous

- Restore proper fits for keys, split rings, and so on.
- Base work scope should include the replacement of consumables such as gaskets, keys, split rings, and so on.

Repair and Assembly Criteria

The repair and assembly criteria outlined in this section are based on industry standards, OEM input, lessons learned, and plant documentation (pump instruction manual). When the as-found condition of a pump is evaluated against these criteria the pump owner has the option of accepting a component that does not meet any of the criteria; however, because the contract was awarded based in part on the repair vendor’s stated ability to be able to perform the tasks outlined within the specification, the pump owner should not accept new or repaired parts that do not meet the criteria.

- Shaft straightness criteria shall be a Total Indicated Run-out (TIR) of a maximum of 0.002 inch. **Note:** Typical manufacturing standards allow 0.0005 inch per foot with a maximum of 0.005 inch.
- Balance the pump shaft and impeller(s) as an assembly to a maximum residual unbalance criteria of 4 W/N.

> **Key Technical Point**
> The calculation W/N, where W equals the weight of the component being balanced and N equals the operating speed at the plant, yields balance criteria in ounce-inches (oz-in). This calculation provides total residual unbalance criteria; divide by 2 to obtain unbalance per plane.

- Mandrel balance the line shaft coupling as an assembly to a maximum unbalance of W/N.
- Male and female registers and flanges should be repaired as required to obtain concentricity, perpendicularity, and parallelism within 0.002 inch.

Key Technical Point

Restoration of concentricity, perpendicularity, and parallelism is a machining process. Welding is required only if fit does not meet the criteria.

- Fit criteria should be 0.002 inch clearance for male and female registers. Pad weld alignment registers a minimum of eight locations to restore unacceptable fits.

Key Technical Point

An interference fit of up to 0.002 inch for male and female registers is often used in vertical pumps, particularly in condensate and heater drain pump service where bearing clearances are small and maximum reliable service life is desired.

Key Technical Point

An even number of pad welds (if not eight) should be required, so that there would be two pads 180° apart where the diameter for the register can be verified. A 100% weld is acceptable and might be preferable, depending on register size and material. Minimize welding and heat to cast iron parts.

- Impeller(s) shall be repaired or machined as required to remove nominal damage and to obtain fit to shaft of not greater than 0.002 inch loose on the diameter.
- Impeller liner/shroud and impeller vane edges shall be machined as required to obtain proper contour. The maximum gap between the vane and liner/shroud should not exceed 0.010 inch.

The previous step applies only to semi-open and open impellers. In the case of a multi-stage pump, it applies to each stage as the pump is being built.

- The bearing ID to journal sleeve OD clearance can be obtained from the vendor manual.
- The bearing OD fit in housings should be 0.000 inch to 0.002 inch clearance.
- Install all new bolting and fasteners.
- Coat wetted non-machined surfaces with coal tar epoxy or a customer-approved equivalent. Do not coat the impeller housing surface, which establishes impeller lift (running clearance) or running position. Paint non-wetted surfaces.

Key O&M Cost Point

Coatings are usually applied to pumps used in environments such as river water, lakes, circulating water, and so on. Coatings are usually not applied in condensate and heater drain applications.

- Assemble the pump, recording critical dimensions and package.
- Record the as-left or as-repaired condition of all items required to be documented during disassembly and inspection.
- Any criteria that are not specifically addressed shall default to manufacturer specifications.

Bid and Work Scope Evaluations

- Obtain a line item for the base work scope. The base work scope should not include performing the perpendicularity and concentricity checks outlined in the Detail Inspection section. After a review of the dimensional inspection results, a determination can be made to repair or leave the components as is. In the event of an acceptable dimensional inspection, the option to check the perpendicularity and concentricity will be evaluated. Repaired and new components should meet all criteria required by this specification.
- Obtain line item adders to repair registers and fits.
- Require a report documenting as found and as left condition. Record dimensions of critical parts, including diameters, perpendicularity, parallelism, concentricity, total indicated run-out (TIR) and fit clearances as applicable.
- Obtain parts pricing.

4.4 Vertical Pump Alignment

Vertical pump alignment is a key issue affecting reliable pump operation. Obtaining a proper internal alignment of the pump stationary parts is paramount to a sound repair/refurbishment program. Through the use of detailed repair specifications and oversight, confidence should be developed that the vertical pump has been aligned internally. The next step is a proper installation and pump-to-motor alignment in the field.

The first step in a proper field installation would be to have a level pump base or sole plate. This is particularly significant if the motor-to-pump alignment is performed using dial indicators on the pump shaft or pump shaft coupling, and if the motor has sleeve (babbitt) radial bearings with a Kingsbury (self-leveling) type of thrust bearing. Remember that the pump and motor shafts will hang plumb during alignment and operations. If the pump stationary components are not plumb, maximum pump reliability will never be obtained and, if a significant problem exists, failures at startup may be experienced. One situation where a problem with sole plate levelness could be noted is when significant shimming between the pump and motor is required to obtain angular alignment.

Key Technical Point

One situation when a problem with sole plate levelness could be noted is when significant shimming between the pump and motor is required to obtain angular alignment. It is recommended that the criteria for base (sole) plate levelness be 0.001 inch per foot of diameter, with a maximum of 0.005 inch total.

The discharge head and base plate mounting is the starting point for misalignment problems experienced in vertical pumps. Some manufacturers require a base plate that is level within 0.002 inch per foot of diameter. This might not sound like much but, for a pump suspended 15 feet below the pump base plate, the offset (radial deflection) between the pump's stationary parts and the pump shaft could be 0.030 inch. Most condensate and heater drain pump bearings have total radial running clearances of 0.007 inch to 0.009 inch. It is recommended that the criteria for base (sole) plate levelness be 0.001 inch per foot of diameter, with a maximum of 0.005 inch total.

Key Technical Point

The pump-to-motor alignment is about aligning the pump stuffing box centerline with the centerline of the motor bearings. Any of the procedures published, regardless of measurements being made at the pump coupling, shaft, or stuffing box, are designed to accomplish stuffing box and motor bearing centerline alignment. It is that fact that makes it imperative to establish the centerline of the pump bearings, wear rings, column registers, and discharge head registers, relative to the pump stuffing box centerline. It is imperative that the centerline of the pump bearings, wear rings, column registers, and discharge head registers, be ultimately corrected relative to the pump stuffing box centerline.

Most vertical pumps, where the entire rotor and liquid weight and pump thrust are supported by thrust bearings in the motor, employ rigid couplings between the pump and motor. Rigid couplings by definition provide no flexibility between the two coupled shafts; as such, rigid couplings require precise alignment. Flanged and threaded couplings are the most widely used rigid couplings. Flanged couplings consist of precisely machined flanges on each shaft, connected by a series of coupling bolts around the perimeter of the flanges, and an adjusting nut between the coupling flanges to enable setting of the pump lift (Figure 4-2). If the pump has a mechanical seal instead of packing in the stuffing box, a spacer piece has been incorporated. This spacer facilitates installation and removal of the seal without removal of the motor (Figure 4-3). Threaded couplings are more commonly used to connect the line (internal) shafts of vertical turbine pumps, but are also used to couple pumps to motors (particularly hollow shaft motors). These couplings are cylindrically shaped with internal threads matching the external threads of the shaft. Flanged couplings are more common in the size of the pumps that are identified in this guide.

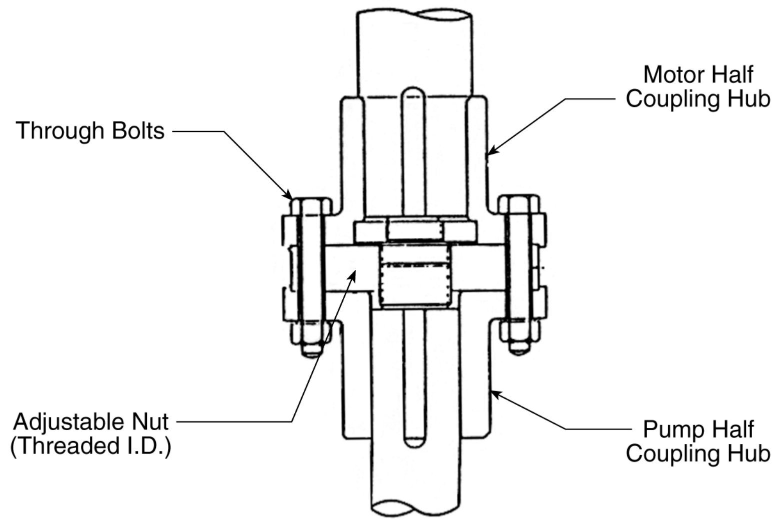


Figure 4-2
Rigid Coupling Typical of Vertical Pumps with Packing

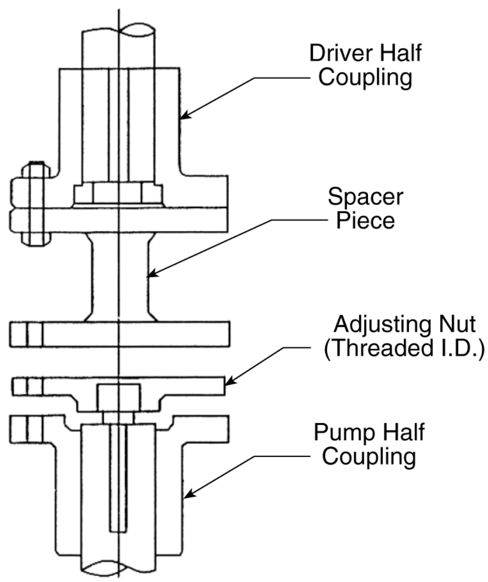


Figure 4-3
Rigid Coupling Typical of Vertical Pumps with Mechanical Seals

Figure 4-4 illustrates a common alignment method, where wedges are used to center the pump shaft in the pump stuffing box. A dial indicator attached to the motor coupling hub is set to indicate on the pump shaft as the motor shaft rotates. The dial indicator readings provide the parallel offset (misalignment) between the two shafts. The angular misalignment can be obtained by setting the dial indicator on a surface perpendicular to the pump shaft centerline, such as the pump coupling half, stuffing box, or end of the pump shaft. It is recommended that the pump shaft be centered and perpendicular within 0.001 inch on a side or 0.002 inch TIR.

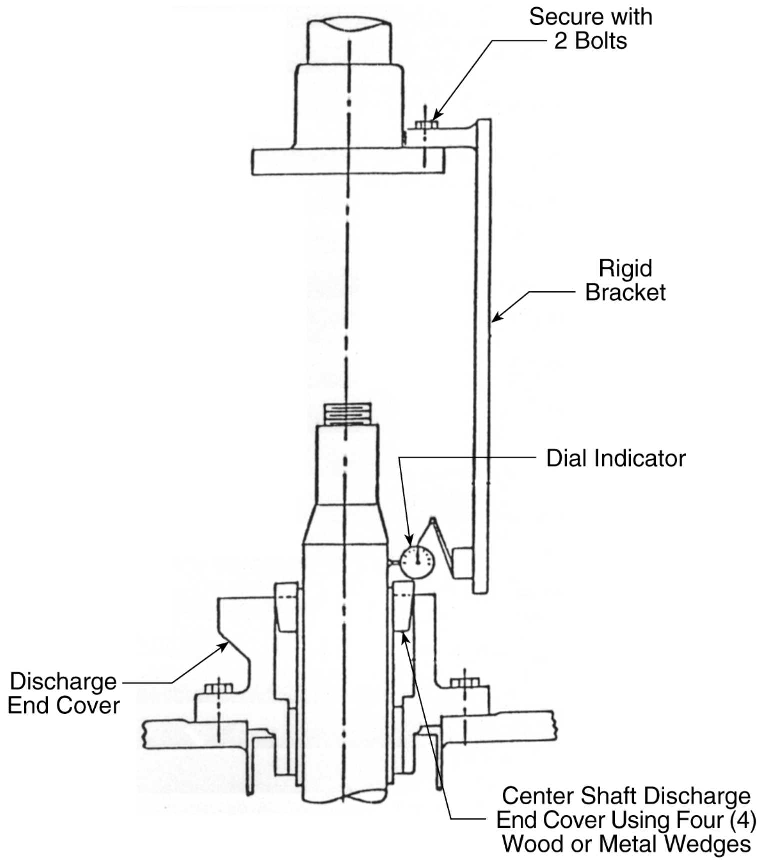


Figure 4-4
Shaft Alignment Using Wedges to Center Pump Shaft

Figure 4-5 illustrates a method where a bracket attached to the motor coupling half supports a dial indicator in such a way that direct readings inside the stuffing box can be obtained. The top of the stuffing box would be indicated for the angular alignment readings, but the end of the shaft or perpendicular face on the pump coupling half could be used. Again, the motor shaft must be rotated to obtain the parallel and angular offset between the two shafts.

Numerous ways exist to obtain a proper alignment; the creativity required is dictated by the physical restraints or confinements associated with the existing pump and shaft sealing arrangement.

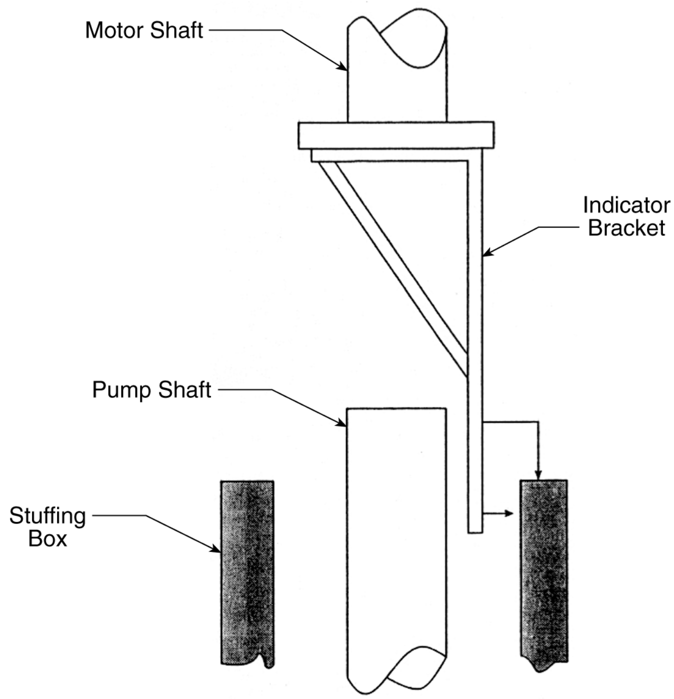


Figure 4-5
Shaft Alignment Using Bracket to Measure Inside and Top of Stuffing Box

In an effort to speed up and increase the reliability of the alignment process, a *tapered alignment collar* can be made to fit into the stuffing box. As shown in Figure 4-6, the tapered alignment collar provides the surfaces for taking both the parallel and angular offset readings. A properly manufactured alignment collar can:

- Eliminate the time-consuming process of centering the shaft in the stuffing box
- Provide readily assessable machined surfaces for running dial indicators

The tapered collar is split axially to allow for installation and removal with the pump coupling half installed. When the tapered alignment collar is manufactured, it is important that the split lines be ground or lapped smooth. To manufacture the collar, first rough machine the basic shape of the collar. Then axially split the collar and grind or lap smooth the split line. Clamp the two halves together and install a dowel pin at each split line prior to final machining; be sure not to clamp and dowel in areas where the taper and alignment surfaces will be machined. Finish bore the tapered collar. The finish bore should be machined larger than the pump shaft by at least 0.050 inch so that the collar can be installed around the pump shaft without much interference. The bore will be the location for attachment in the lathe. It is important that the tapered collar be placed in the lathe in such a way that the taper and two aligning surfaces can be machined without changing the collar setup in the lathe. This usually requires mounting the collar in the lathe such that the tapered end is toward the lathe spindle. A half-degree taper over a 1 to 2 inch length should be adequate. Calculate the midpoint of the taper length to coincide with the ID of the stuffing box. A 1 inch machined surface for the alignment registers should be adequate, but the larger the diameter for the alignment surfaces, the better the accuracy. For installation, clean the inside surface of the stuffing box; position the two tapered collar halves around the pump shaft; lightly tap in the dowel pins, and lower the tapered collar into the stuffing box.

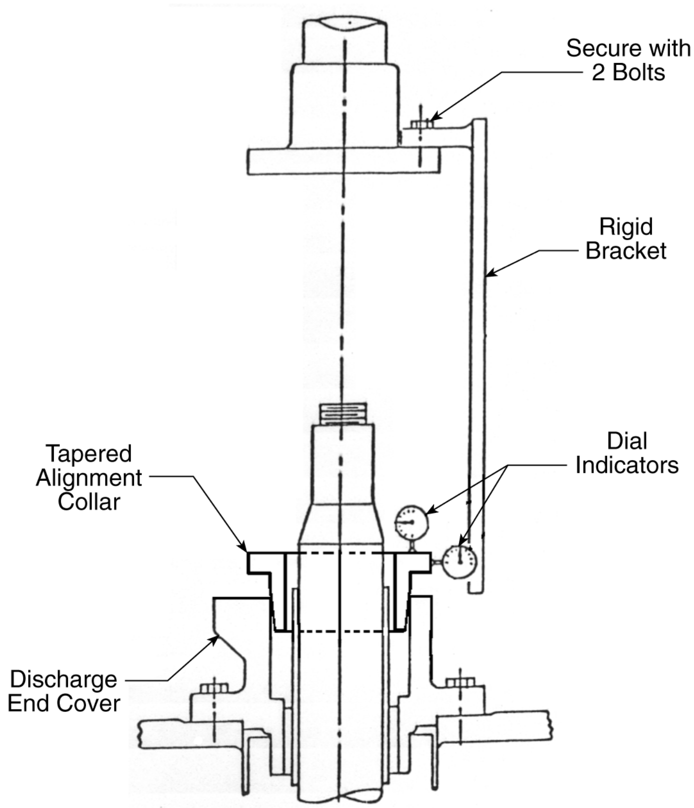


Figure 4-6
Alignment Method Using Tapered Alignment Collar

With each of the techniques described above, the motor shaft must be rotated as indicator readings are taken on the stationary pump shaft/coupling, stuffing box, or alignment collar. For large motors, getting the motor shaft to rotate can be difficult due to the weight of the rotor.

Pump and motor alignment first requires that the motor shaft be centered in its lower journal bearing. This is accomplished by pushing the shaft from side to side in two orthogonal directions and setting the shaft at the midpoint of the total travel. Jack bolts with brass tips or roller bearings can be used to carefully push the shaft back and forth and to set the shaft at the midpoint of the total travel. It is very important to have prior knowledge of the expected lower bearing total clearance. With babbitt bearings and a strong hand, it is easy to obtain an erroneously high total travel, and the shaft can be set significantly off center. For example, if a 0.022 inch travel is obtained for a bearing having nominally 0.011 inch to 0.014 inch clearance, and the shaft is set at the midpoint of 0.022 inch, the motor shaft could be hard against the journal bearing. This condition could lead to wiping the motor bearing at startup. If the total travel does not make sense, double check the reading using a different person, method, or both. It could also be helpful to review previous alignment history records.

Key Technical Point

If the total travel does not make sense, double check the reading using a different person, method, or both.

Once the center of the lower motor bearing has been determined, the shaft must be secured and still be capable of rotating without affecting its center position. Jack bolts with brass tips or rolling element bearings in contact with the shaft are usually used to maintain the motor shaft center. With the shaft secure in the bearing centerline, devices such as cheater bars or a come-along are often needed to rotate the shaft. Rotating the pump/motor shaft from the top of the motor might provide steadier shaft rotation and more reliable alignment readings, but it will require the removal of the motor end cover. Usually, once the shaft has been rotated, an oil film is established between the thrust bearing pads and the thrust block, and the shaft rotation is not as difficult. For motors with rolling element bearings, the alignment procedure is greatly simplified because the shaft centerline is fixed by the bearings.

An alignment tool has been developed at one utility, which could be attached to the motor coupling half and rotated while the motor shaft remained fixed. This method proved to improve the accuracy, timeliness, and ease of the alignment process. The alignment tool (shown in Figure 4-7) consists of a stationary part that is bolted to the motor coupling half and a rotating component that is connected to the stationary component by a precision rolling element bearing. Dial indicators attached to extension rods are used to center the alignment tool to the motor shaft. The dial indicators are turned around so that parallel and angular offset readings can be obtained on the shaft, the stuffing box, or the tapered alignment collar.

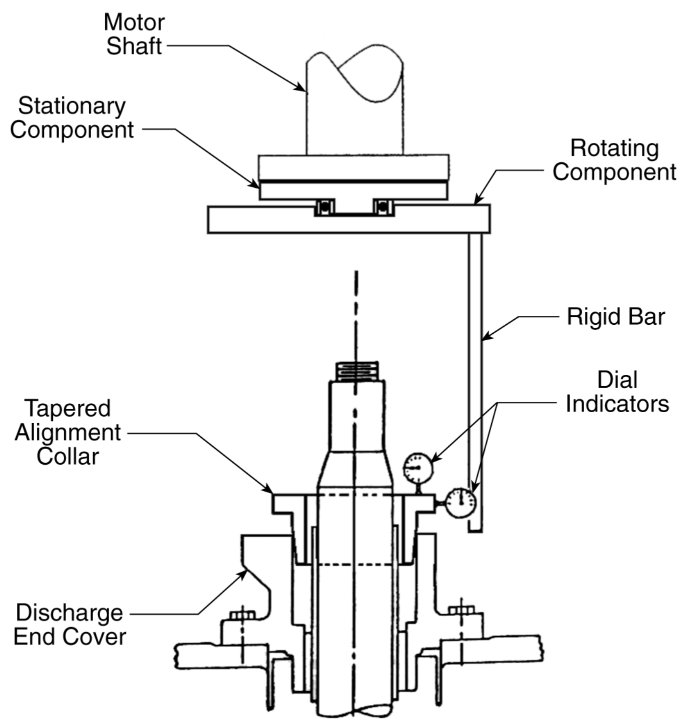


Figure 4-7
Vertical Pump Alignment Tool

Vertical Pump Laser Alignment

As more and more facilities upgrade to laser alignment equipment, questions arise regarding its effectiveness in vertical pump applications. The opinions run from "it cannot be used" to "I am using it at my plant." Laser alignment, like the reverse dial indicator method, requires that the motor and pump shaft be rotated to obtain the parallel and angular offset between the two shafts. In a vertical pump, this would require that the two shafts be coupled to enable the pump shaft to be rotated. As such, once the alignment process is completed, the pump must be uncoupled to set the pump lift and to install the mechanical seals or packing. Even if laser alignment techniques can be used to center pump and motor shafts relative to each other, the motor shaft must still be centered in the lower motor bearing and aligned with the pump stuffing box centerline. It is not clear how advantageous laser alignment of vertical pumps can be, however, a case study has been provided in Appendix I.

4.5 Cavitation Erosion, Abrasive Erosion, and Corrosion

Pump impellers, casings/bowls, and their related components can be damaged by a number of different actions, the most common being cavitation erosion, abrasive erosion, and corrosion. The appropriate repair procedure will depend on the cause of the damage.

For example, in one case, a screen wash pump failure was experienced due to galvanic corrosion. This is because material upgrades can create a dissimilar metal condition. Column upgrade to 316L material with a bowl assembly of 3% nickel iron can result in galvanic corrosion of the iron material when used in brackish water. This particular failure was experienced after only eight years of service.

Two causes were identified in this instance. The first was improper configuration control because the columns—but not the suction bowls—were changed. This created a galvanic cell that over time accelerated the graphitic corrosion until the bowl flange material had no strength. The second cause was the lack of understanding that the two materials were dissimilar enough to create an adverse galvanic reaction.

Key Technical Point

Caution should be exercised when upgrading materials to preclude the creation of a potentially harmful galvanic environment.

Cavitation is the formation of vapor bubbles or cavities in a flowing liquid subjected to an absolute pressure equal to, or less than, the vapor pressure of the liquid. These bubbles collapse violently as they move to a region of higher pressure, causing shock pressures that can be greater than 100,000 psi. When audible, cavitation makes a steady crackling sound similar to rocks passing through the pump. Cavitation erosion or pitting occurs when the bubbles collapse against the metal surface of the impeller. It occurs most frequently on the low-pressure side of the impeller inlet vanes. Cavitation cannot only severely damage the pump, but it can also substantially reduce its capacity and therefore lessen efficiency.

Key Technical Point

Cavitation, erosion, or pitting occurs when vapor bubbles are transported into higher-pressure regions, causing them to form and collapse violently against the metal surface of the impeller. It occurs most frequently on the low-pressure side of the impeller inlet vanes.

Abrasive erosion is the mechanical removal of metal by suspended solids, such as sand, in the liquid flowing through an impeller. The rate of abrasive wear is directly related to the velocity of the liquid, so wear will be more pronounced near the exit vanes and shrouds of pump impellers where the liquid velocity is highest.

Key Technical Point

The rate of abrasive wear is directly related to the velocity of the liquid, so wear will be more pronounced near the exit vanes and shrouds of pump impellers where the liquid velocity is highest.

Corrosion damage to submerged or wet metal is the result of an electrochemical reaction. The electrochemical reaction occurs when a galvanic cell is created by immersing two different elements in an electrolyte, causing an electric current to flow between the two elements. The anode, or the positive electrode of the cell, gradually dissolves as a result of the reaction. With the water acting as an electrolyte, irregularities (such as variation in surface finish or imperfections in the metal's composition) create small galvanic cells over the entire surface of the metal. Corrosion damage occurs as the anodes of these cells dissolve. Corrosion, unlike abrasive erosion, is generally independent of the liquid velocity. Pitting caused strictly by corrosion will be uniform over the entire surface.

Key Technical Point

Corrosion, unlike abrasive erosion, is generally independent of the liquid velocity. Pitting caused strictly by corrosion will be uniform over the entire surface.

Diagnosis of the problem can be difficult as the damage can be caused by more than one action. As a metal corrodes, the products of corrosion form a protective film on the metal surface. This film protects the base metal from further corrosive attack. An erosive environment will tend to remove this film, leaving the metal susceptible to corrosion damage. Similarly, where cavitation erosion is occurring, the metal will be prone to further damage from corrosion.

Severe erosion or corrosion damage might warrant the replacement of the damaged parts with parts constructed of a material that is more erosion or corrosion resistant. If severe cavitation erosion occurs during normal operation, a new or modified impeller or other design changes might be required. Obviously, replacing an impeller or other major components can be a very expensive endeavor and should only be done after careful economic analysis. Some factors to take into consideration when making an analysis are the cost and effectiveness of past repairs and any gain in efficiency or output that might be obtained by replacement. Modification to the existing impeller geometry, as discussed later in the section, normally proves to be very cost-effective.

Key O&M Cost Point

Except for severe cases, repair instead of replacement is the most economical solution.

The repair procedure will depend on the cause of the damage. Welding is the most successful repair method of cavitation damage. Repair with non-fusing materials, such as epoxy and ceramic coatings, is generally not successful because the low bond strength of these materials, usually less than 3,000 psi, is not capable of withstanding the high shock pressures encountered during cavitation. However, it has been used successfully in large vertical, low-speed (<500 rpm), mixed-flow pump applications, such as circulating water pumps. In circulating water pump applications, epoxy coatings have been applied for vane geometry corrections to eliminate cavitation regions and to provide a protective surface against the erosive/corrosive environment of the circulating water system.

For raw water pump applications, common defects identified after long service (~25 years) for fabricated steel-type vertical pumps include the following:

- Erosion/corrosion at weld fusion lines,
- Cavitation damage at the casing shroud adjacent to impeller trailing edge. Typical wall thickness in this region is generally 1 inch, and damage/pitting was found to be as high as 50% of the wall thickness.
- Impeller vane thinning due to erosion.

4.5.1 Impeller Cavitation Evaluation and Remedies

As pump energy levels have increased through the years, so has cavitation erosion damage in the suction stage impellers. This occurs because the higher head of a reasonably efficient centrifugal impeller is accompanied by a correspondingly greater peripheral speed of the impeller blades. This, in turn, requires more NPSH, with the consequent increase in erosive pitting of the surfaces as bubbles collapse nearby. Progress has been made in extending the life of these suction-stage impellers, but the problem has become complicated by the unsteady two-phase flow which results from operating at flow rates other than the pump *best efficiency point* (BEP).

Studies of cavitation at both design and lower pump flow rates have led to a new understanding of the role that the two-phase flow phenomenon plays in pump hydraulic instabilities. Flow visualization research revealed oscillation of the unsteady two-phase flow behavior in which cavities on the suction, or visible, side of the blades are seen to oscillate rather violently as the pump flow rate is reduced below that of the BEP. This oscillating cavitation has been recognized as a major source of the fluid/structure interactions that are characterized by pressure pulsations and associated forces. If oscillating cavitation occurs in a high-energy pump, excessive stresses in the pump and system structures can result.

The goal then becomes one of reducing the total volume of vapor or cavity volume that can exist within the impeller through the introduction of new design features and practices. This increases the pump's resistance to cavitation erosion and increases its reliability through elimination of the oscillating cavitation occurring in the suction stage of the pump. The primary zone of cavitation within an impeller is the suction (visible) surfaces of the blades. This is where most of the total cavity volume exists.

For flow rates at or below the BEP, cavities occur on the suction (visible) or trailing faces of the blades. At flow rates sufficiently below that of the BEP, recirculation breaks up these cavities, creating an unsteady two-phase flow in which bubbles are scattered within random vortices generated by the shearing action of the emerging backflow. Some of these bubbles collapse on both the pressure (non-visible) and suction (visible) sides of the blades, and some within the main flow stream. This chaotic flow configuration no longer approximates a cavity, and the understanding of it could yield a reliable life prediction.

It may be appropriate to specify a replaceable wear ring with a stainless steel suction bell instead of a cast iron or steel suction bell with no wear ring. This will prevent replacement of the suction bell due to erosion wear of this surface.

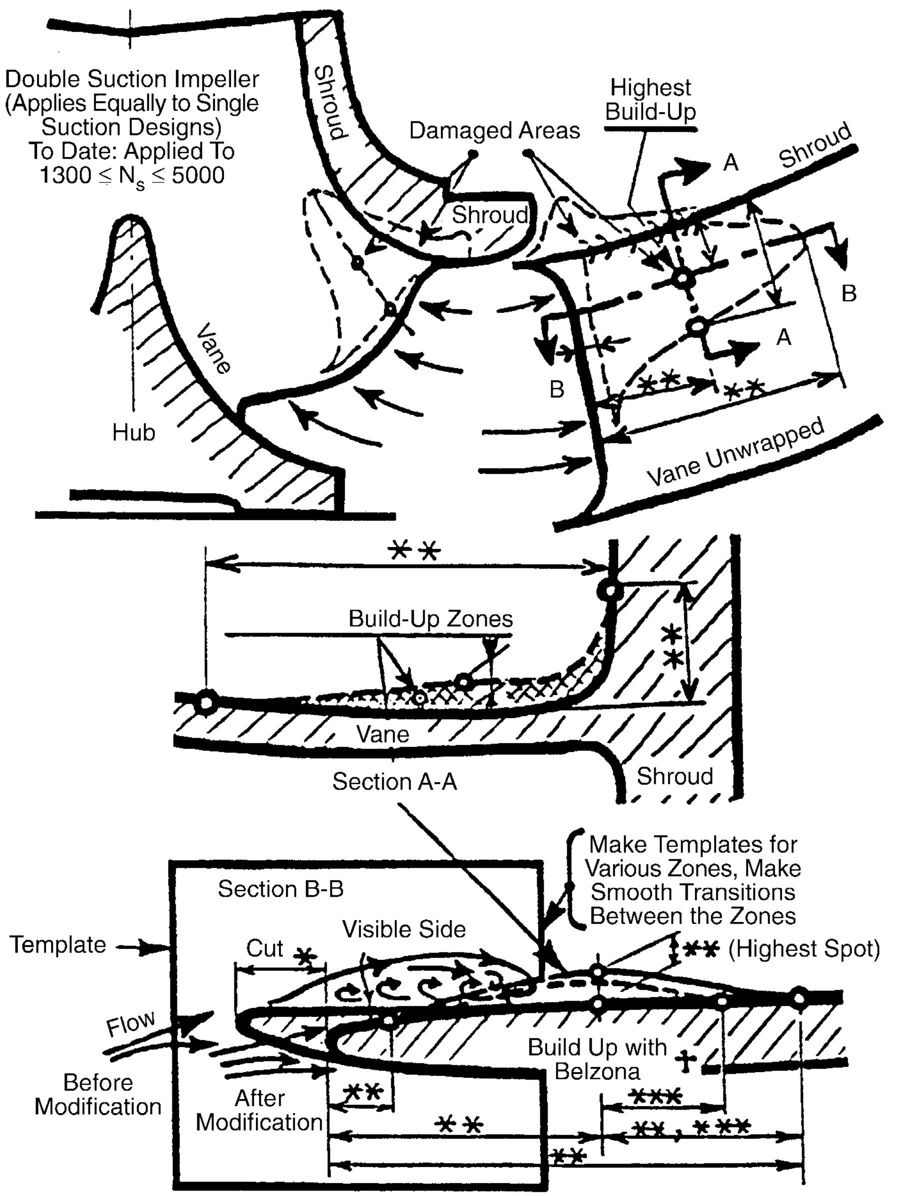
4.5.2 Anti-Stall Hump Development

Almost simultaneously, the following three hydraulic research scientists were independently working on modifications to prevent impeller inlet vane cavitation:

- Dr. E. Makay, Energy Research and Consultants Corp., Morrisville, PA
- P. Hergt, KSB, Frankenthal, West Germany
- Dr. P. Cooper, Ingersoll-Rand Pump Co., Phillipsburg, NJ

All three researchers developed similar modifications designed to prevent flow stall along the vane inlet surface.

Dr. Makay developed the anti-stall hump shown in Figure 4-8. The first experimental modifications were performed in 1976 and the first practical applications were performed on a series of circulating water pumps in 1983, with 218 applications implemented by 1993. The majority of the modifications were made on large circulating pumps, including some reactor coolant pump (RCP) applications in PWRs, and a smaller number of modifications were applied to high-energy-input large feed pumps.



* Determined by the amount of damage and the vane inlet geometry
** For dimensions, see file of each application. (customized for each case)
*** Depending on the amount of damage
† Belzona if low flow vel. and cast iron weld if possible

Figure 4-8
Example of Anti-Stall Hump Applied by Dr. E. Makay in 1983

P. Hergt patented his concept (Figure 4-9) in 1983 but the geometry was very restrictive. The patent specifies the shape of the hump line (that is, straight), limits the length of the hump line, limits the vane inlet angles (such as, 0° to 5° to hump line), specifies the inlet vane radius, and limits the design to radial and mixed-flow impellers. The limitations restrict the application so much that most all modifications fall outside the patent and, therefore, there are no known modifications in service in the United States that meet the patent's requirements.

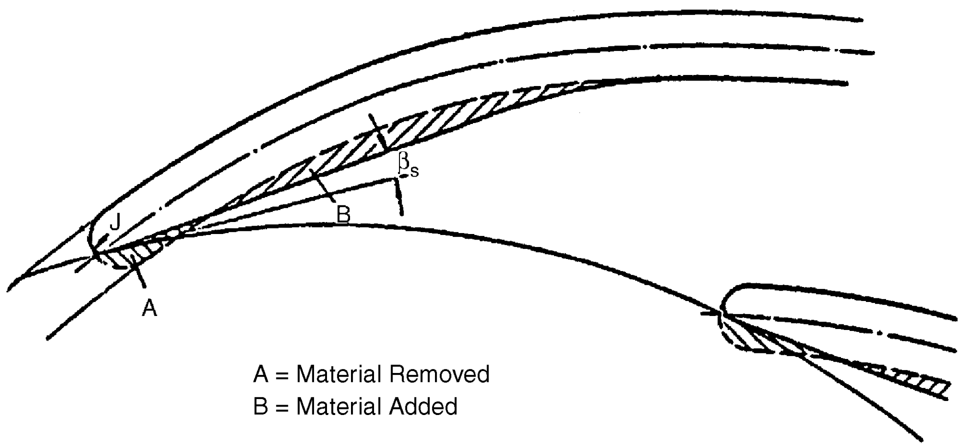
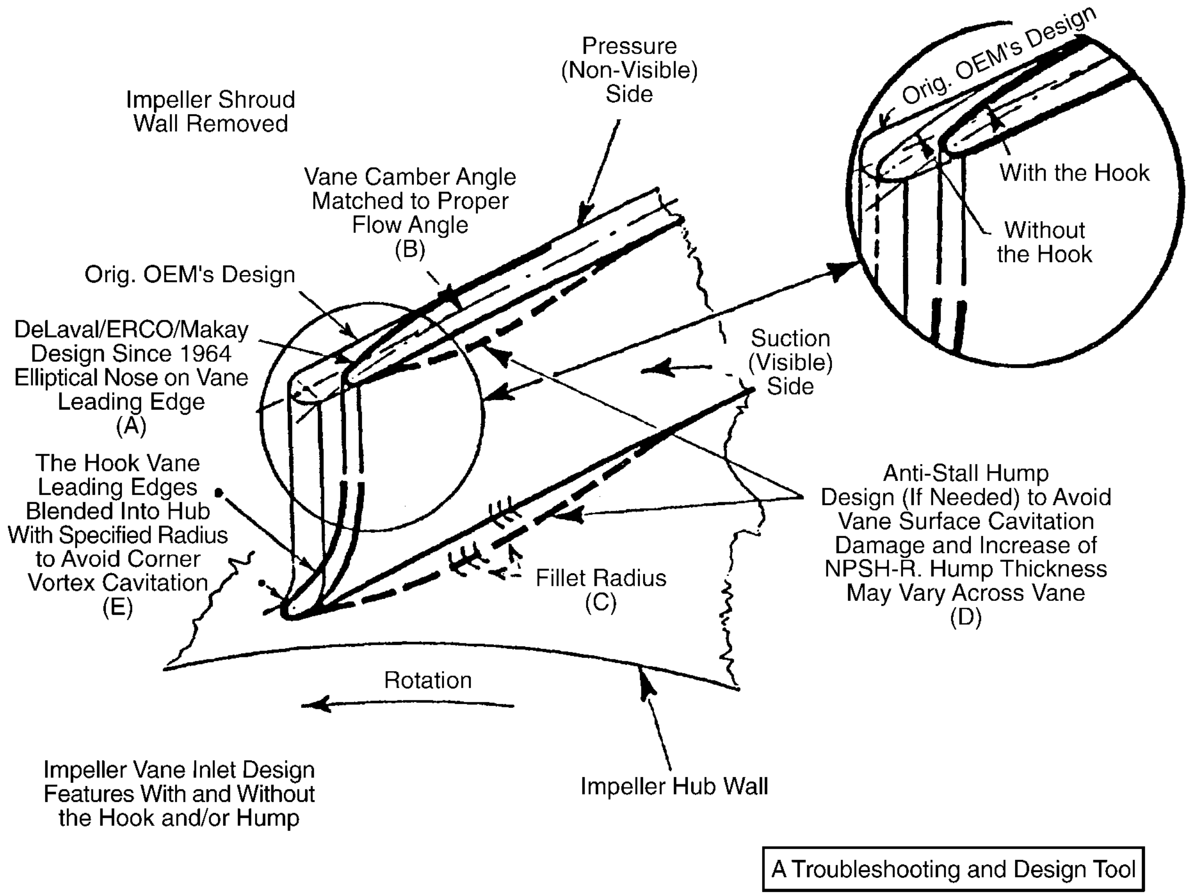


Figure 4-9
Design of Hump as Patented by Peter Hergt of KSB (Germany)

Dr. Cooper's version of the anti-stall hump was called the bias wedge, as shown in Figure 4-10. The first field application of the bias wedge did not occur until the late 1980s, and only at a handful of power stations with various reports of success. Likewise, the bias wedge has not been implemented as a modification to existing impellers; instead, it has been applied only in the design of new impellers.



Design Features of Proper Impeller Vane Inlet (Eye): The features shown on this representative vane correspond to the features listed above as (A) to (E). Each feature is integrated into the total design to ensure reduced cavitation activity with no adverse impact on overall performance levels (Patent applied by Ingersoll-Rand Co.). However, (A) & (B) used by DeLaval and ERCO (Makay) since 1964, (C) by more or less everybody, and (D) & (E) by ERCO (Makay) since 1976. NOTE: If the hump (D) is applied during troubleshooting, remember that each application is different; one solution may not apply to another (if the impeller is sand-cast, solution may be different from vane to vane). Thickness of the hump may vary across the vane, depending on the amount of the cavitation damage.

Figure 4-10
Proper Impeller Inlet Vane (Eye) Design Features (Including ERCO's Anti-Stall Hump, or, as Called by Dr. P. Cooper, the Bias Wedge)

The anti-stall hump developed by Dr. Makay was not patented. Due to the different impeller inlet vane geometries, inlet flow conditions, inlet cavitation damage, pump applications, and pump service requirements, a single design was not feasible to address all situations. Each case is unique and probably no two cases are identical. The designs of P. Hergt and Dr. P. Cooper addressed cavitation only on the suction side, but the anti-stall hump was applied with equal success on both the suction side and pressure side of the impeller inlet vanes by Dr. Makay.

If it is diligently applied, the advantages of applying the anti-stall hump are:

- The cost of modification is at least one order of magnitude less than the design of a new impeller, particularly considering costs associated with engineering, making a new pattern, casting, and the risk of design or manufacturing errors.
- The impeller is customized to its environment (for example, its operating point and inlet flow conditions).
- Starting out with a known impeller behavior or condition can be easier to correct and is more practical than designing a new impeller.

4.5.3 Anti-Stall Hump Case Histories in Circulating Water Pumps

The following methods address the typical cavitation damage observed on the suction side of impeller inlet vanes:

- Replace the impeller with a new impeller that has the same inlet vane design or repair (if possible) to the original design configuration.
- Replace the impeller in kind with upgraded material.
- Redesign the impeller, which can be a risky proposition because the people who developed the original design would probably be asked to develop the new design.
- Install the anti-stall hump.

Cavitation damage in large condenser circulating pumps can be addressed from the following two points of view:

- Eliminate cavitation damage at the impeller inlet (the impeller eye) using the anti-stall hump
- Increase the capacity of the pumps. In many cases, a 5–10% increase in circulating pump capacity can be obtained. The increased pump capacity can change the NPSH requirements of the pump, which normally computes directly to increased megawatts out of the plant turbine as a result of reduced condenser vacuum. The operating NPSH requirements of a mixed-flow, high specific speed pump (that is, circulating water pump) can be reduced by increasing pump performance if system requirements have the pump operating at 80% or less than its BEP.

Early in 1983, the inlet vanes of circulating pumps at a nuclear plant were modified as shown in Figure 4-8. The impeller vanes were thick and had experienced unusually large damaged areas. The inlet angle of the flow, and the location and shape of the stall shape/region were estimated. The inlet vanes of the impeller were cut back and shaped to better match the inlet flow angle and the stall region was built up on the vanes with Belzona®, an epoxy material. This material adhered very well to the rough surface caused by cavitation erosion and inlet vane velocities were low enough that the Belzona adhered and did not peel. Several inspections have shown that the epoxy remained on the steel surface with only minor, if any, surface damage experienced.

Since that time, several other circulating pumps have been successfully modified with various combinations of the modifications. The modifications/alterations cannot be applied uniformly to every case. Each case must be judged by its own characteristics in the following list:

- The nature, location, and amount of damage
- Inlet angles
- Vane thickness
- Vane shape
- Flow velocities
- Composition and temperature of the pumped fluid
- The effect of build-up on inlet flow area
- Changes in flow velocities
- Impeller material and size

Modifications of the anti-stall hump on a 48x47S-type circulating water pump are shown in Figure 4-11. Application of the anti-stall hump to a 114x78 WMCC circulating water pump is shown in Figure 4-12.

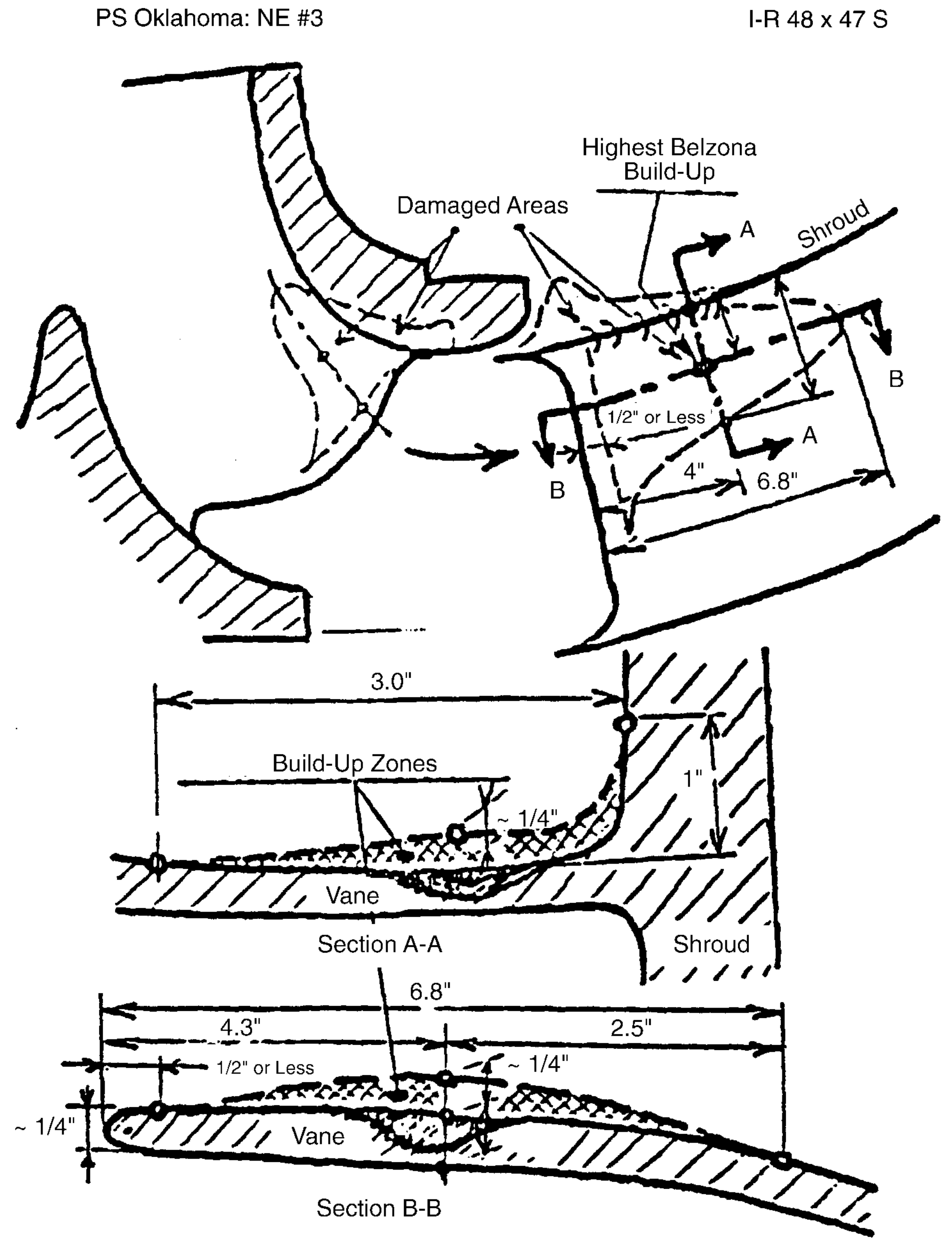


Figure 4-11
The Anti-Stall Hump Applied to a 48x47S-Type Large Circulating Pump at PSO's NE Unit 3 in 1990

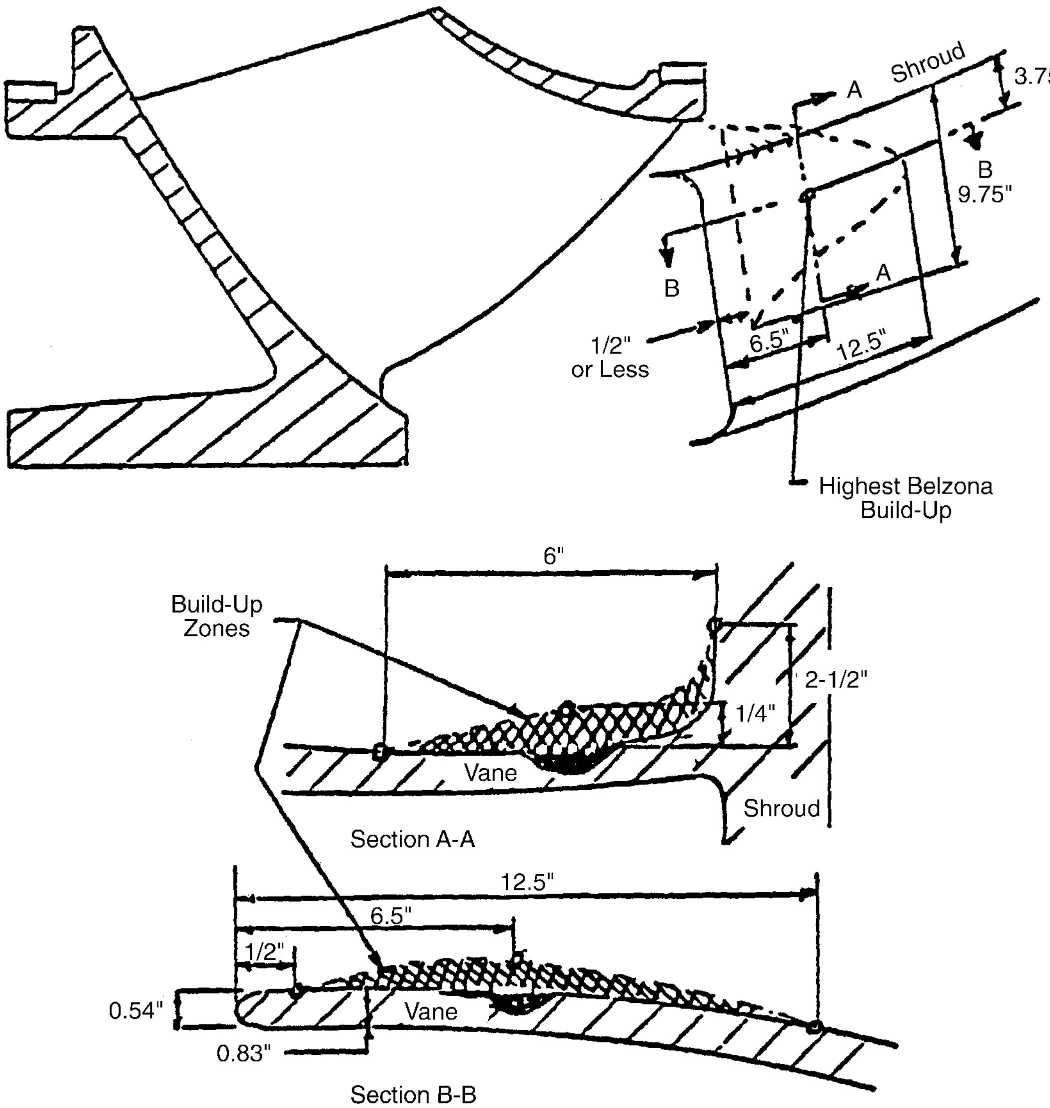
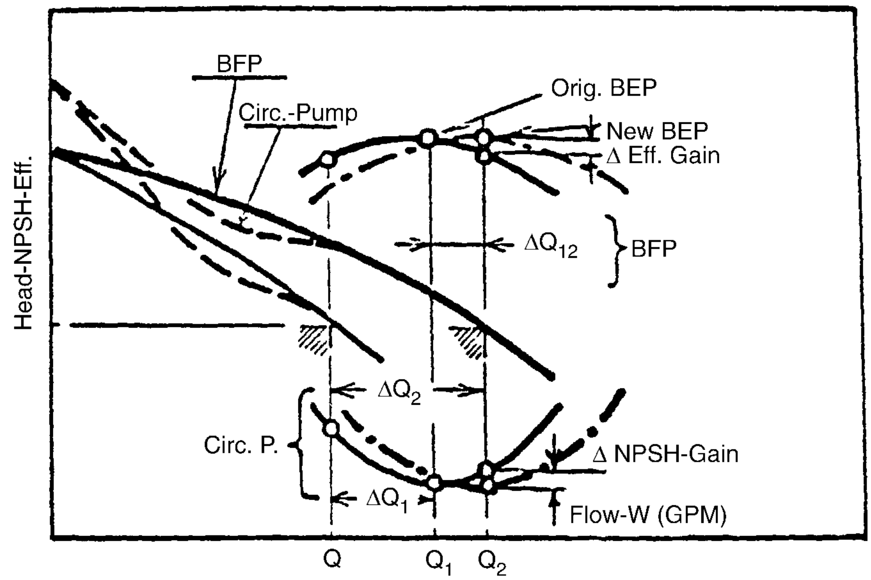


Figure 4-12
Application of the Anti-Stall Hump to an Al-Ch-Type 114x78 WMCC Circulating Pump in a Large BWR Nuclear Plant in 1992 (Belzona was used to build up the hump.)

Modifications of the single-stage, single-suction, vertical circulating water pumps at the Farley Nuclear station took place in 1983 (see Figure 4-13). The impeller material was carbon steel and the impeller eye diameter was 108 inches. The anti-stall hump was added using Belzona. The impeller vane geometry at the exit was also modified. The goal was to prevent impeller eye cavitation and increase flow that would aid the main turbine in increasing megawatts and reducing NPSH requirements. The cavitation was completely eliminated and 12% flow was gained, which was very much welcomed during the hot summer months.



Extension of condenser (cooling tower) circulating pump capacity by modification of both impeller inlet and exit. All modifications were performed by manual filing. Examples are: Allegh-Harrison (Gained 14%), Alabama-Farley Nuclear (Gained 12.5%). Total of 203 pumps modified between 1978 and 1993 with ERCO supervision.

Figure 4-13
Extension of Condenser Circulating Pump Capacity

Modifications to the horizontal, double-suction cooling tower circulating water pumps at the Harrison Fossil Station (Allegheny Power Company) were performed in April 1983. The impeller material was bronze and the impeller OD equaled 48 inches. The OEM had the first chance to make corrections. The hydraulic designer used their proprietary computer program to calculate new vane inlet angles. Modifications were made in their repair shop. Cavitation damage became considerably worse. The existing impeller was subsequently modified with an anti-stall hump and the impeller vane geometry at the exit was again modified to increase flow. The result was that the cavitation damage was completely eliminated and a 14% gain in flow was measured.

Evaluation of Cavitation Damage

Cavitation damage is caused by the production of very low local pressures adjacent to fluid flow boundaries. This causes vapor-filled pockets to form and then collapse violently when they are transported into higher-pressure regions. Damage is most likely to occur in the inlet of the pump impeller if caused by lack of NPSH but can be carried through the impeller, causing erosion of the impeller exit or the diffuser/volute inlet. Possible failure locations can be:

- Impeller eye (inlet) at various places, as described in Table 4-1
- Impeller exit
- Diffuser or volute inlet, as illustrated with an example in Figure 4-14

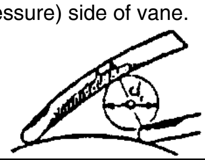
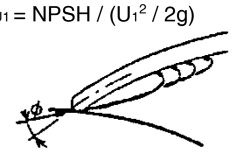
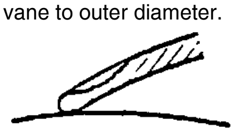
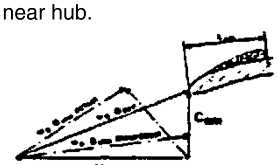
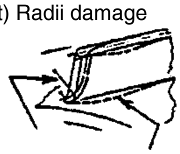
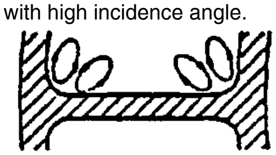
Six locations of cavitation damage at the impeller eye are shown in Table 4-1, which is organized to show (left to right) the following:

- Type of cavitation damage
- Flow mechanisms that induce damage
- Possible root causes
- Possible solutions for eliminating damage

Some of the most common cavitation damage can be caused by:

- Insufficient available NPSH or the fact that the impeller is not performing in accordance with its design. The damage nearly always shows on the impeller inlet and on its downstream portion.
- Flow recirculation in the impeller eye. This most likely occurs if the pump is operated at off-design flow conditions. The damage is always on the impeller eye.
- Incorrect impeller vane inlet angles at design flow or operating at off-design flows for extended periods of time. These two are the same because they both involve mismatched flow angles. The damage always shows on the vane with mismatched angles. Cavitation, if strong enough, can get carried downstream and damage additional downstream surfaces.
- Incorrect diffuser or volute vane inlet angle, or operating at off-design flows. Damage appearance will be the same as described for the impeller.
- Incorrect hydraulic channel geometry (impeller or diffuser) allowing flow separation (curved diffuser or guide vane channel). Most damage occurs on that channel; some might occur downstream.
- High localized velocities caused by sharp corners and other flow disturbances, such as a misplaced inlet guide vane. Damage always starts from the point of origin of high velocities and carries downstream.
- Vortex formation due to obstacles in the flow path, sharp elbows in the suction piping, incorrect pump inlet geometry, and blunt trailing edges of guide vanes. The damage is special in its appearance and occurs where the vortex touches a solid surface.

Table 4-1
Cavitation Damage Mechanisms and Remedies at the Impeller Eye

	Cavitation Damage Mechanisms and Remedies: Impeller Eye			
	Type of Cavitation/Damage	**Flow Mechanisms That Induce Damage**	**Possible Root Causes**	**Possible Solutions**
1.	Visible (suction) side of vane starting close to leading edge of vane. 	Sheet cavitation on suction of vane at Q < BEP • Damage hub to shroud • Damage near shroud • Damage near hub 	• High incidence angle • Improper leading edge profile • Incorrect vane design • Shroud inlet angle (β) too large • Hub inlet angle (β) too large	• Introduce anti-stall hump (least expensive) • Increase flow rate (if practical) • Reduce vane inlet angles, improve inlet edge profile • Introduce eye ring • Reduce eye diameter • Increase pre-rotation
2.	Visible (suction) side of vane, within channel. Erosion can also be observed on the shroud and/or non-visible (pressure) side of vane. 	Vortex cavitation on suction side of vane at low flow and low σ_{U1} σ_{U1} = NPSH / (U_1^2 / 2g) 	• High incidence angle and low σ_{U1} ($\leq$ 0.3) σ_{U2}= NPSH / (U_1^2 / 2g) NPSH based on 3% head breakdown	• Introduce anti-stall hump • Reduce vane inlet angle • Increase $NPSH_A$ • Reduce eye diameter • Increase cavitation resistance of impeller material (means new casting needed, expensive)
3.	Non-visible (pressure) side of vane; any location, forming close to leading edge of vane. 	Sheet cavitation due to flows above BEP (high incidence angle). 	• Incorrect flow angle at operating point • Incorrect vane design • Improper vane profile • Q > BEP Standard design feature at De Laval and at ERCO (has the least cavitation damage among all BFP applications). Used by ERCO to troubleshoot other OEM's designs.	• Reduce flow rate (if practical) • Introduce anti-stall-hump • Modify vane geometry/profile
4.	Non-visible (pressure) side of vane, damage occurs middle of vane to outer diameter. 	Part-load recirculation resulting in cavitation bubble impingement on the vane. 	• Impeller eye too large • Impeller throat area too large • Incorrect flow inlet angle	• Introduce anti-stall hump • Increase flow rate (if practical) • Insert inlet ring to reduce eye and throat area • Modify vane geometry/profile
5.	Non-visible (pressure) side of vane; damage near hub. Difficult to distinguish from 3.unless known if pump operated below or above BEP.	Part-load recirculation creating high incidence angle near hub. 	• Incorrect flow angle near hub causing recirculation at part-load	• Introduce anti-stall hump • Increase flow rate (if practical) • Modify vane geometry/profile (if possible) • Reduce pre-rotation at the impeller eye (if possible)
6.	Hub, or shroud, or corner (Fillet) Radii damage  The hook and the anti-stall hump.	Vortex cavitation at the corner; might be combined with high incidence angle. 	• Increase flow angle • Fillet radii too large • Operating at low flow (if on visible side)	• Try anti-stall hump • Modify fillet radii • Add proper hook • Modify vane profile as in 3. • Impeller redesign (modify vane angles - most expensive, least practical)

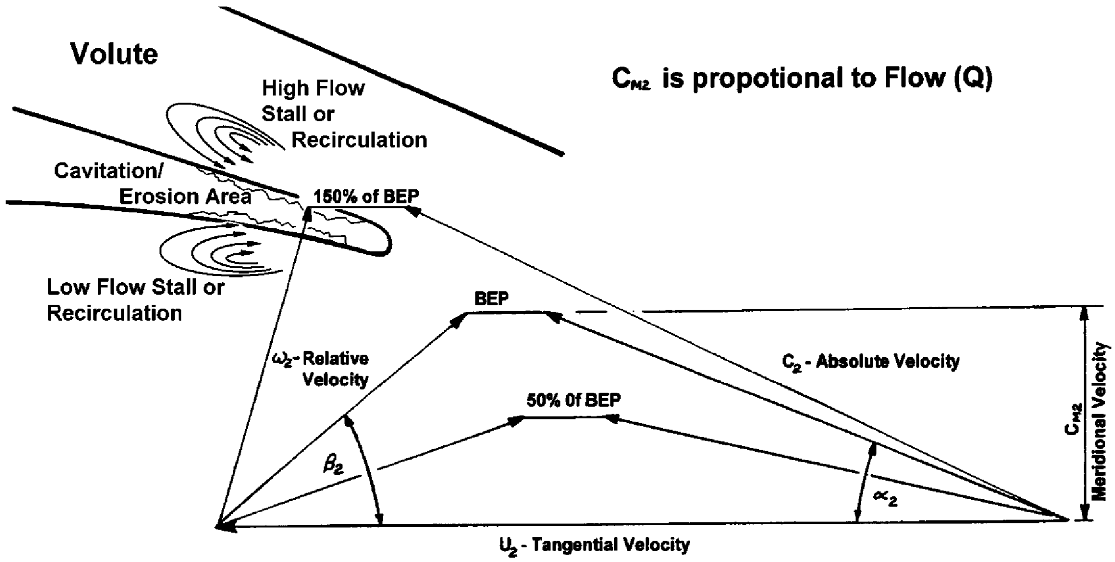


Figure 4-14
Typical Volute and Diffuser Cavitation Damage Caused by Off-BEP Operation

NPSH-required, as quoted by all OEMs, has been misunderstood by many utilities and A/Es for decades. NPSH-required, as defined by the OEM, refers to the NPSH at which a 3% head breakdown occurs at a given or constant flow. For a 3% head breakdown to occur, the pump must be in a fully developed state of cavitation. However, operation at this NPSH will not have immediate or devastating effects on the impeller or on pump operation. The pump performance will be reduced and, depending on the impeller material, inlet vane damage could be experienced after several months or years of operation at the minimum NPSH-required. As shown in Figure 4-15, the NPSH-available should be at least 1.5 times greater than the minimum NPSH-required value at a given flow point to reduce the possibility of impeller damage due to NPSH conditions.

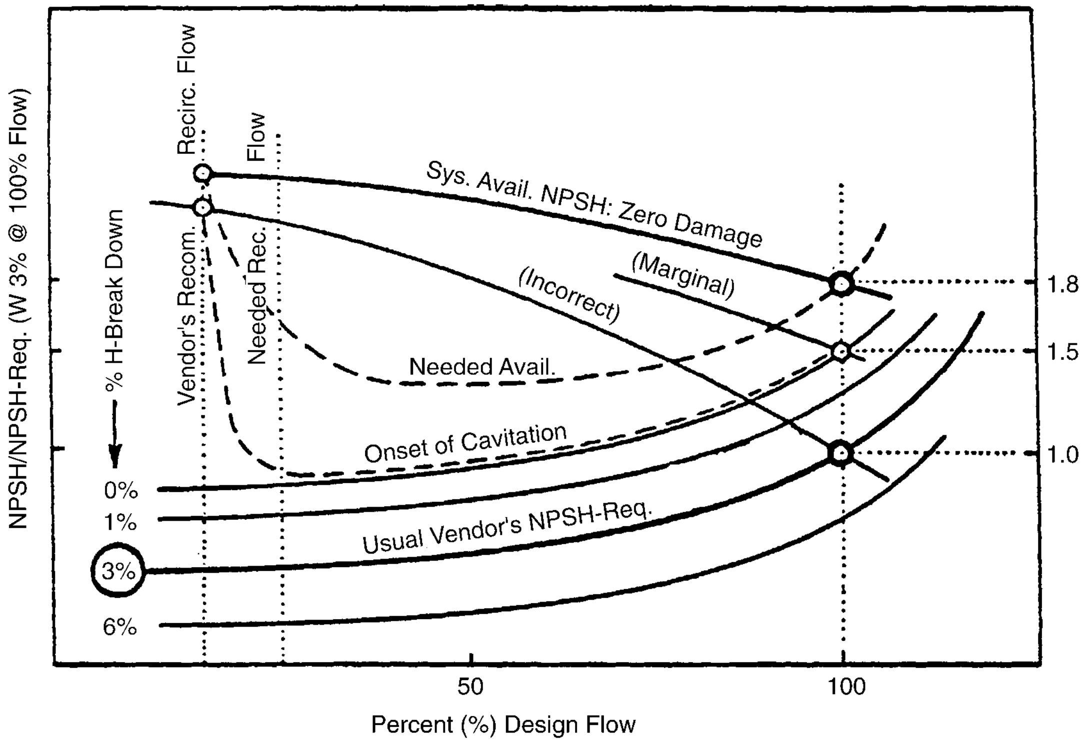


Figure 4-15
System-Available NPSH Versus NPSH-Required with 3% Head Breakdown

4.5.4 Inspection of Impeller Critical Hydraulic Geometry

Replacement pump impellers should be inspected to verify that hydraulic performance will equal that of the original supplied impeller. During manufacturing, impellers may be modified by grinding and polishing of vanes that affect performance. A drawing provided in Appendix J shows detailed measurements that should be taken of the original and the replacement impeller to ensure equal hydraulic performance.

4.6 Vertical Pump Lift

4.6.1 Overview

Vertical pump *lift* refers to the running position of the pump rotor [impeller(s) and shaft(s)]. Lift typically is referenced relative to the bottom of the pump, or when the pump is sitting down unsupported by the motor. The pump, when coupled to the motor, will be raised a vertical distance or lift into its running position. In all vertical pump applications, the lift must be adequate to account for shaft stretch due to pump thrust loads and thermal expansion differences between the pump shafts and the pump columns. The lift must be sufficient to prevent the pump impeller from contacting the pump bowls/casings once at load and at operating temperatures. For pumps subject to environmental changes such as ambient and water temperature changes, the lift must account for the extremes in environmental conditions.

Key Technical Point

In all vertical pump applications, the lift must be adequate to account for shaft stretch due to pump thrust loads and thermal expansion differences between the pump shafts and the pump columns.

In pumps with semi-open impellers, the lift establishes the distance between the open vane side of the impeller and the liner or shroud. The distance between the impeller vane and the liner/shroud directly impacts the head-flow output and efficiency of the pump. Test data, at two different lift settings, for a two-stage service water pump (Byron Jackson 32 RXL-28 KXL) have been plotted in Figure 4-16. The figure shows that, for a lift setting of 0.033 inch, the BEP of the pump was 9050 gpm, with 211 feet of head, 614 brake horsepower, and 79% efficiency. While at a lift setting of 0.077 inch, the pump BEP was again 9050 gpm, but the head was 197 feet, the brake horsepower was 594, and efficiency was 76%. Probably more important for the same head of 211 feet, the flow rate difference was 713 gpm (9050 gpm versus 8333 gpm) or 7.8%.

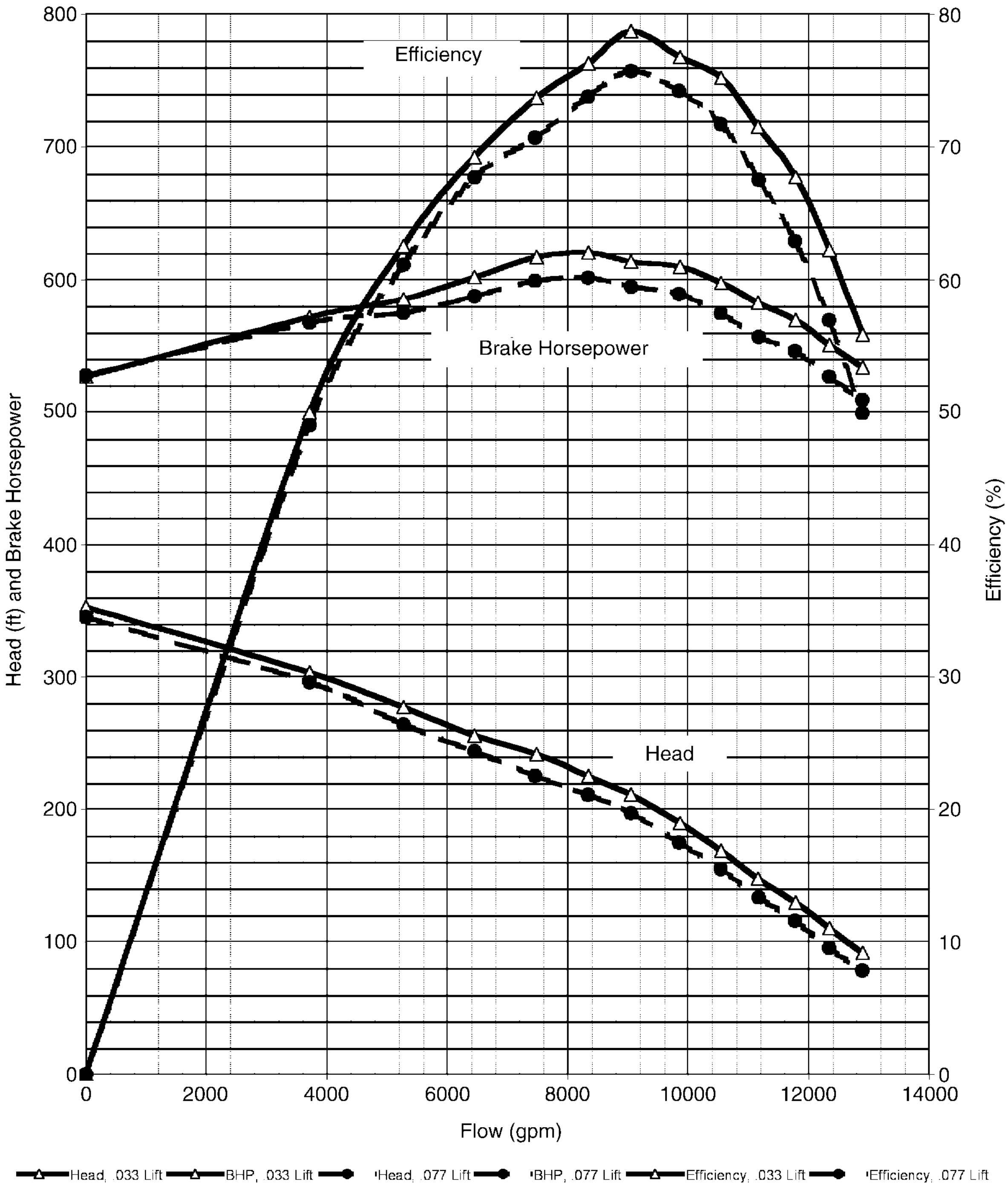


Figure 4-16
Effect of Pump Lift on Semi-Open Impeller Pump Performance

For pumps with semi-open impellers in abrasive environments (river water, holding ponds, circulating water), many times it is the gap between the impeller vane and the liner/shroud that increases and results in degraded pump performance. Restoring its original lift setting can usually return a pump's performance. For a safety-related pump, it would be appropriate to establish a new pump hydraulic baseline after the pump lift has been reset. However, it is not recommended that the vibration baseline be adjusted when a lift setting change has been made. The need to reset the pump lift has been the abrasive wear between the rotating and stationary components. In most cases, the edges of the impeller vanes experience most of the wear. Wear of the impeller can cause unbalance and uneven impeller loading, affecting pump vibration. Resetting the lift does not change these parameters and, as such, the vibration baseline should not be adjusted with lift adjustments.

The lift of pumps with closed impellers will not be affected in abrasive environments. Pump performance degradation would be the result of increased impeller wear ring clearances. A change to the pump lift will not affect the pump wear ring clearances. Only after removing the pump, can the wear ring clearances be restored.

The performance of a pump with semi-open impellers can also be affected by temperature changes of the water pumped. A paper by Randall Noon from the Cooper Nuclear station, provided in Section 4.6.2, details the effects of temperature changes on pump lift and performance.

It is important during pump repairs that the contour between the impeller vanes and the liner/shroud be evaluated and repaired as required. An effective way to inspect the gap is with long (12 inches) feeler gages (see Figures 4-17 and 4-18). It is recommended that the gap between the impeller and the liner/shroud be 0.010 inch or less. Many times the liner/shroud will be in an acceptable condition, so the edges of the impeller vanes are skim cut to match the liner/shroud contour. When repairing multi-stage vertical pumps with semi-open impellers, the gap between the impeller and the liner/shroud must be checked and corrected as each stage is assembled.

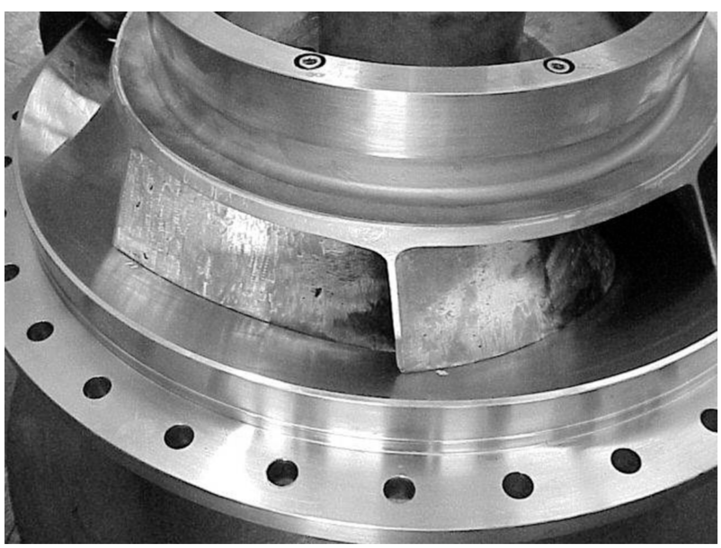


Figure 4-17
Impeller Set in Liner/Shroud

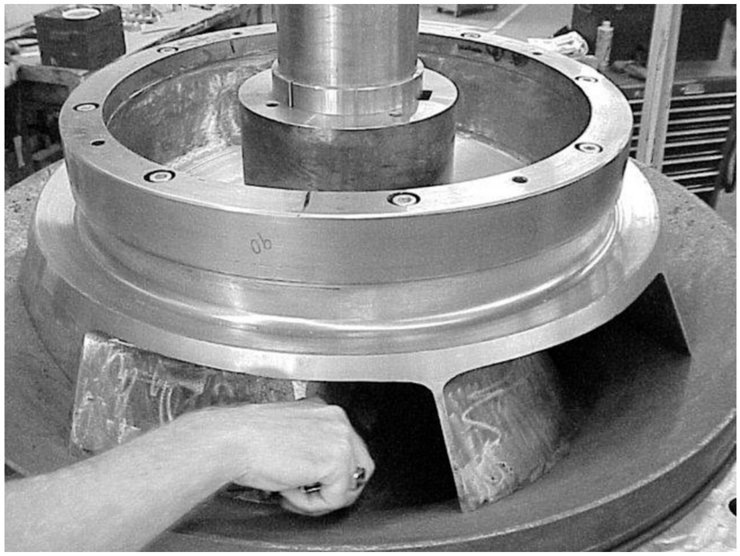


Figure 4-18
Using Long Feeler Gage to Inspect the Impeller and Liner/Shroud Gap

4.6.2 Pump Lift Case Study

The following is a paper written by Randall Noon of the Cooper Nuclear Station. The paper describes and demonstrates the performance impact of temperature on pumps with semi-open impellers. The performance impact results from the significant change in pump lift due to seasonal changes in water temperatures.

4.6.2.1 Introduction

The following paper explains what pump lift is, how it is accomplished, and the theoretical underpinnings of why it is needed. An example is provided about how a pump lift adjustment on a service water vertical pump at Cooper Nuclear Station significantly improved its performance. The paper then discusses how increasing and regularly scheduling the number of pump lifts in the service water pumps at Cooper Nuclear Station can significantly increase the life of the pumps, reduce maintenance, and reduce operating costs.

4.6.2.2 Pump Lift

Pump lift is an adjustment procedure generally associated with vertical, mixed-flow-type pumps. These pumps also go by the descriptors *vertical turbine pumps*, *irrigation pumps*, *barrel pumps*, *fire pumps*, or *propeller pumps*. In general, this type of pump takes water from a reservoir and pumps it vertically through a riser, called a *flow tube*, to a higher elevation. The flow tube also contains the shaft that connects the motor (which is located on top) to the pump impeller, which is immersed in the reservoir. Water from the reservoir enters the impeller axially at the bottom of the pump and is discharged both axially and radially into a volute-type casing located just above the impeller.

Figure 4-19 shows a simplified schematic of a service water vertical pump at Cooper Nuclear Station.

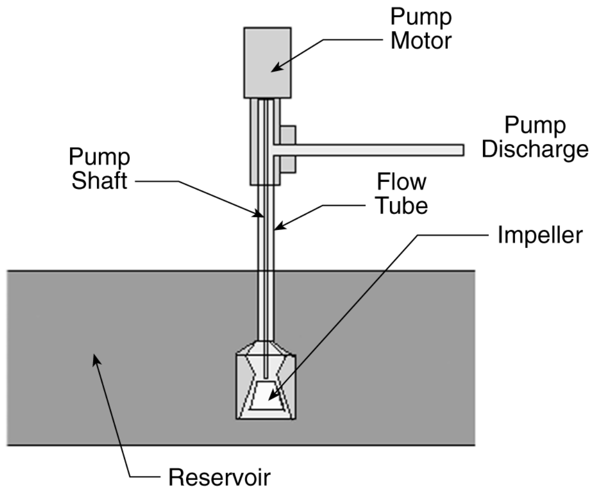


Figure 4-19
Schematic of Vertical Pump

Mixed flow, vertical pumps typically are for medium-head applications where the specific speed of the pump ranges from 4000 to 9000 gpm. At Cooper Nuclear Station, there are several systems that contain vertical, mixed-flow pumps. The two systems that contain relatively large vertical pumps of this type are the Service Water System and the Circulating Water System. The service water system has four vertical pumps, each rated at 8000 gpm, and the circulating water system has four vertical pumps rated at 159,000 gpm each.

In a vertical pump, the pump impeller sits in a casing or bowl. The outer diameter contours of the impeller vanes match (that is, they are supposed to match) those of the bowl so that the tips of the impeller vanes are always parallel to the surface of the bowl. The parallel gap between the impeller vanes and the bowl (the clearance) significantly influences the efficiency of the pump.

If the clearance is too large, water can re-circulate from the high-pressure portion downstream of the impeller (above the impeller) to the low-pressure portion upstream of the impeller (below the impeller). This not only causes a loss in efficiency of the pump, but it can also lead to accelerated erosion of the bowl.

On the other hand, if the clearance is too close, the surface hydraulic boundary layers of the impeller and bowl can interfere with each other. This causes the hydraulic friction (due to viscous shear) between the two boundary layers to increase, which decreases pump efficiency.

Further, if the clearance is much too close, the impeller and bowl might directly interfere and scrape on each other. This causes a significant decrease in pump efficiency. Energy intended for pumping water is diverted and consumed by the impeller grinding itself into the pump bowl. This contact causes permanent damage to both the impeller and bowl, and shortens the service life of the pump.

Pump impellers and pump bowls are never perfectly round. A pump bowl about 30 inches in diameter can have a diametric tolerance of perhaps ±5 mils (1 mil = 0.001 inch). Likewise, the outside diameter of the pump impeller that matches the bowl has a similar tolerance. If the clearance between the pump bowl and the impeller is too tight, one or more of the impeller vanes will impinge on a common high point or asperity between the bowl and the impeller. When this occurs, the effect is detectable by:

- A sudden increase in amperage, a decrease in pump output pressure, or both
- An increase in pump vibrations, which have a frequency of the shaft speed times the number of vanes on the impeller contacting the bowl. (Note: when there are two symmetric high spots in the bowl, as would occur if it were elliptical, the frequency might be two times the shaft speed times the number of vanes making contact.)

Between the two extremes of too tight and too loose, there is a *just right* clearance dimension. This just right clearance dimension allows the boundary layers of the pump impeller and the bowl to slide over each other with minimal shear, but it is not so large as to allow excessive re-circulation between the upstream and downstream sides of the impeller. At the just right clearance, pump efficiency will be at its maximum.

Impeller clearances are usually specified by the manufacturer. Table 4-2 provides the magnitude of typical impeller clearances.

Table 4-2
Mixed Flow Impeller Clearances

Bowl Diameter	Impeller Clearance
6 to 12 inches	15 to 20 mils
13 to 24 inches	20 to 30 mils
25 to 36 inches	30 to 50 mils
37 to 48 inches	50 to 80 mils

At Cooper Nuclear Station, the current Service Water Pump Lift Adjustment Procedure, CNS Maintenance Procedure 7.2.45, indicates that the lift for the 28 inch service water pumps should be between 40 to 60 mils when the lift adjustment procedure has been completed. The manufacturer of the service water pumps, the Byron Jackson Pump Company, recommends a lift adjustment of 56 mils.

A pump lift is accomplished by actually lifting the impeller upwards, so that the measured vertical gap between the impeller and bowl is within the range of acceptable values provided by the manufacturer or by the engineer in charge. Because the pump impeller and bowl themselves are normally immersed in water and are inaccessible, this is done at the top of the pump by loosening the pump shaft from the motor shaft at their coupling, and then allowing the impeller and shaft to rest on the bowl. This is the *zero-clearance* position. The shaft is then lifted upwards from the zero clearance position, usually by tightening coupling bolts or adjustment plates. The amount of lift is the amount of upward displacement of the shaft and impeller that is created by tightening the coupling bolts or adjustment plates from the zero position. The clearance measurement is usually made with feeler gages or dial indicators.

4.6.2.3 Theoretical Underpinnings

The total developed head (TDH) of an individual pump is given in Equation 4-1. Total developed head is fundamentally a measure of the total output energy of a pump. The pressure term and elevation term are the potential energy components and the velocity term is the kinetic energy component. By adding all three terms, the total energy is obtained.

$TDH = P/r + V^2/(2g_c) + Z$ **Eq. 4-1**

Where
P = pressure
r = density of water
Z = change in elevation
V = flow velocity
g_c = gravitational constant to units conversion

The relationship between volumetric flow and flow velocity for water in a pipe is given by the basic continuity equation for incompressible fluid flow (Equation 4-2).

$Q = VA$ or $Q/A = V$ **Eq. 4-2**

Where
Q = volumetric flow (that is, volume of flow per unit of time)
V = flow velocity
A = cross-sectional area of the pipe

Substituting Equation 4-2 into Equation 4-1 allows us to directly relate TDH to volumetric flow in the following way.

$TDH = P/r + Q^2/(A^2)(2g_c) + Z$ **Eq. 4-3**

If the TDH remains constant within a particular operating range, and the elevation (the *Z* term) does not change, Equation 4-3 indicates that, when pressure goes down, the volumetric flow must increase. Likewise, when the volumetric flow decreases, the pressure must increase. By taking the differential of Equation 4-3, the exact relationship between pressure and volumetric flow when TDH is a constant can be determined.

$TDH = P_1/r + Q^2/(A^2)(2g_c) + Z$
$0 = [1/r][dP/dQ] + [1/(A^2)(g_c)][Q]$
$dP/dQ = - [r/(A^2)(g_c)][Q] = - [K]Q$

Where K = a constant = $[r/(A^2)(g_c)]$

To relate pump motor energy input to pump output, the following relationship holds.

$E – L = TDH_1 = [P/r + Q^2/(A^2)(2g_c) + Z][dm/dt]$ **Eq. 4-4**

Where E = electrical energy input to the pump motor
L = losses due to friction and heat in both the pump and motor
dm/dt = mass flow rate of water through the pump

Of course, Equation 4-4 is basically a restatement of the Conservation of Energy Law as it applies to pumps driven by electric motors.

Assuming that the normal losses from heat and friction in both the pump and the motor are simply a constant percentage of the electrical energy input for a particular operating range, then Equation 4-4 can be re-written as follows:

$E – L = hE = [TDH][dm/dt] = [P/r + Q^2/(A^2)(2g_c) + Z][dm/dt]$ **Eq. 4-5**

Where h = overall mechanical efficiency of the motor and pump combination

The electrical energy input is composed of several parameters. Typically, when a three-phase motor is used to drive the pump, those parameters are as follows:

$$E = (I_{rms})(V_{rms})(1.732)(PF) \qquad \textbf{Eq. 4-6}$$

Where I_{rms} = measured root mean squared current value
V_{rms} = measured root mean squared voltage value
1.732 = factor to account for three-phase power
PF = power factor of motor

Substituting Equation 4-6 into Equation 4-5 gives the following relationship that relates pump output pressure, flow output, and electrical consumption of the pump motor.

$$hE = h(I_{rms})(V_{rms})(1.732)(PF) = [TDH][dm/dt] = [P_1/r + Q^2/(A^2)(2g_c) + Z][dm/dt] \qquad \textbf{Eq. 4-7}$$

When the impeller clearance of a vertical pump is out of its recommended range, it creates energy losses in addition to the normal losses captured in the *h* term. This additional loss of energy is accounted for by the addition of an extra term, $L_{clearance}$, to the left side of Equation 4-7:

$$h(I_{rms})(V_{rms})(1.732)(PF) - L_{clearance} = [P_1/r + Q^2/(A^2)(2g_c) + Z][dm/dt] \qquad \textbf{Eq. 4-8}$$

Where $L_{clearance}$ = energy losses associated with an out-of-range clearance

At Cooper Nuclear Station, the vertical pump motors are the synchronous type. They maintain nearly constant RPMs and will, within certain limits, maintain constant flow output unless throttling is done.

As indicated by inspection of Equation 4-8, if the clearance losses in a vertical pump are significant at a given total developed head, these losses will cause a corresponding loss in the output pressure of the pump. Likewise, if a lift adjustment is made and the clearance losses are minimized (that is, $L_{clearance} \sim 0$), an increase in output pressure will be realized.

Thus, anytime the lift clearance on a particular pump is too tight or too loose, output pressure improvements can be made by resetting the lift so that it is within the recommended range.

4.6.2.4 Things That Cause Clearance Values to Change

Wear

Normal wear on the impeller vanes and the interior surface of the pump bowl are typically the result of the abrasive action of moving water, especially river water containing sand and other suspended solid particles. In general, this type of wear tends to cause pump clearances to increase over time, which consequently causes pump efficiency to decrease over time.

At Cooper Nuclear Station, typically the intake water contains 0.1 to 0.3% solid particles by volume. Thus, for every 120 gallons of water pumped, there will also be 1.28 to 3.8 pounds of abrasive sand and sediment pumped along with it. Clearance losses of this type usually occur gradually over time.

As already discussed, wear on the impeller and bowl can also be caused by the clearance between the impeller and bowl being too tight. This type of accelerated wear occurs rapidly and will cause a corresponding loss of pump performance to occur in a relatively short period of time.

Relative Expansion and Contraction

If the impeller, impeller shaft, bowl, and flow tube are made of different materials, and the pump assembly and reservoir are located outside in an uncontrolled environment where there are significant temperature changes, like in a river, it is likely that the assembly will experience differential expansion and contraction. A lift adjustment made when the river temperature is hot might not be within tolerance when the river temperature is cold, and vice-versa (see Figure 4-20).

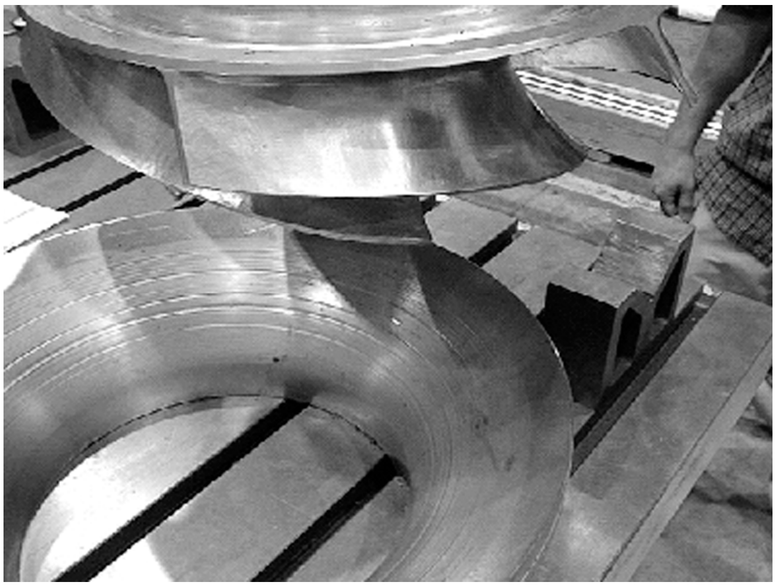


Figure 4-20
Gouging in Impeller Bowl due to Expansion and Contraction Effects of Impeller Shaft

For example, the service water pumps at Cooper Nuclear Station have an overall length of about 46 feet from motor coupling to the bottom of the impeller. The pump shafts are made of hardened 410 stainless steel, which has an expansion coefficient of about 5.5 x 10^{-6} in per in per °F. On the other hand, the AISI 1030 cast steel flow tube has an average expansion coefficient of about 8.3 x 10^{-6} in per in per °F. During the winter, the minimum intake water temperature to the service water system is about 35°F. During the summer, the maximum temperature is about 90°F. Thus, the greatest relative expansion or contraction between the service water pump shaft and the flow tube, due to the extremes of weather, is estimated as follows:

$dL = (a)*(dT)*L$ **Eq. 4-9**

$dL_{tube} = (46 \times 12 \text{ in})(8.2 \times 10^{-6} \text{ in per in per °F})(55°F) = 0.249 \text{ in.}$

$dL_{shaft} = (46 \times 12 \text{ in})(5.5 \times 10^{-6} \text{ in per in per °F})(55°F) = 0.167 \text{ in.}$

Difference = 0.082 in. or 82 mils

Thus, if the pump is set to a lift position of 40 mils during the summer on the hottest river day and no further lift adjustments are made, the pump impeller will grind itself into the pump bowl to a depth of 42 mils by the time the coldest river day occurs. This will, of course, cause accelerated wear in the both the impeller and bowl, and significantly shorten the service life of the pump.

Similarly, if the pump is set to a lift position of 40 mils on the coldest river day and no other lift adjustments are made, the clearance will open to 122 mils by the time the hottest river day occurs. This will cause the output pressure and the total developed head to slowly drop over a six-month period.

4.6.2.5 Example of Performance Improvement by a Simple Lift Adjustment

The following is an example of how a lift adjustment has improved the performance of a service water pump at CNS.

The “C” service water pump is a single-stage, vertical mixed-flow pump manufactured by the Byron Jackson Pump Company. The pump is a model 28 KXL VC, rated at 8000 gpm. Its driver is a three-phase, synchronous motor rated at 300 horsepower, 1180 rpm, and 4160 V with a power factor of 87%. The nominal impeller diameter is 28 inches.

On July 13, 2000, a surveillance of service water pump “C” was made. When such a surveillance is done, the pump flow is throttled to 5500 gpm and the pump is run by itself within the division, as opposed to being run in tandem with another similarly sized pump in the same division.

In that surveillance, it was reported that the pump was performing poorly. The total developed head was found to be 133.6 ft., and the differential pressure (TDH converted to psig) of the pump was found to be 57.7 psig. Because the lowest acceptable differential pressure for this pump is 57.1 psig, and the lowest acceptable total developed head is 130 ft., service water pump “C” was in jeopardy of being declared inoperable, especially because a scheduled overhaul was a year away.

Records indicated that the pump had last had a lift adjustment on April 5, 2000, when the water temperature was about 52°F. The river temperature was about 75°F on July 13, 2000. The differential expansion due to this change in temperature is about 34 mils. This amount alone would have increased the pump lift to about 85 mils, which is not acceptable for efficient performance. Impeller and bowl wear, and any settlement would have added even more. Thus, it was expected that the impeller clearance had opened too much by July 13, 2000, and that this was the cause of the poor performance.

Table 4-3 lists the performance data of the "C" pump, as it was measured on July 13, 2000.

Table 4-3
Performance Data for Pump "C" on 7/13/00

PF = 0.87 (assumed)
V_{rms} = 4160 v
I_{rms} = 34.9 a
E = (4160 v)(34.9 a)(1.732)(0.87) = 219,000 watts = 161,500 ft lbf/sec
Q = 5500 gpm = 735.24 ft^3/min = 12.25 ft^3/sec
A = 299 sq in = 2.07 ft^2 (20 in diameter pipe)
Z = 25 ft (correction for river elevation)
P = 47 psig = 6768 lbf/ft^2 (measured output pressure)
r = 62.27 lbf/ft^3 (at 75°F)
dm/dt = (763.0 lbf/sec)
Plugging the above data into Equation 4-8 gives the following.
$h(I_{rms})(V_{rms})(1.732)(PF) - L_{clearance} = [P_1/r + Q^2/(A^2)(2g_c) + Z][dm/dt]$
h(161,500 ft lbf/sec) - $L_{clearance}$ = [(6768 lbf/ft^2) /(62.27 lbf/ft^3) + (12.25 ft^3/sec)^2/(2.07 ft^2)^2(2)(32.17 ft/sec^2) + 25 ft][763.0 lbf/sec]
h(161,500 ft lbf/sec) - $L_{clearance}$ = [108.7 ft + 0.54 ft + 25 ft][763.0 lbf/sec]
h(211.7 ft) – [$L_{clearance}$]/[763.0 lbf/sec] = 134.2 ft

Because previous experience with this pump indicated that the overall efficiency of the motor and pump was 64.85%, then from the data gathered on July 13, 2000, the losses associated with the clearance in this case were about 3.1 ft. of head. This accounts for a loss of output pressure of about 1.33 psig.

From the above data and analysis, and assuming that the same operating conditions would continue to exist, an increase of 1.33 psig in the differential pressure could be obtained if the lift were adjusted. If the 1.33 psig amount was added to the 57.7 psig figure, which was obtained in the surveillance, it would give a DP of 59.0 psig. This would take the pump out of jeopardy.

On July 22, 2000, a lift adjustment on the same pump was made. Prior to the adjustment, the existing clearance was found to be 128 mils, which is 68 mils larger than the upper limit for lift settings noted in CNS Procedure 7.2.45. After the lift adjustment was made, the as-left clearance was measured at 51 mils, which is very close to the manufacturer's recommended value, and well within the range of values listed in CNS Procedure 7.2.45. Table 4-4 lists the performance parameters of the pump after the lift adjustment was made and another surveillance was run.

Table 4-4
Performance Data for Pump "C" on 7/22/00

PF = 0.87 (given)
V_{rms} = 4190 v
I_{rms} = 36.5 a
E = (4190 v)(36.5 a)(1.732)(0.87) = 230,000 watts = 169,640 ft lbf/sec
Q = 5500 gpm = 735.24 ft^3/min = 12.25 ft^3/sec
A = 299 sq in = 2.07 ft^2
Z = 25.83 ft
P = 51 psig = 7344 lbf/ft^2
r = 62.22 lbf/ft^3 (at 80°F)
dm/dt = 762.36 lbf/sec
Plugging the above data into Equation 4-8 gives the following.
$h(I_{rms})(V_{rms})(1.732)(PF) - L_{clearance} = [P_1/r + Q^2/(A^2)(2g_c) + Z][dm/dt]$
$h(169{,}640 \text{ ft lbf/sec}) - L_{clearance} = [(7344 \text{ lbf/ft}^2)/(62.22 \text{ lbf/ft}^3)$
$+ (12.25 \text{ ft}^3/\text{sec})^2/(2.07 \text{ ft}^2)^2(2)(32.17 \text{ ft/sec}^2) + 25.8 \text{ ft}][762.36 \text{ lbf/sec}]$
$h(169{,}640 \text{ ft lbf/sec}) - L_{clearance} = [118.0 \text{ ft} + 0.54 \text{ ft} + 25.8 \text{ ft}][762.36 \text{ lbf/sec}]$
$h(222.9 \text{ ft}) – [L_{clearance}]/[762.36 \text{ lbf/sec}] = 144.3 \text{ ft}$
Assuming that the overall pump and motor efficiency had not changed in the interim, then the clearance losses after the lift are estimated as follows.
$(0.6485)(222.9 \text{ ft}) – [L_{clearance}]/[762.36 \text{ lbf/sec}] = 144.3 \text{ ft}$
$144.56 - L_{clearance} = 144.3 \text{ ft}$
$L_{clearance}$ = 0.26 ft which is below the threshold of significant figures, and is therefore approximately equal to zero.

Thus, the lift adjustment done on July 22, 2000, removed the losses occurring in the pump due to excessive lift clearance. It also increased the total developed head of the pump from the previous value of 134.2 to 144.3 ft. This improvement placed the pump well above the 130 ft. minimum head requirement.

The improved TDH after the lift adjustment converts to a differential pressure of 62 psig. This is apparently much better than the 59 psig DP figure that was predicted from the pre-lift data. The apparent variance between the two figures is attributed to instrumentation error. The error in reading the pressure gage is ±1 psi. The error in reading the ammeter is ±1 ampere. The error in reading the voltmeter is ±10 volts. The error in reading the river level is about ±1 inch, and the error in reading the flow is about ±100 gpm.

Based upon these tolerances, the error in determining the electrical energy input is about 6%. Similarly, the error in determining the energy output of the pump is about 5%. Thus, the improved DP of 62 psig, as measured by the second surveillance, is within the error range associated with the predicted 59 psig DP figure.

4.6.2.6 Implications

Scheduling of Lift Adjustments for Service Water Pumps

Currently, the regular period between overhauls for the service water pumps at Cooper Nuclear Station is two years. The primary limiting condition that requires this relatively short turn-around period is wear on the impeller and pump bowl. The wear observed or measured on the other expendable pump components, such as the pump bearings, Cutlass bearings, the check valve, the pump shafts and shaft couplings, pump wear rings, etc., do not prevent the pumps from being safely run for longer periods of time between turn-arounds. In fact, an informal telephone survey of several nuclear plants, some with the same type and manufacturer of vertical pumps, found that the turn-around time associated with the service water pumps at the Cooper Nuclear Station was the shortest. The usual turn-around time was four years; some were longer.

In the past, lift adjustments on the service water pumps at Cooper Nuclear Station have only been done once or, perhaps, twice a year. As shown previously in this report, because of differential expansion, one or even two lift adjustments per year is not enough to ensure that the pump operates within the recommended clearance range.

More importantly, one or two lift adjustments per year are not enough to ensure that the impeller will not contact the bowl and grind itself into the casing, causing accelerated wear and degraded performance. As noted previously, about 42 mils of material might be ground off each year of operation if only one or two lift adjustments are made. After two years of operating this way, up to 84 mils of bowl and impeller material could be ground off. In short, the largest contributor to impeller-bowl wear in service water pumps has been the lack of timely lift adjustments.

Inspection of the bowl and impeller when they have been routinely removed for inspection and maintenance confirms that such grinding damage has occurred. Re-machining of the impeller and bowl due to excessive wear is typically required in a service water pump overhaul, and examination of the wear has found that it not only consists of rounded and smooth erosive cuts associated with water flow and sand abrasion, but also of hard, sharp striations that are parallel with the rotational plane of the impeller.

Thus, to ensure that the pump is operated within reasonable clearance dimensions, and to ensure that the impeller does not inadvertently grind itself into the bowl, it is necessary to perform **four** pump lifts per year on the service water pumps at CNS, one lift per quarter. Because performance surveillances (CNS Procedures 6.1SW.101 and 6.2SW.101) are required directly after a lift adjustment is made to re-establish the baseline characteristics of a pump, it is expedient to simply perform a lift adjustment and surveillance at the same time. For this reason, it is recommended that lift adjustments, followed by surveillance runs, be done on each pump on a quarterly basis during a calendar year.

Change in Service Water Pump Turn-Around Periods

Because much of the accelerated wear observed on the bowl and impeller has been caused by not adjusting the lift enough times in a year, when lifts are done according to a regular schedule, the time between overhauls for the service water pumps can be significantly increased.

As noted in various inspections, when a service water pump has been dismantled after two years of service, wear in the expendable parts has been minimal. In fact, when these parts have only been used for a one-to-two-year service period, they have been found to be within tolerance with sufficient margin to be reusable a second time as is. Thus, it is recommended that the turn-around period for service water pumps be changed from a two-year turn-around period to a four-year turn-around period.

Economic Assessment of Service Water Maintenance Costs

Prior to the summer of 1999, typical costs for a service water pump turn-around were approximately $198,000 per pump. On an annualized basis, this is $99,000 per year per pump. It was costing Cooper Nuclear Station approximately $396,000 per year, or $792,000 every two years, to maintain the four service water pumps.

This remarkably high cost for the maintenance of the service water vertical mixed-flow pumps was due to several factors. One factor is the fact that new intermediate pump shafts purchased from the OEM on a non-competitive basis cost $15,000 each, and each pump required four of them. This was an annualized cost per pump of $30,000, just for pump intermediate shafts.

Since the summer of 1999, however, the cost for intermediate shafts has been reduced to about $2000 per intermediate shaft. This was possible because the machine shop at Cooper Nuclear Station is now making replacement intermediate shafts for the service water pumps from heat-treated bar stock that is readily available from competitive suppliers. Further, because it has been found that a new shaft is not needed every two years; an intermediate shaft can be reused at least once, if not more. Thus, the annualized cost for these shafts per pump has dropped from $30,000 to less than $2,000.

The other major factor causing high maintenance costs has been the relatively short turn-around time for the service water pumps. This relatively short turn-around time was necessitated by the high amount of wear observed in the bowl and impeller. As noted previously, this wear was primarily caused by a lack of timely administered lift adjustments. Thus, it can be significantly reduced by performing lift adjustments on a quarterly basis. By doing so, it is estimated that the wear on the pump and bowl will be at least halved. Consequently, this allows the scheduled turn-around period to be increased from two years to four years.

Table 4-5 indicates the reduction in cost in present constant dollars of maintaining the service water pumps since early 1999. The first column indicates typical turn-around maintenance costs prior to summer 1999 when intermediate shaft replacement parts were bought from the OEM. The second column indicates the savings accrued due to making the intermediate shafts in-house. The last column indicates the total savings accrued if the turn-around period is lengthened from two to four years. As Table 4-5 shows, the savings are significant. Instead of spending nearly 4 million dollars for routine service water pump maintenance, only 1.42 million dollars will be spent.

Table 4-5
Typical Maintenance Costs Per Service Water Pump Per Year at CNS

Annualized Basis			
Item	**Pre-1999**	**Summer 1999**	**4-Year Pump Maintenance**
Labor to change out pump	5,000	5,000	2,500
3 workers – 3 days			
90 mhrs at $111/hr			
Intermediate shafts	30,000	2,000	2,000
Check valve	19,000	19,000	8,500
Other parts and miscellaneous	30,000	30,000	15,000
Machining bowl and impeller/weld buildup	15,000	15,000	7,500
Totals:	$99,000	$71,000	$35,500
Savings (%)	0%	28%	64%

4.6.2.7 Conclusion

For the past twenty years, Cooper Nuclear Station has replaced their pumps every two years due to excessive wear. Until this year, no one had looked into why CNS pumps had to be replaced so much more often than other plants. It was assumed that sand in the water was the reason for the wear. There was, after all, nothing unusual; the pumps have always worn that fast there. Cooper Nuclear Station spent millions of dollars replacing and repairing pumps twice as often as necessary, all because of thermal expansion. The thermal expansion caused the propeller to grind into the bowl, decreasing both the life of the pump and the pump's efficiency. This problem, once identified, was easily solved with regular lift adjustments to prevent the clearance from becoming too small. This just demonstrates that the application of even the simplest equation can produce a dramatic difference.

4.7 Rotor Balancing (In a Shop or Repair Facility)

4.7.1 Overview

Rotor (that is, shafts and impellers) unbalance represents one of the primary sources of vibration in vertical pumps. Because the weight of the rotor is in the axial direction of the shaft, there are no restrictions of radial motion induced by the centrifugal forces resulting from unbalance. The bearings tend to respond more as bumpers, than as the rotor dynamic supports of horizontal machines. The bearings offer damping to the vibration but not much stiffness. That is one reason why using pump and motor housing vibration only to monitor vertical pump vibration will not always provide a good picture of rotor condition. The vibration is so dampened by the system that a problem might not be detected until the pump is approaching end of life, particularly if the problem is originating from within the pump. Proximity probes should also be used to examine shaft motion, in addition to the pump or motor housing data.

Centrifugal force acts upon the entire mass of a rotating component, impelling each particle outward and away from the axis of rotation in a radial direction. If the mass of a rotating component is evenly distributed about its shaft axis, the part is balanced and rotates without vibration. However, if an excess of mass exists on one side of a rotor, the centrifugal force acting upon this heavy side exceeds the centrifugal force exerted by the light side and pulls the rotor in the direction of the heavy side. The excess of mass on one side of a rotor is called *unbalance* and can be caused by a variety of reasons, including the following:

- Tolerances in fabrication, including casting, machining, and assembly
- Variation within materials, such as voids, porosity, inclusions, grain, density, and finishes
- Non-symmetry of design, including motor windings, part shapes, location, and density of finishes
- Non-symmetry in use, including distortion, dimensional changes, and shifting of parts due to rotational stresses, aerodynamic forces, and temperature changes

It can be seen from this list of causes of unbalance, that careful attention to detail during manufacturing and repair is essential to reducing the amount of unbalance in a rotor.

An unbalanced rotor will cause vibration and stress in the rotor itself and in its supporting structure. Balancing of the rotor is therefore necessary to accomplish one or more of the following:

- Increase product quality
- Minimize vibration
- Minimize audible and signal noises
- Minimize structural stresses
- Minimize operator annoyance and fatigue
- Increase bearing life
- Minimize power loss
- Improve overall pump reliability

Unbalance, in just one rotating component of an assembly, can cause the entire assembly to vibrate. This induced vibration, in turn, can cause excessive wear in bearings, bushings, shafts, couplings, etc., and substantially reduce their service life. Vibrations are very highly undesirable alternating stresses in structural supports and housings, which may eventually lead to their complete failure. Performance is decreased because of the absorption of energy by the supporting structure. Vibrations can also be transmitted through the floor to adjacent machinery and seriously impair its accuracy or proper functioning.

In the shop, a balancing machine is used to detect, locate, and measure unbalance. The data furnished by the machine permit changing the mass distribution of a rotor, which, when done accurately, will balance the rotor. Balance is a zero quantity and is, therefore, detected by observing an absence of unbalance. The balancing machine measures only unbalance, never balance.

Key Technical Point

The balancing machine measures only unbalance, never balance.

4.7.2 Balancing Machines

The balance machines most commonly found in the industry are either soft bearing or hard bearing machines. The most common soft bearing balancing machines are typically manufactured by IRD, while Schenck-Trebel machines are the most common hard bearing machines found in the pump industry. In general, the hard bearing balancing machines have a reputation for being able to achieve unbalance levels that are lower than the soft bearing machines.

The soft bearing balancing machine derives its name from the fact that it supports the rotor to be balanced on bearings that are very flexibly suspended. This permits the rotor to vibrate freely in at least one direction, usually horizontal and perpendicular to the rotor shaft mass. Hard bearing balancing machines are essentially of the same construction as soft bearing balancing machines, except that their bearing supports are significantly stiffer in the transverse horizontal direction. Neither bearing mass, rotor mass, nor inertia influences the output from a hard bearing machine, so that a permanent relation between unbalance and sensing element output can be established. Centrifugal force from a given unbalance rises with the square of the balancing speed. Output from the pickups rise proportionally with the second or third power of the speed, depending on the type of pickup used. Suitable integrator circuitry then reduces the pickup signal inversely proportional to the square (or respectively, cube) of the balancing speed increase, resulting in a constant unbalance readout. Unlike soft bearing machines, the use of calibration masses or shakers is not required to calibrate the machine for a given rotor.

Key Technical Point

Unlike soft bearing machines, the use of calibration masses or shakers is not required to calibrate a hard bearing machine for a given rotor.

4.7.3 Shop Balancing

The most important aspect to remember when witnessing or reviewing balance results and procedures is that an unbalance value without a distance has no value or meaning. Balance standards and tolerances are based on an unbalance mass at a standard distance from the rotating centerline of 1 inch, for example ounce-inch and gram inch. An unbalance of 10 ounces at a radius of 5 inches has an unbalance of 10/5 or 2 ounce-inches, while at a radius of 10 inches the unbalance would be 1 ounce-inch or 1/2 the value at 5 inches. This concept is important because typically the balance machine is set to display an unbalance weight for the given radius at which the unbalance weight will be removed or added.

Key Technical Point

An unbalance value without a distance has no value or meaning.

The balance machine operator must first set the rotor geometry in the balance machine hardware prior to performing any balancing. The typical rotor dimensions required for a dynamic balance when both balance planes are between the bearings of the balance machine are shown in Figure 4-21. Dimensions a, b, c, r1 and r2 are required on all rotors being dynamically balanced. The operator picks the configuration that matches the rotor being balanced and completes the information as required (for example, the measurements would be different for an overhung rotor versus a rotor with one plane overhung and the other between bearing supports). In most cases, the two balance planes are at the same radius, which simplifies evaluation of the balance results displayed on the screen of the balance machine. The operator also puts the balance tolerance (ounce-inches or gram-inches) in the setup for the rotor balance. This allows the operator to know when the desired tolerance has been reached.

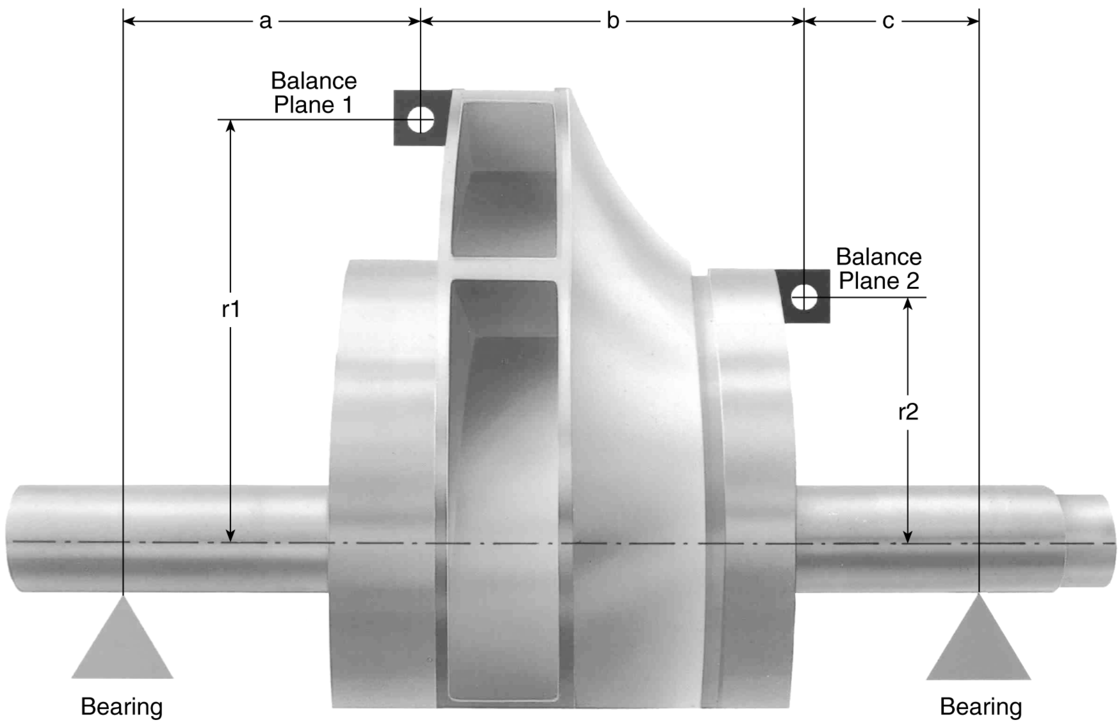


Figure 4-21
Balance Machine Setup for Between Bearing Rotor Balance

A static balance is a single-plane balance that is used when the distance between the available balance planes is small. In the setup, the c and r2 dimensions are not required. The majority of the balances are performed as a dynamic (two-plane) balance. For multi-stage vertical pumps, such as in heater drain and condensate pump applications, a dynamic balance can be performed for the impellers between the bearings, which normally are the second-stage through last-stage impellers. Once the rotor has been dynamically balanced, the first-stage impeller should be installed. The first-stage impeller will be mounted on the shaft outside the bearing pedestals of the balance machine and a static (single-plane) balance will be performed on the first-stage impeller. The shaft with impellers becomes the mandrel for the first-stage impeller balance.

Once the rotating element meets balance tolerance, as indicated by the balance machine, a confirmation run of the rotor in the balance machine should be performed. In performing the confirmation run, in addition to the unbalance value, the phase angle of the unbalance should be compared between the two balance runs. A significant deviation (for example, 45°) in phase angle provides an indication that the balance sensitivity of the machine has been exceeded. This is also true if the operator has difficulty meeting the balance tolerance or the balance machine appears to be malfunctioning.

It is possible to evaluate the balance sensitivity of the machine by introducing known unbalance weights on the rotor and evaluating against the expected response. For a given radius where the unbalance can be added, it is recommended that the weight added be equivalent to the balance tolerance. If the balance machine can identify the weight and location, it can be concluded that the unbalance sensitivity of the machine was adequate for the balance tolerance and the rotor unbalance was within that tolerance. If the balance machine will not reliably identify the weight added, continue to add weight in increments of the balance tolerance and check the machine. Once the machine can identify the weight added, one has an idea of the sensitivity of the machine as it relates to the balance tolerance (for example, within 1, 2, 3, 4, etc. times the balance tolerance).

4.7.4 Balance Tolerances

In the U.S., the balance tolerance (permissible unbalance, U) will typically be calculated as a ratio of KW/N, where W = the weight in pounds of the component or rotor being balanced, N = the operating speed in rpm of the pump when installed at the plant, and K = a balance tolerance multiplier. The results of the calculation will be a value expressed in ounce-inch.

For example, the balance tolerance for a rotor weighing 300 pounds and operating at 1200 rpm would be:

for, K = 1

U = KW/N = 1(300)/1200	=	0.25 ounce-inch	
	=	0.25(28.35)	= 7.1 gram-inch
		(28.35 grams	= 1 ounce)

for, K = 4

U = 4W/N = 4(300)/1200	=	1.0 ounce-inch = 28.35 gram-inch

This report recommends a 4 W/N balance tolerance as the standard for vertical pump rotating assemblies, with impellers individually balanced to W/N. ISO 1940/1 and ANSI S2.19 provide balance grades for various groups of rigid rotors as shown in Figure 4-22. The ISO recommends a G 6.3 balance grade for pump impellers. The following formulas can also be used to calculate the balance tolerance for a given ISO standard.

U = 6 (ISO) W/N permissible unbalance in ounce-inch

U = 170 (ISO) W/N permissible unbalance in gram-inch

From these formulas, the G 6.3 balance grade would be equivalent to 37.8 W/N or 9.45 times greater than the tolerance of 4 W/N. ISO G 1.0 grade is equivalent to a 6 W/N balance tolerance. From the previous example, the balance tolerance for ISO G 1.0 would be as follows:

U = 6 (1) 300/1200 = 1.5 ounce-inch

or

U = 170 (1) 300/1200 = 42.5 gram-inch

ISO provides nomograms for the different balance grades to determine the balance tolerance for a component being balanced (Figures 4-23, 4-24, and 4-25). Note that the maximum rotor weight for balance grades 0.4 and 1.0 would be 100 pounds.

W/N and 4 W/N are tight tolerances for rotor unbalance, but they can be achieved without much difficulty if the balance machine, setup, and personnel are capable. Reducing rotor unbalance reduces alternating stresses in the pump shaft and rotating components, which increases the possibility of maximum trouble free pump service.

Balance Quality Grade	Rotor Types - General Examples
G 4000(1)	Crankshaft-drives (2) of rigidly mounted slow marine diesel engines with uneven number of cylinders (3).
G 1600	Crankshaft-drives of rigidly mounted large two-cycle engines.
G 630	Crankshaft-drives of rigidly mounted large four-cycle engines. Crankshaft-drives of elastically mounted marine diesel engines.
G 250	Crankshaft-drives of rigidly mounted fast four-cylinder diesel engines (3).
G 100	Crankshaft-drives of fast diesel engines with six or more cylinders (3). Complete engines (gasoline or diesel) for cars, trucks, and locomotives (4).
G 40	Car wheels (5), wheel rims, wheel sets, drive shafts. Crankshaft-drives of elastically mounted, fast four-cycle engines (gasoline or diesel) with six and more cylinders (3). Crankshaft-drives for engines of cars, trucks, and locomotives.
G 16	Drive shafts (propeller shafts, cardan shafts) with special requirements. Parts of crushing machinery. Parts of agricultural machinery. Individual components of engines (gasoline or diesel) for cars, trucks, and locomotives. Crankshaft-drives of engines with six or more cylinders under special requirements.
G 6.3	Parts of process plant machines. Marine main turbine gears (merchant service). Centrifuge drums. Paper machinery rolls. Printing rolls. Fans. Assembled air craft gas turbine rotors. Fly wheels. Pump impellers. Machine-tool and general machinery parts. Medium and large electric armatures (of electric motors having at least 80 mm shaft height) without special requirements. Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration isolating mountings. Individual components of engines under special requirements.
G 2.5	Gas and steam turbines, including marine main turbines (merchant service). Rigid turbogenerator rotors. Computer memory drums and discs. Turbo-compressors. Machine tool drives. Medium and large electric armatures with special requirements. Small electrical armatures not qualifying for one or both of the conditions stated in G 6.3 for such. Turbine-driven pumps.
G 1	Tape recorder and phonograph drives. Grinding-machine drives. Small electrical armatures with special requirements.
G 0.4	Spindles, discs, and armatures of precision grinders. Gyroscopes.

NOTES:

(1) The quality grade number represents the maximum permissible orbital velocity $e_{per} \cdot$ of the center of gravity in mm/sec around the shaft axis (in radians/sec).

(2) A crankshaft drive is an assembly which includes the crankshaft, a flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc.

(3) For the purposes of this recommendation, slow diesel engines are those with a piston velocity of less than 30 ft/sec, fast diesel engines are those with a piston velocity of greater than 30 ft/sec.

(4) In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft-drive.

(5) G 16 is advisable for off-the-car balancing due to clearance or runout in central pilots or bolt hole circles.

Figure 4-22
ISO 1940/1 and ANSI S2.19 Balance Grades for Various Groups of Rigid Rotors

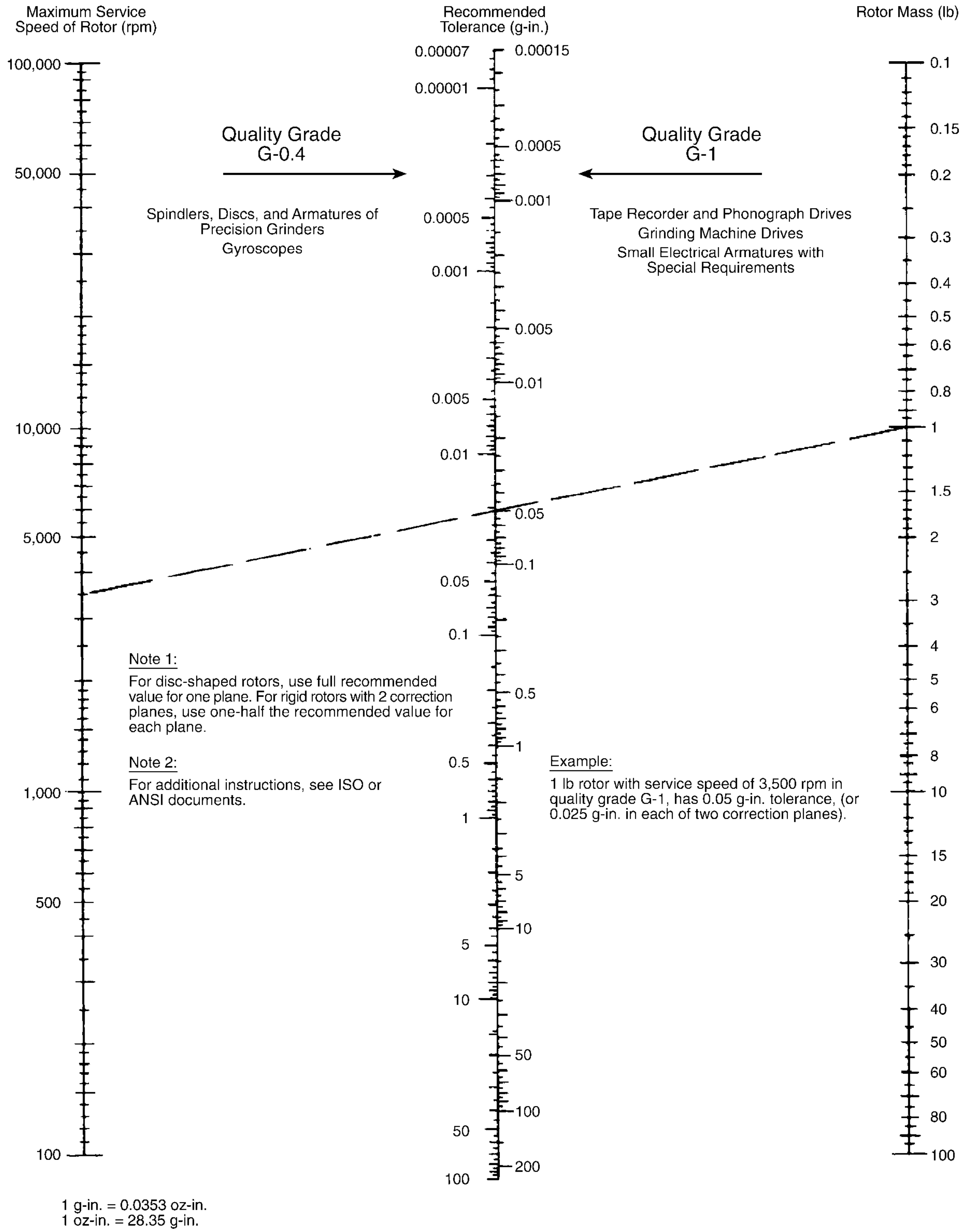


Figure 4-23
ISO 1940/1 Nomogram for Grades G 0.4 and G 1.0 – Rotor Weight 0.100 to 100 Pounds

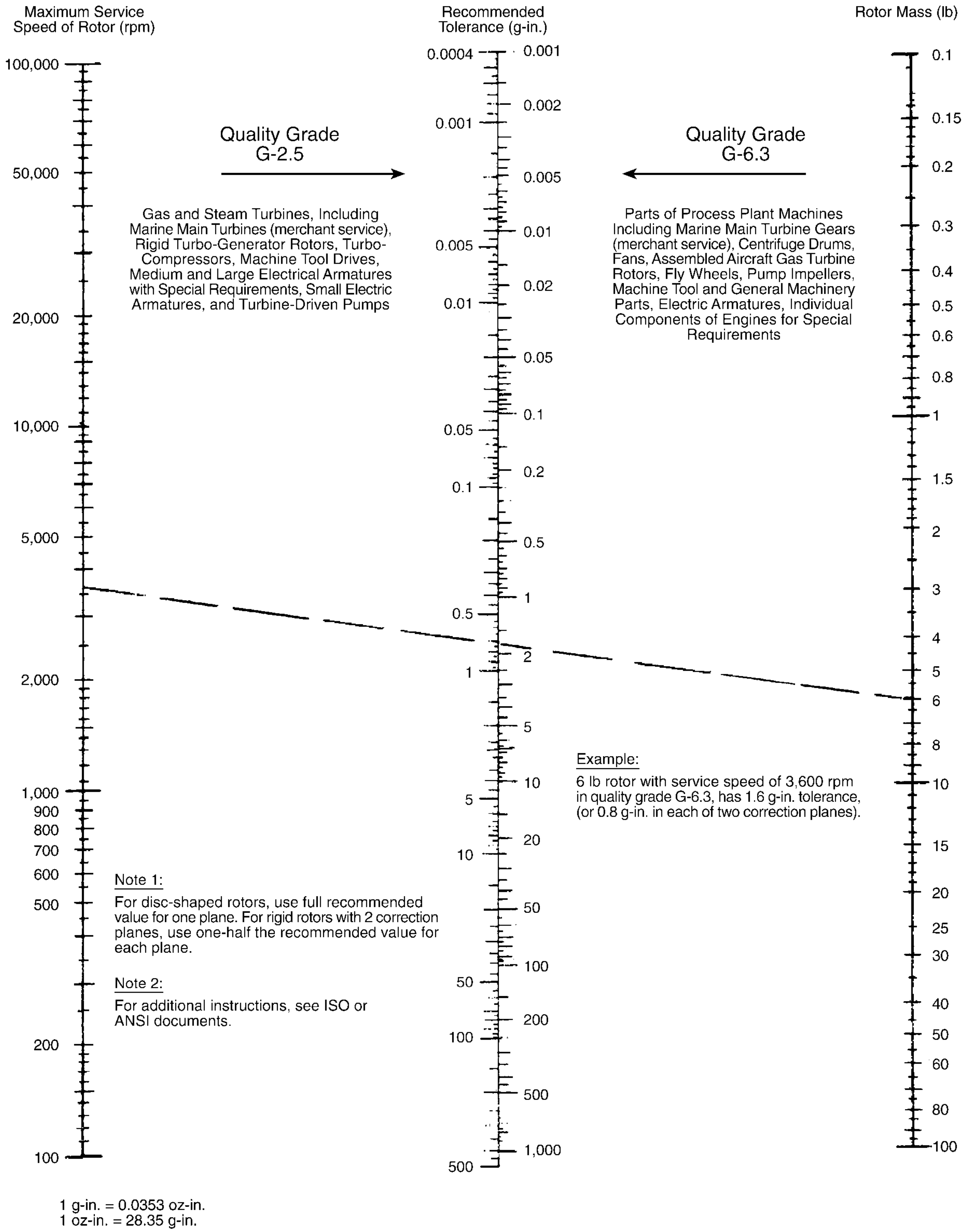


Figure 4-24
ISO 1940/1 Nomogram for Grades G 2.5 and G 6.3 – Rotor Weight 0.100 to 100 Pounds

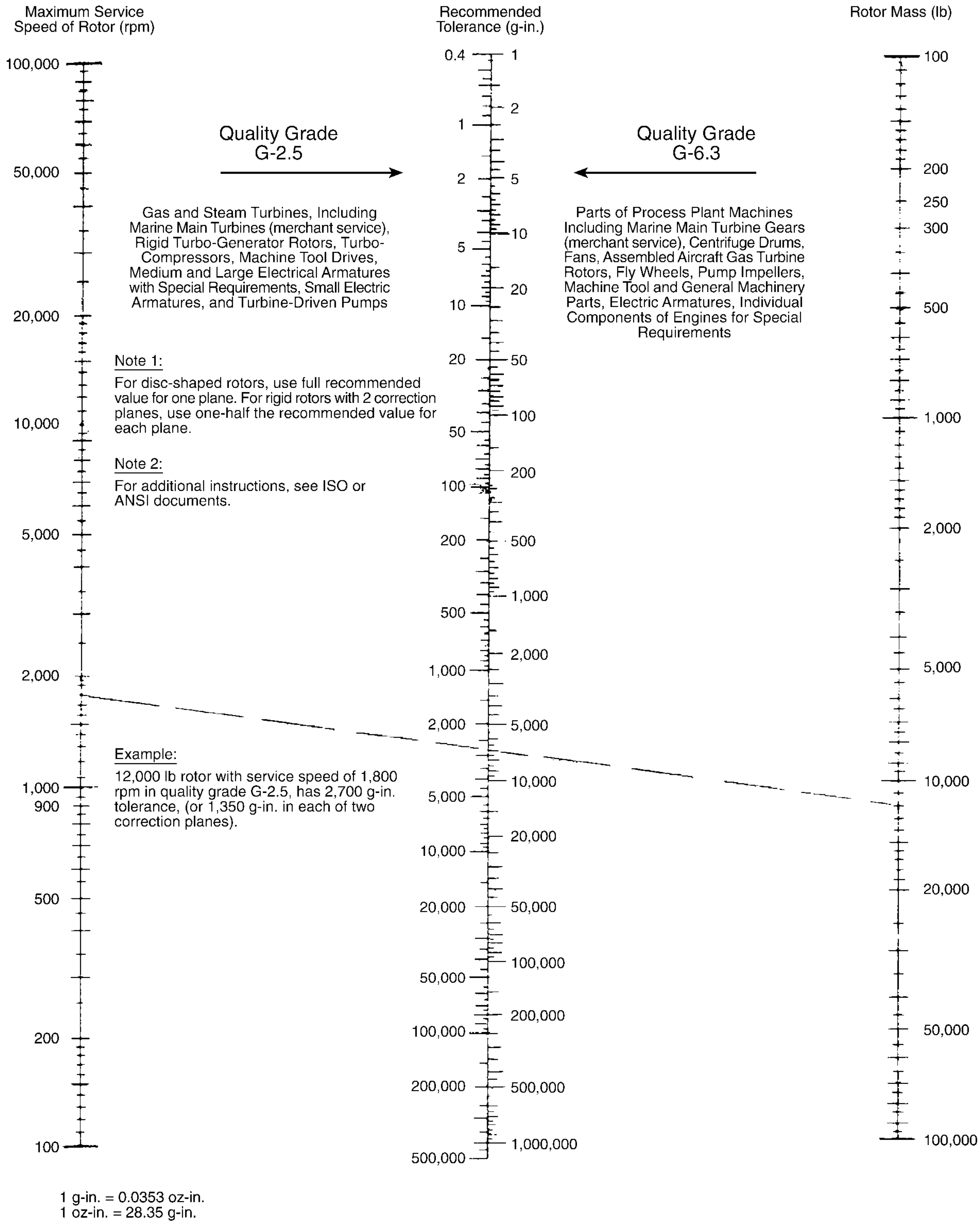


Figure 4-25
ISO 1940/1 Nomogram for Grades G 2.5 and G 6.3 – Rotor Weight 100 to 100,000 Pounds

4.8 Abrasive Wear of Shaft/Shaft Sleeves and Bearings

One of the major causes of pump degradation is abrasive wear in the bearing journals and gland packing areas. This causes running clearances, pump vibration, and seal leakage to increase. In any abrasive bearing application, the optimum mating surfaces are the hardest practical shaft surface combined with the softest bearing material that will support the load. With this combination, abrasive particles are allowed to be pushed into the soft bearing surface and roll or slide through the contact sector with very little damage to the shaft or bearing. Typical bearing materials used in vertical pump bearing applications include bronze, rubber, graphite-impregnated carbon (for example, Grafaloy), and elastomeric made from thermosetting resins (for example, Thordon). The 400 and 300 series stainless steel and carbon steel shaft materials used in vertical pumps have good ductility properties, which is a must in alternating bending stress applications such as pumps. Higher shaft hardness properties can be obtained but at the expense of shaft ductility. Therefore, it becomes necessary to locally harden the shaft bearing journal areas to improve pump service life in abrasive applications. Different coating wear performances are presented with Thordon Composite bearing material (Figure 4-26).

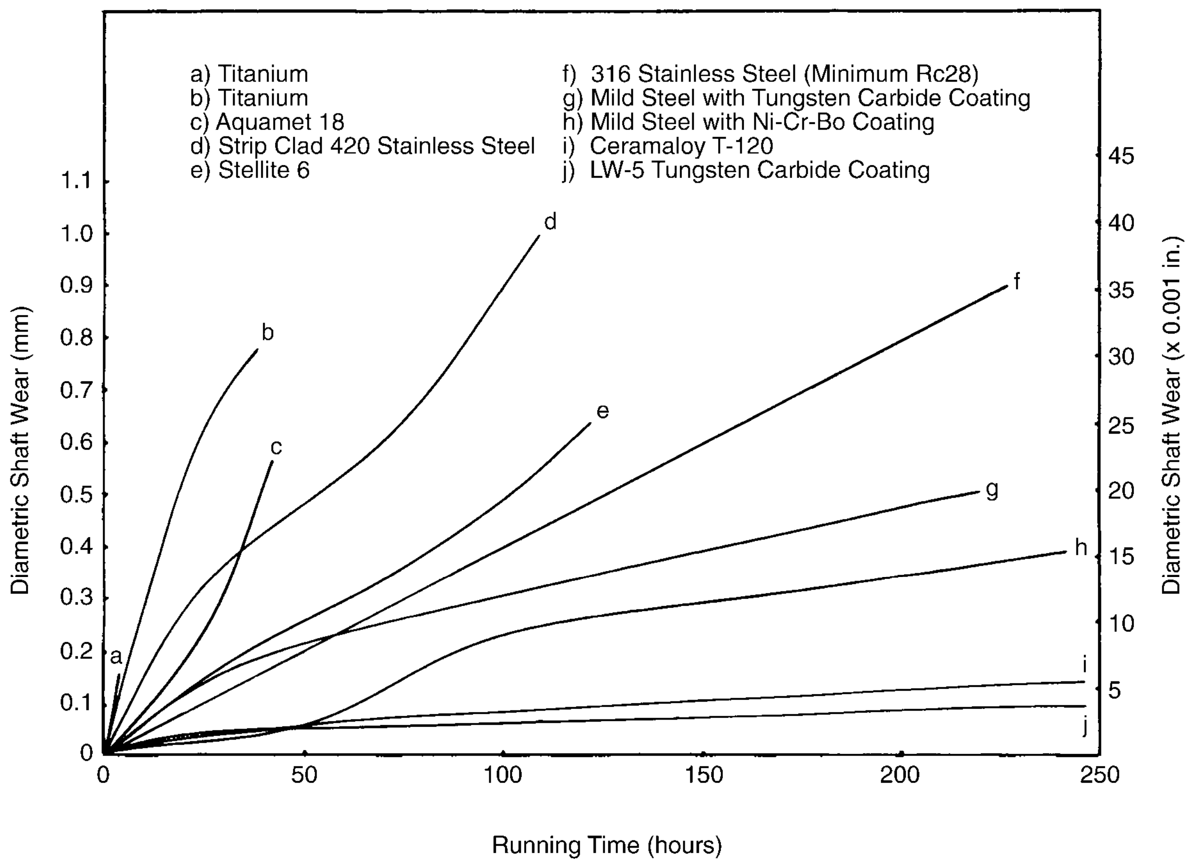


Figure 4-26
Coating Wear Performance with Thordon Composite Bearing Material

There are four basic families of hard coated materials: ceramics, tungsten carbides, cobalt base alloys, and nickel-chrome boron (NCB) alloys. Tungsten carbide and NCB coatings are the more prevalent coatings used for pump shaft journal surfacing. The basic difference between the two coatings is the bonding mechanism with the base material. The NCB coatings are metallurgical bonds and the tungsten carbide coatings, as well as the ceramic and cobalt base alloys, are mechanical bonds. The metallurgical bond coating (or fuse coating) provides a dense, porous-free surface that will not chip, crack, or spall and has a very low coefficient of friction. However, fused coatings are usually heated to 1900°F to 2100°F to allow bonding to the substrate, which can make it impractical for restoring shaft and shaft sleeve surfaces (that is, the heat during application distorts the components). A thermal spray process called High Velocity Oxygen Fuel (HVOF) can be used to apply tungsten carbide materials, which does not distort the base material. The HVOF bond is a mechanical bond with the shaft; however, it appears superior to any other non-metallurgical (mechanical) bonding process available on the market.

Key Technical Point

The HVOF bond is a mechanical bond with the shaft; however, it appears superior to any other mechanical bonding process available on the market.

4.8.1 High Velocity Oxygen Fuel (HVOF) Thermal Spray Process

The HVOF Thermal Spray Process (Figures 4-27 and 4-28) is basically the same as the combustion powder spray process except that this process has been developed to produce an extremely high spray velocity. There are a number of HVOF guns that use different methods to achieve high velocity spraying. One method is basically a high pressure water cooled combustion chamber and long nozzle. Fuel (kerosene, acetylene, propylene and hydrogen) and oxygen are fed into the chamber where combustion produces a hot high pressure flame which is forced down a nozzle increasing its velocity. Powder may be fed axially into the combustion chamber under high pressure or fed through the side of the nozzle where the pressure is lower. Another method uses a simpler system of a high pressure combustion nozzle and air cap. Fuel gas (propane, propylene or hydrogen) and oxygen are supplied at high pressure, combustion occurs outside the nozzle but within an air cap supplied with compressed air. The compressed air pinches and accelerates the flame and acts as a coolant for the gun. Powder is fed at high pressure axially from the centre of the nozzle.

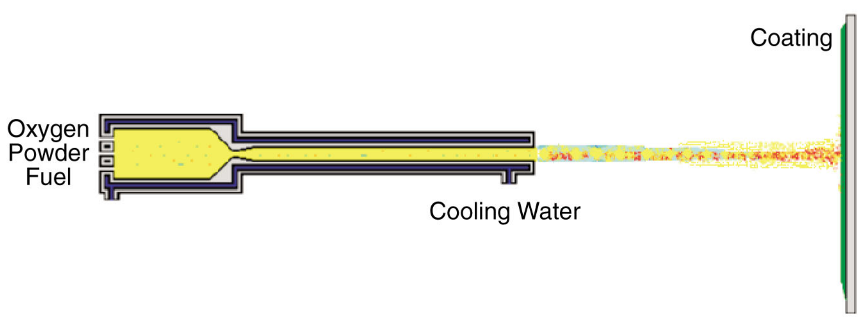


Figure 4-27
Schematic Diagram of the HVOF Process

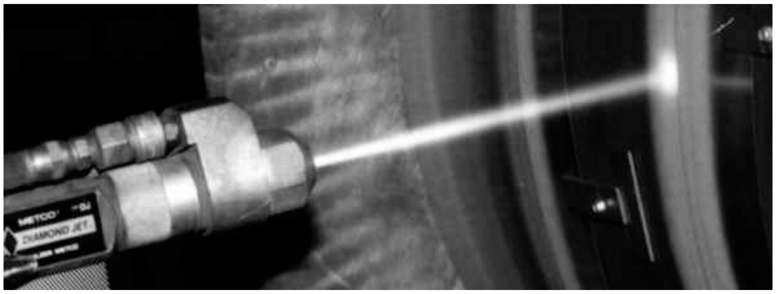


Figure 4-28
HVOF Application

The coatings produced by HVOF are similar to those produce by the detonation process. Coatings are very dense, strong and show low residual tensile stress or in some cases compressive stress, which enable very much thicker coatings to be applied than previously possible with the other processes.

The very high kinetic energy of particles striking the substrate surface reduces the need for the particles to be fully molten to form high quality coatings. This is certainly an advantage for the carbide type coatings and is where this process really excels.

Key Technical Point

HVOF coatings are used in applications requiring the highest density and strength not found in most other thermal spray processes.

4.8.2 Nature of Thermal Spray Coatings

What is a thermal (flame) spray coating? It is a coating produced by a process in which molten or semi-molten particles are applied by impact onto a substrate. A common feature of all thermal spray coatings is their lenticular or lamellar grain structure resulting from the rapid solidification of small globules, flattened from striking a cold surface at high velocities. The bonding mechanisms at the coating/substrate interface and between the particles making up the coating is an area which in many cases is still subject to speculation. It generally suffices to state that both mechanical interlocking and diffusion bonding occur (see Figures 4-29, 4-30, and 4-31).

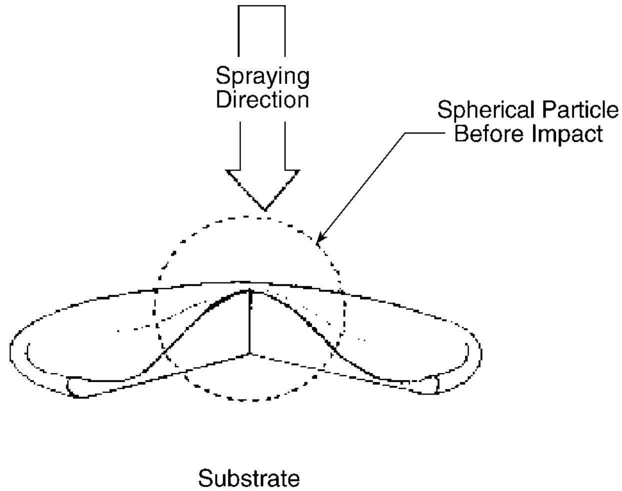


Figure 4-29
Schematic Diagram of Spherical Particle Impinged onto a Flat Substrate

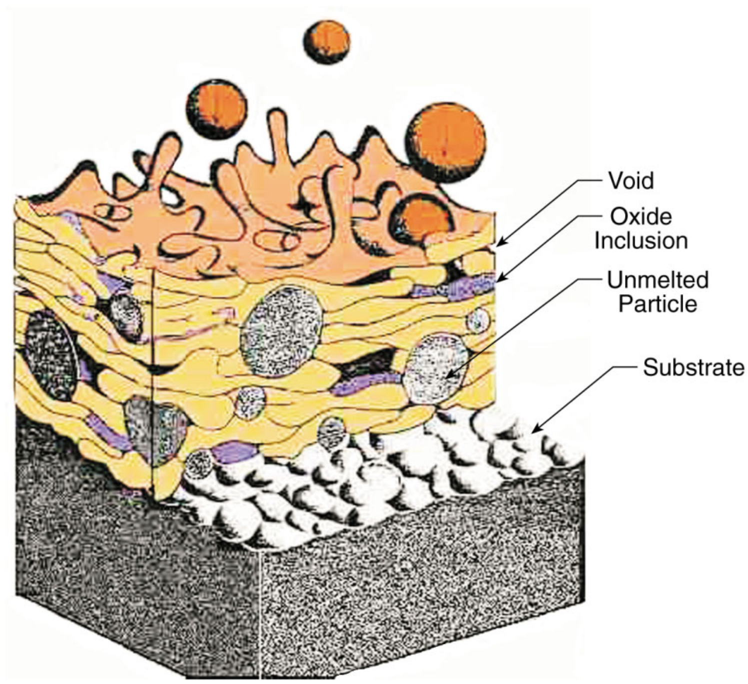


Figure 4-30
Schematic Diagram of Thermal Spray Metal Coating

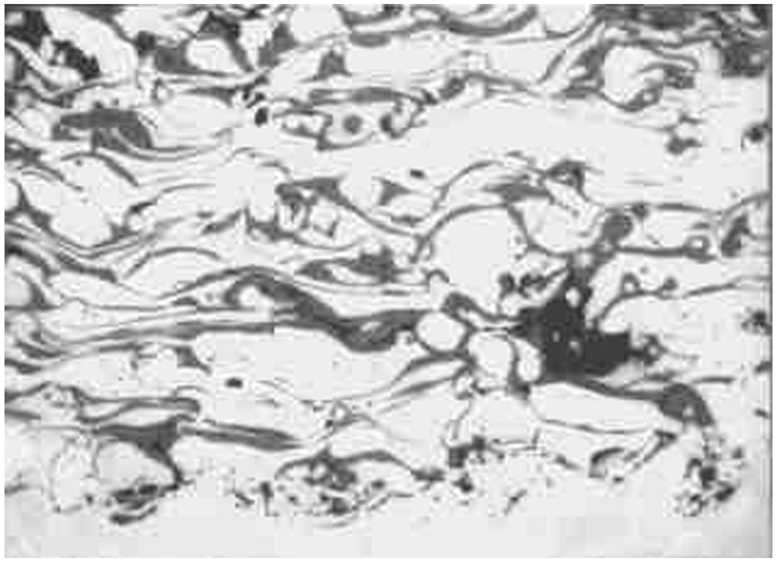


Figure 4-31
A Typical Microstructure of a Metallic Sprayed Coating (The lamellar structure is interspersed with oxide inclusions and porosity.)

Bonding Mechanisms:

- Mechanical keying or interlocking.
- Diffusion bonding or Metallurgical bonding.
- Other adhesive, chemical and physical bonding mechanisms -oxide films, Van der Waals forces etc.

Factors effecting bonding and subsequent build up of the coating:

- Cleanliness
- Surface area
- Surface topography or profile
- Temperature (thermal energy)
- Time (reaction rates and cooling rates etc.)
- Velocity (kinetic energy)
- Physical and chemical properties
- Physical and chemical reactions

Cleaning and grit blasting are important for substrate preparation. This provides a more chemically and physically active surface needed for good bonding. The surface area is increased which will increase the coating bond strength. The rough surface profile will promote mechanical keying.

Individual particle cooling rates on impact can be of the order of 1 million degrees C per second ($10^6 Ks^{-1}$). Thermal interaction is obviously very limited which is important with regard to diffusion bonding (temperature and time dependent).

Increase in thermal and kinetic energy increases chances of metallurgical bonding (temperature, velocity, enthalpy, mass, density, specific heat content, etc.). Spray materials like Molybdenum, Tungsten, and Aluminum/metal composites produce so called “self bonding” coatings. These materials have comparatively high bond strengths (increased metallurgical or diffusion bonding) and can bond to clean polished substrates.

Molybdenum and other refractory metals have very high melting points thus the interaction between substrate and coating particles will be increased due to the higher temperatures involved and longer cooling cycles. Also molybdenum oxide volatilizes and does not get in the way of metallurgical bonding.

Aluminum/metal composites produce increased levels of exothermic reaction due to reactions of aluminum with metals like nickel to produce nickel aluminide and with oxygen producing aluminum oxide. The increased thermal action increases the degree of diffusion bonding. Higher preheat temperatures for the substrate increases diffusion bonding activities but will also increase oxidation of the substrate which could defeat the objective of higher bond strengths. High kinetic energy spraying using HVOF produce high bond strengths due to the energy liberated from high velocity impacts. The high density tungsten carbide/cobalt materials are a good example.

Metallurgical or diffusion bonding occurs on a limited scale and to a very limited thickness (0.5 µm max. with heat the affected zone @ 25µm) with the above type coatings. Fused coatings are different. These are re-melted and are a complete metallurgical bond with the substrate and itself.

4.8.3 Coating Structure

High cooling rates or super cooling (10^6 Ks^{-1}) of particles can cause the formation of unusual amorphous (glassy metals) micro-crystalline and meta-stable phases not normally found in wrought or cast materials.

A large proportion of thermal spraying is conducted in air or uses air for atomization. Chemical interactions occur during spraying, notably oxidation. Metallic particles oxidize over their surface forming an oxide shell. This is evident in the coating microstructure as oxide inclusions outlining the grain or particle boundaries. Some materials (such as titanium) interact with or absorb other gases such as hydrogen and nitrogen.

Coatings show lamellar or flattened grains appearing to flow parallel to the substrate. The structure is not isotropic, with physical properties being different, parallel to substrate (longitudinal) rather than across the coating thickness (transverse). Strength in the longitudinal direction can be 5 to 10 times that of the transverse direction.

The coating structure is heterogeneous relative to wrought and cast materials. This is due to variations in the condition of the individual particles on impact. It is virtually impossible to ensure that all particles are the exact same size and achieve the same temperature and velocity. All conventionally thermal sprayed coatings contain some porosity (0.025% to 50%) caused by:

- Low impact energy (un-melted particles/low velocity)
- Shadowing effects (un-melted particles/spray angle)
- Shrinkage and stress relieve effects

The above interactions can make the coatings very different from their starting materials chemically and physically.

4.8.4 Coating Stress

Cooling and solidification of most materials is accompanied by contraction or shrinkage. As particles strike they rapidly cool and solidify. This generates a tensile stress within the particle and a compressive stress within the surface of the substrate. As the coating is built up, so are the tensile stresses in the coating. With a lot of coatings a thickness will be reached where the tensile stresses will exceed that of the bond strength or cohesive strength and coating failure will occur (see Figure 4-32).

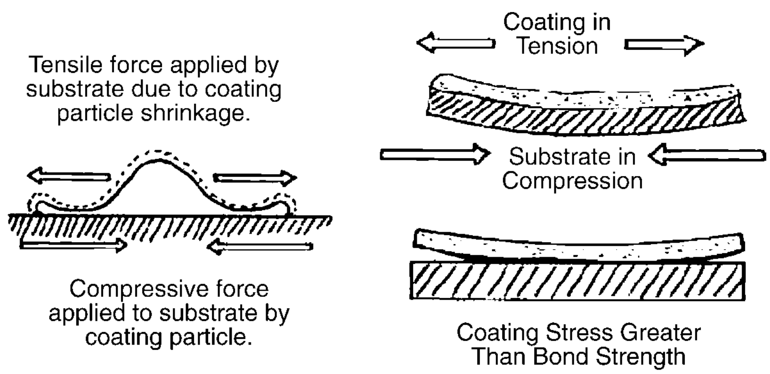


Figure 4-32
Tensile and Compressive Forces

High shrink materials like some austenitic stainless steels are prone to high levels of stress build up and thus have low thickness limitations. Look for thickness limitation information on coating data sheets. Generally thin coatings are more durable than thick coatings.

Spraying method and coating microstructure influence the level of stress build up in coatings. Dense coatings are generally more stressed than porous coatings. Notice that combustion powder sprayed coatings generally have greater thickness limitations than plasma coatings.

Contrary to that just mentioned, the systems using very high kinetic energy and low thermal energy (HVOF) can produce relatively stress free coatings that are extremely dense. This is thought to be due to compressive stresses formed from mechanical deformation (similar to shot peening) during particle impact counteracting the tensile shrinkage stresses caused by solidification and cooling.

4.8.5 Coating Properties

Compare coatings to their wrought or cast equivalents:

Property………………..	Coating………………….	Cast/Wrought
Strength………………………	low (5-30%)…….………….	100%
Ductility………………………	very low (1-10%)……………	100%
Impact………………………..	low………………………..….	high
Porosity………………………	yes (not if fused)……….……	in some castings
Hardness……………………..	slightly higher (micro-hardness)	

Wear resistance………………	high……………………….…….	low
Corrosion…………………….	low resistance…………….……….	high resistance
Machining……………………	poor…………………….………	good

This comparison generally shows coating properties in a bad light, and does not take into consideration that coatings are usually supported by a substrate. Coatings are generally only used to give surface properties such as wear resistance and not to add strength. Remember, bulk strength supplied by the substrate (cheap, strong and ductile). Surface properties supplied by the coating (wear and corrosion, etc.). Due to the small quantity of material required for a coating, more exotic materials can be used economically. The properties of some coatings cannot be fabricated by any other method. Properties of coatings should be considered in their own right and not the properties of the original material prior to spraying as they can be very different physically and chemically.

Key Technical Point

Remember, bulk strength is supplied by the substrate (cheap, strong and ductile). Surface properties are supplied by the coating (wear and corrosion, etc.).

Porosity

Porosity is present in most thermally sprayed coatings (except VPS, post heat treated coatings or fused coatings). One to 25% porosity is normal but can be further manipulated by changes in process and materials. Porosity can be detrimental in coatings with respect to:

- Corrosion - (sealing of coatings advised).
- Machined finish.
- Strength, macro-hardness and wear characteristics.

Porosity can be important with respect to:

- Lubrication - porosity acts as reservoir for lubricants.
- Increasing thermal barrier properties.
- Reducing stress levels and increasing thickness limitations.
- Increasing shock resisting properties.
- Abradable in clearance control coatings.
- Applications in prosthetic devices and nucleate boiling etc.

Oxide

Most metallic coatings suffer oxidation during normal thermal spraying in air. The products of oxidation are usually included in the coating. Oxides are generally much harder than the parent metal. Coatings of high oxide content are usually harder and more wear resistant. Oxides in coatings can be detrimental towards corrosion, strength and machining.

Surface Texture

Generally the as-sprayed surface is rough and textured. The rough and high bond strength coatings are ideal for bond coats for less strongly bonding coatings. Many coatings have high friction surfaces as-sprayed and this property is made use of in many applications. Some plasma sprayed ceramic coatings produce smooth but textured coatings important in the textile industry. Other applications make use of the abrasive nature of some coating surfaces. Thermally sprayed coatings do not provide bright high finish coatings with out finishing for example with electroplated deposits.

Strength

Coatings generally have poor strength, ductility and impact properties. These properties tend to be dictated by the “weakest link in the chain” which in coatings tends to be the particle or grain boundaries and coating/substrate interface. Coatings are limited to the load they can carry, and thus require a substrate for support. Coatings are poor when point loaded.

Internal tensile coating stresses generally adversely effect properties. Effective bond strength is reduced and can be destroyed by increasing levels of internal stress. This in turn affects coating thickness limits. Coatings on external diameters can be built up to greater thickness than that on internal diameters.

Surface properties such as wear resistance are usually good, but the properties are more specific to the material or materials used in the coating. The properties of a substrate need only to be strength, ease of fabrication and economic (like mild steel). The coating supplies the specific surface properties desired. For example, materials used for applications of thermal barrier and abradable clearance control by nature have poor strength and thus benefit from being applied as a coating onto a substrate which supplies the strength. Some Properties Thermally Sprayed Coatings can provide:

- Tribological (wear, resistance)
- Corrosion resistance
- Heat resistance
- Thermal barrier
- Electrical conductivity or resistivity
- Abradable or abrasive
- Textured surfaces
- Catalyst and prosthetic properties
- Restoration of dimension
- Copying of intricate surfaces

4.8.6 Nondestructive Testing of Thermally Sprayed Coatings

There are very few reliable NDT methods available for thermally sprayed coatings. The majority of tests for coatings tend to be of a destructive nature, which, obviously can not be used on the actual coated part going into service and therefore, must be considered as a test for process control. The main practical NDT methods used are:

- Dimensional measurements- micrometer, eddy current and magnetic thickness measurement devices, etc. (see Figure 4-33).
- Machining tests or response of coating during machining operations is a good test for general integrity.
- Visual inspection - grit blast, spraying, coating/substrate, machined finish.
- Dye penetrant - used in limited applications, natural coating porosity fogs flaw indications.

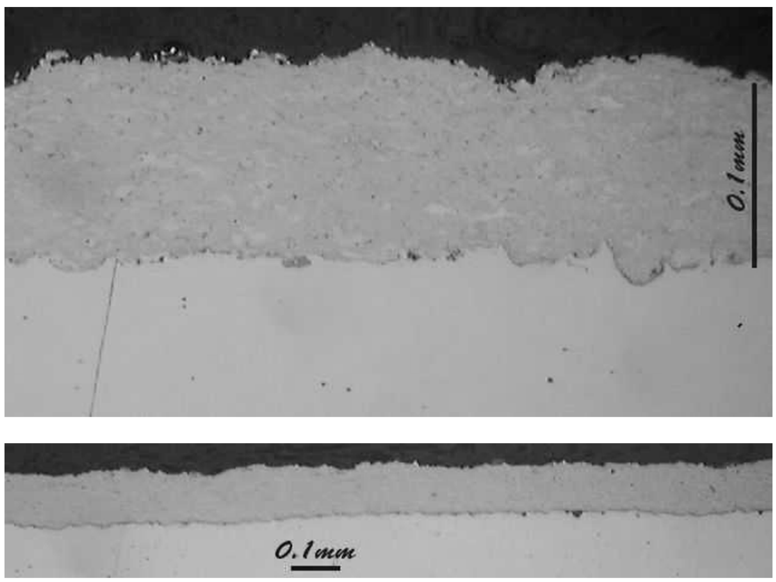


Figure 4-33
Thermal Spray Coating Photomicrographs [HVOF Sprayed Tungsten Carbide/Chromium Nickel Coating (WC/207NiCr)]

Ultrasonic and magnetic particle flaw detection methods have proved to be poor with thermally sprayed coatings due to the very high number of particle boundaries giving flaw like responses and causing high levels of interference. Hardness testing is generally considered a destructive test for coatings unless made in a non-working area. Advanced techniques like thermography, thermal wave interferometry, and acoustic emission are presently being researched and are still laboratory set-ups with limited practical use for industry.

Destructive testing such as hardness, bend, bond strength, metallography, etc., are important to prove the process and coating integrity expected in the component. The limited non-destructive testing available for thermally sprayed coatings should emphasize the need for a high standard of quality control over the process, to ensure a high level of confidence in the coated products.

> **Key Human Performance Point**
>
> The limited non-destructive testing available for thermally sprayed coatings should emphasize the need for a high standard of quality control over the process, to ensure a high level of confidence in the coated products.

4.9 NPSH and Submergence

The function of the intake, whether an open channel or a tunnel having 100% wetted perimeter, is to supply an evenly distributed flow of water to the suction bell. An uneven distribution of flow, characterized by strong local currents, favors formation of vortices and, with certain low values of submergence, will introduce air into the pump with reduction of capacity accompanied by noise. Uneven distribution can also increase or decrease the power consumption with a change in total developed head. There can be vortices that do not appear on the surface and these might also have adverse effects.

The amount of submergence for successful operation will depend greatly on the approaches to the intake and the size of the pump. Centrifugal pumps in intake sumps must be submerged deeply enough to provide:

- A pressure or NPSH sufficient to prevent cavitation in pump first stage
- Prevention of vortices and associated pit flow problems detrimental to pump operation

A pump can have adequate submergence from a NPSH standpoint and still be lacking in sufficient depth of cover above the suction inlet to prevent surface air from being drawn in. Any wet-pit pump must have its suction inlet submerged at all times and, for continuous pumping, every pump will have a fixed minimum submergence requirement. Because this relates to velocity, there are two basic parameters:

- Suction inlet diameter
- Depth of water above the inlet lip

As pump size and flow increase, the inlet velocity might stay constant as the bell diameter increases but, at the same time, the impeller distance above the suction inlet becomes larger, so that a fixed submergence value would lead to increased surface velocity, peripheral drawdown, and an increase in air intake. Basically, submergence must increase with pump size.

The Hydraulic Institute defines NPSH as the total suction head in feet absolute, determined at the suction nozzle and corrected to a datum, less the vapor pressure of the liquid in feet absolute. Simply stated, it is an analysis of energy conditions on the suction side of a pump to determine if the liquid will vaporize at the lowest pressure point in the pump.

> **Key Technical Point**
>
> Simply stated, a calculation of NPSH is an analysis of energy conditions on the suction side of a pump to determine if the liquid will vaporize at the lowest pressure point in the pump.

The pressure that a liquid exerts on its surroundings is dependent upon its temperature. This pressure, called *vapor pressure*, is a unique characteristic of every fluid and increases with increasing temperature. When the vapor pressure within the fluid reaches the pressure of the surrounding medium, the fluid begins to vaporize or boil. The temperature at which this vaporization occurs will decrease as the pressure of the surrounding medium decreases.

A liquid increases greatly in volume when it vaporizes. One cubic foot of water at room temperature becomes 1700 cu. ft. of vapor at the same temperature. NPSH is simply a measure of the amount of suction head present to prevent this vapor fluxion at the lowest pressure point in the pump.

The NPSH-required is a function of the pump design. As the liquid passes from the pump suction to the eye of the impeller, the velocity increases and the pressures decrease. There are also pressure losses due to shock and turbulence as the liquid strikes the impeller. The centrifugal force of the impeller vanes increases the velocity and decreases the pressure of the liquid. The NPSH-required is the positive head in feet absolute required at the pump suction to overcome these pressure drops in the pump and maintain the liquid above its vapor pressure. The NPSH-required varies with speed and capacity within any particular pump. Pump manufacturers' curves normally provide this information.

NPSH-available is a function of the system in which the pump operates. It is the excess pressure of the liquid in feet absolute over its vapor pressure as it arrives at the pump suction.

PB = Barometric pressure, in feet absolute.	VP = Vapor pressure of the liquid at maximum pumping temperature, in feet absolute.
p = Pressure on surface of liquid in closed suction tank, in feet absolute.	Ls = Maximum static suction lift in feet.
LH = Maximum static suction head in feet.	hf = Friction loss in feet in suction pipe at required capacity.

In an existing system, the NPSH-available can be determined by a gage on the pump suction. The following formula applies:

$$NPSH_A = P_S = V_P + Gr \pm H_V$$

Where

Gr = Gage reading at the pump suction expressed in feet (plus if above atmospheric, minus if below atmospheric), corrected to the pump centerline

H_V = Velocity head in the suction pipe at the gage connection, expressed in feet

Refer to Appendix B for the NPSH calculation and further explanation.

4.10 Shaft Material

Materials used in vertical pump shafts and shaft couplings have failed in service due to intergranular stress corrosion cracking (IGSCC). Common shaft materials, Type 410 and Type 416 martensitic stainless steel, are susceptible to tempering embrittlement. The NRC has published IN 2007-05, which explains that 23 pump shaft and coupling failures have occurred since 1983; many of these failures were due to IGSCC.

Examples include the following:

- Perry Nuclear Plant May 21, 2004, 416 SS coupling failure, keyed design that was replaced with 17-4 precipitation-hardened SS and thicker wall thickness coupling.
- Perry Nuclear Plant September 1, 2003, 416 SS coupling failure, keyed design in which three root causes were identified: unclear guidance for the use of maintenance procedures, procedure content less than adequate, and couplings made from SCC-susceptible material with high stress concentrations.
- Virgil C. Summer Nuclear Station failure of C service water pump during testing; pump shaft coupling failure.
- Beaver Valley Nuclear Power Plant, June 20, 1991, pump shaft coupling failure, threaded design, Type 410 SS, low-impact strength due to temper embrittlement resulting from improper heat treatment.
- Beaver Valley Nuclear Station, 1992, coupling cracks, keyed design.
- Grand Gulf, March 3, 1994, extensive corrosion of carbon steel bolts and lock washers used in pump shaft coupling assemblies. Failure not detected by vibration monitoring or in-service testing program.
- Palisades Nuclear Plant, August 9, 2011, coupling failure due to IGSCC—repeat of failure on September 29, 2009, ASTM A582 Type 416 SS, hardness higher than design specification (28 to 32 HRC) IGSCC requires three items for occurrence: a susceptible material, tensile stress, and a corrosive environment. The material was changed to 17-4 PH SS.

  The following was determined from the Palisades root cause analysis: For 416 SS, temper embrittlement occurs between ~700 and 1050°F, resulting in a material that has significantly lower fracture toughness. This loss of toughness increases susceptibility to IGSCC and is not revealed by measuring material hardness. Hardness is not a good indicator of susceptibility to SCC because it is the heat treatment method that produces temper embrittlement. Two materials with the same hardness will have different toughness based on the method used during heat treatment.
- Watts Bar Nuclear Plant, August, 2008, keyed type coupling failure, 410 SS. This coupling failed with the pump in service completely separating the shaft at the failed coupling. Failure was due to IGSCC that originated at pits in the keyways of the coupling. Hardness of the coupling exceeded the maximum allowed by the material specification. Hardness values for this coupling ranged from 31.6 HRC at the OD to 39.6 HRC at the ID.

Figures 4-34 through 4-37 show the damaged couplings from the Watts Bar pump.

Figure 4-34
Sample of 410 SS Coupling Failure due to IGSCC

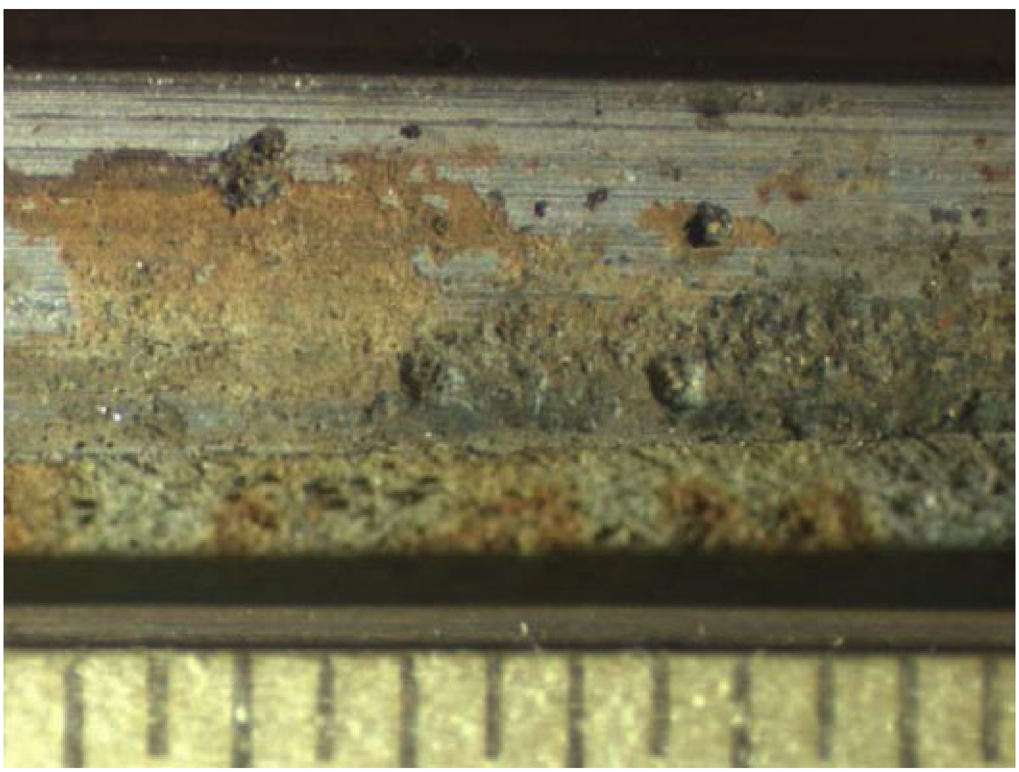


Figure 4-35
Pitting on Keyway Surface

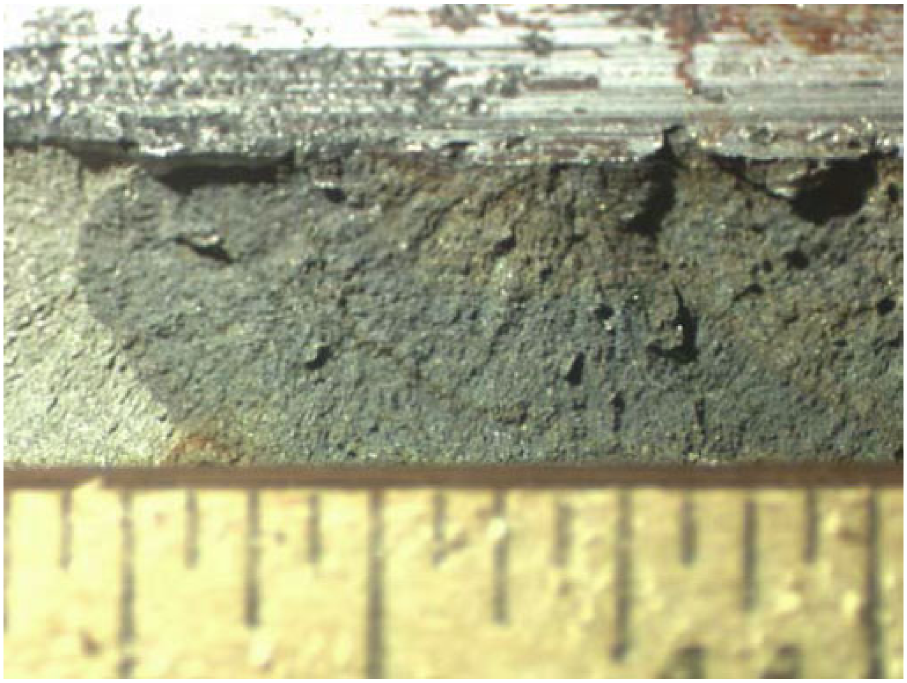


Figure 4-36
Crack Initiation at Pit

Figure 4-37
Second Coupling with Pitting on Top End

Replacement shaft and material recommendations:

- Pump shafts and line shafts shall be rolled or forged stainless steel conforming to ASTM A479 Grade XM-19 with bearing journal and throttle bushing surfaces hard-faced with tungsten carbide ceramic coating applied using the high-velocity oxygen fuel thermal spray process. Shafts shall be precision ground over the entire length to 63 micro-inch RMS finish or smoother. Shafts shall be straight to total indicated run-out (TIR) not greater than 0.005 inch per shaft section.
- Line shaft couplings shall be forged stainless steel conforming to ASTM A479 Grade XM-19. Line shaft couplings shall be precision ground throughout and shall be mechanically secured against axial movement in both directions on shafts. Line shaft couplings shall use keys to drive shafts and capture split-rings to join shaft ends. Threaded couplings may be used as an alternative design. Couplings shall not be chrome plated.

Other EPRI resources exist that discuss vertical pump materials and IGSCC, including the following:

- 1022960, *Plant Engineering: Critical Pump Refurbishment and Procurement*, December 2011
- 1019154, *Plant Support Engineering: Large Vertical Pump End-of-Expected-Life Report*, December 2009
- 1009721, *Stress Corrosion Cracking in PWR and BWR Closed Cooling Water Systems*, October 2004

5
BIBLIOGRAPHY

Deep Draft Vertical Centrifugal Pump Maintenance and Application Guide. EPRI, Palo Alto, CA: 1994. NP-7413.

Engineering Manual, Version 4, Thordon Bearing, Inc.

Condensate Pump Application and Maintenance Guide. EPRI, Palo Alto, CA: 2000. TR-1000052.

Fundamentals of Balancing, 3rd Edition, Schenck-Trebel, April 1990.

Gordon England, "Thermal Spray Coatings." www.gordonengland.co.uk.

Hydraulic Institute Standard for Centrifugal Pumps for Nomenclature, Definitions, Application and Operation. ANSI/HI 1.1-1.5, 1994.

Hydraulic Institute Standards for Centrifugal, Rotary, and Reciprocating Pumps, 14th Edition. Hydraulic Institute, 1983.

Illustrated Vibration Diagnostic Chart by Technical Associates of Charlotte, Inc.

Karassik, I. J., "Centrifugal Pump Clinic, 2nd Edition," Marcel Dekker, Inc., 1989.

Karassik, I. J. Krutzsch, W. C., Fraser, W. H., Messina, J. P., *Pump Handbook, 2nd Edition.* Mcgraw-Hill Book Company, 1986.

Lobanoff, V. S. and Ross, R. R., *Centrifugal Pumps: Design & Applications, 2nd Edition.* Gulf Publishing Company, 1992.

Luedeking, A., "Vertical Condenser Pump Alignment, Case Study." Ludeca, Inc., Miami, FL, April 2001.

Luedeking, A., "Vertical Pump Laser Alignment Case Study." Ludeca, Inc., Miami, FL, April 2001.

Mueller, R. A., "Pump Hand Book." www.ramueller.com/handbook.htm.

Noon, R., "Lift Adjustments on Vertical, Mixed Flow Pumps." Cooper Nuclear Station, http://em-ntserver.unl.edu/Mechanics-Pages/jenniferglenn/jenniferglenn.htm.

"Potential Common-Mode Failure Mechanism for Large Vertical Pumps." *NRC Information Notice,* 94-45.

Pump Troubleshooting, Volume 1. EPRI, Palo Alto, CA: 2000. TR-114612.

Pump Troubleshooting, Volume 2. EPRI, Palo Alto, CA: 2000. TR-114612.

Thompson, D. M., "Thermal Coatings—Methods of Applications." Tri-State Coating & Machine, Inc.

Vertical Pump Troubleshooting Guide, Johnston Pump Company, 1994.

A

GLOSSARY OF TERMS AND SI UNIT CONVERSION FACTORS

A.1 Terminology

The following is not only a glossary of terms used in this report but also includes commonly used pump terms.

Abradable coatings – Coatings that are designed to be abraded by a mating surface to form a tight gas or air seal, while retaining good erosion resistance.

Abrasive blasting – A process for cleaning and roughening a surface by means of an abrasive directed at high velocity against the workpiece.

Abrasive wear– Wear due to hard particles or hard protuberances forced against and moving along a solid surface.

Adhesive wear – Wear due to localized bonding between contacting solid surfaces, leading to material transfer between the two surfaces or a loss from either surface.

Allowable Operating Range – The flow range for operation of the pump at a specified speed, as limited by cavitation, heating, vibration, noise, shaft deflection, fatigue, or other criteria. This range is a function of the impeller and pump design and is specified by the manufacturer.

Anode – Positively charged electrode (nozzle in plasma gun) – the electrode of an electrolytic cell at which oxidation is the principle reaction. (Electrons flow away from the anode in the external circuit. It is usually the anode where corrosion occurs and metal ions enter solution.)

Anodic coating – A coating that becomes the anode in an electrochemical cell with the substrate (cathode). The only metals in common use for thermal spraying that are anodic to iron and steel are zinc and aluminum.

Best Efficiency Point – The capacity and head at which the pump efficiency is at its maximum.

Bond – This represents the state of adhesion between the coating and the substrate. Its strength will depend on the details of the spraying process and the materials used. Bonding mechanisms can be mechanical, physical, chemical, metallurgical, or a combination of these.

Bond strength – The strength of the adhesion between the coating and the substrate. A number of test methods are in use to measure the bond strength of coatings.

Bowl Assembly Efficiency – The ratio of the bowl output to the bowl assembly input, expressed in percent. This is the efficiency that is usually shown on catalog rating charts.

Bowl Assembly Head (H_1) – The energy imparted to the liquid by the pump bowl assembly, expressed in feet or meters of liquid. It is the head developed at the discharge connection of the bowl assembly per stage as shown on the curve in the pump manufacturer's catalog.

Bowl Assembly Power Input – The power required by the bowls to deliver only a specified capacity against bowl head, as follows:

(U.S. Units) $$\text{Bowl hp - } \frac{\text{Bowl Head x Capacity}}{\text{3,960 x Bowl Efficiency}}$$

(Metric Units) $$\text{Bowl kW - } \frac{\text{Bowl Head x Capacity}}{\text{366 x Bowl Efficiency}}$$

Bowl Output – Defined as: $$\frac{\text{Capacity (Q) x Bowl Head } (H_1)}{3{,}960}$$

Bowl Output is expressed in horsepower (hp x 0.746 = kW) when Q is in gallons per minute and H_1 is in feet of water for water having a specific weight of 62.4 pounds per cubic foot or relative density of 1.0.

Capacity (Q) – The volume rate of flow through the pump per unit of time at the specified suction conditions, such as gallons per minute.

Cathode – Negatively charged electrode. Electrode of an electrolytic cell at which reduction is the principle reaction. (Electrons flow toward the cathode in the external circuit.)

Cathodic coating – Coatings that become the cathode in an electrochemical cell with the substrate (anode). This type of coating protects the substrate from corrosion only by being a complete barrier. If the coating allows the environment to reach the substrate, accelerated corrosion of the substrate will occur.

Cathodic protection – A technique to reduce the corrosion rate of a metal by making it the cathode of an electrochemical cell. Thermal spray zinc and aluminum coatings provide this protection to steel substrates, the coating being the anode and the steel being the cathode.

Cavitation – The formation and rapid collapse within a liquid of cavities or bubbles that contain vapor or gas or both.

Cavitation Erosion – A form of erosion causing material to be removed by the action of vapor bubbles in a very turbulent liquid.

Column Loss – The value of the head loss, expressed in feet or meters, caused by the flow friction in the column pipe.

Corrosion – Chemical or electrochemical reaction between a material and its environment, which results in deterioration in the properties of the material.

Corrosion fatigue – The process in which a metal fractures prematurely under conditions of simultaneous corrosion and repeated cyclic loading at lower stress levels or fewer cycles than would be required in the absence of the corrosive environment.

Corrosive wear – Wear in which chemical or electrochemical reaction with the environment is significant.

Crevice corrosion – Localized corrosion of a metal surface at, or immediately adjacent to, an area that is shielded from the full exposure to the environment because of close proximity between the metal and the surface of another material.

Datum or Grade – The horizontal plane that serves as the reference for head measurements. For vertical pumps, this is usually also the elevation of that surface from which the weight of the pump is supported, normally the elevation of the underside of the discharge head or head base plate or sole plate. For horizontal pumps, the datum is usually at the centerline of the pump shaft.

Discharge Head Loss – The value of the head loss expressed in feet or meters, caused by the flow friction in the discharge head assembly.

Drawdown – The difference in feet or meters between the static liquid level in the condenser hot well and the liquid level when pumping at required capacity.

Driver Efficiency – The ratio of the driver power output to the driver power input, expressed in percent.

Driver Power Input – The power input to the driver, expressed in horsepower (hp) or kilowatts (kW).

Driver Power Output – The power delivered to the pump shaft by the driver, expressed in horsepower (hp) or kilowatts (kW) [also referred to as pump input power or brake horsepower (bhp)].

Erosion – Removal of material from a surface due to mechanical interaction between that surface and a fluid, a multicomponent fluid, or impinging liquid or solid particles.

Erosion Corrosion – Associated action involving corrosion and erosion in the presence of a corrosive substance.

Fatigue – A cumulative effect causing a material to fail after repeated applications of stress, none of which exceeds the ultimate tensile strength. The fatigue strength (or fatigue limits) is the stress that will cause failure after a specified number cycles.

Fatigue wear – Wear of a solid surface caused by fracture arising from material fatigue.

Flame hardening – The localized surface heating of a medium carbon steel by an impinging gas flame so that the temperature is raised above 900ºC. The part is quenched (or self-quenches by virtue of the remaining cool bulk of the component) and tempered to produce a hard martensitic structure at the surface.

Flame spraying – A thermal spraying process in which the particles are heated and accelerated in a flame (combustion flame, plasma flame). Old term for thermal spray process.

Fretting – Small amplitude oscillatory motion, usually tangential, between two solid surfaces in contact.

Fretting corrosion – A form of fretting wear in which corrosion plays a significant role.

Fretting wear – Wear arising as a result of fretting (see fretting).

Friction – The reaction force resulting from surface interaction and adhesion during sliding. The friction coefficient is defined as the friction force divided by the load.

Friction Head (h_f) – The head required to overcome the frictional resistance of a piping system to flow of the liquid. It is dependent upon the size and type of pipe, flow rate, and nature of the liquid.

Fused coatings – A process in which the coating material is deposited by thermal spraying and then fused by post heat treatment. This can be done by flame, induction heating, furnace, or laser.

Galling – Damage to the surfaces of materials sliding in contact with each other, usually caused by the localized welding together of high spots. Common for materials like stainless steel, aluminum alloys, and titanium.

Galvanic corrosion – Accelerated corrosion of a metal because of an electrical contact with a more noble metal or nonmetallic conductor in a corrosive electrolyte.

Grit blasting – A pressurized stream of hard metal or oxide grit material used to clean and roughen surfaces prior to coating.

Hard Chromium plating – The electrolytic deposition of chromium to form a very hard (1000Hv), tough coating with good wear resistance. The structure is micro-cracked.

Hardfacing – The application of a cladding or coating of material designed to resist wear.

Head – The value used to express the energy content of the liquid per unit weight of the liquid, referenced to an arbitrary datum. In terms of foot-pounds or meter-kilograms of energy per pound or kilogram of liquid being pumped, all head quantities have the dimension of feet or meters of liquid.

Head Above Datum (H_a) – The head measure above the datum, expressed in feet or meters of liquid. Plus the velocity head at the point of measurement.

Head Below Datum (H_b) – The vertical distance in feet or meters between the datum and the pumping water level.

Hydrogen embrittlement – Hydrogen-induced cracking or severe loss of ductility caused by the presence of hydrogen in the metal. Hydrogen absorption can occur during electroplating, pickling, etc. (The use of hydrogen as a secondary gas in plasma spraying does not appear to affect substrates and the majority of coatings, one exception being titanium coatings.)

Impeller balancing – Correction of residual unbalance by removing or adding weight. Can be done by single-plane balancing, or can require two plane balancing, depending on the geometry of the impeller.

Internal recirculation – The reversal of a portion of the pump flow back through the impeller at low flows. This recirculation is known as suction recirculation when it occurs at the inlet of the impeller and discharge recirculation when it occurs at the impeller outlet. Internal recirculation can produce cavitation damage to the impeller.

Line Shaft Loss – The power, expressed in horsepower (hp) or kilowatts (kW), required due to the rotational friction of the line shaft. This value is added to the bowl assembly input power to predict the pump input power.

Maximum Allowable Working Pressure – The highest pressure at the specified pumping temperature for which the pump casing is designed. This pressure must be equal to or greater than the maximum discharge pressure. For some pumps (such as double suction, vertical turbine, or multi-stage), the maximum allowable casing working pressure on the suction side might be different from that on the discharge side.

Net Positive Suction Head (NPSH) – The total suction head in feet or meters of liquid absolute determined at the suction nozzle and the referred datum minus the vapor pressure of the liquid in feet or meters absolute. The NPSH available at the pump suction connection at a given operating condition is denoted by NPSHa, and the NPSH required by the pump design to prevent cavitation is denoted by NPSHr. (See Appendix B for additional discussion).

Normal conditions – The point on the rating curve at which the pump normally operates. It can be the same as the rated condition point, but it might be different because of the design head and flow margins included in the pump specifications.

Overall Efficiency (wire to water) – The ratio of the energy imparted to the water by the pump to the energy supplied to the drive motor; the efficiency of the complete pump and motor assembly. Overall efficiency is equal to the total pump efficiency multiplied by the motor efficiency.

Peening – Blasting process using spherical shaped beads or shot for cleaning and/or modifying surface properties.

Pitting corrosion – Corrosion of a metal surface, confined to a point or small area, which takes the form of cavities.

Plasma spraying – A thermal spraying process in which the heat source is a plasma flame.

Power – Expressed in units of horsepower (hp) or kilowatts (kW). One horsepower is equivalent to 33,000 foot-pounds per minute, or 2,545 btu per hour, or 0.746 kW (746 watts).

Pump Input Power – The power delivered to the pump shaft by the driver, expressed in horsepower (hp) or kilowatts (kW). Also referred to as brake horsepower (bhp) or driver power output.

Pump Output Power – The power imparted to the liquid by the pump, expressed in horsepower (hp) or kilowatts (kW). Also referred to as water horsepower.

Pumping Liquid Level – The vertical distance in feet or meters from the datum to the level of the suction water reservoir surface while the specified fluid flow is being drawn by the pump. For condensate pumps, the suction water reservoir is the condenser hot well.

Rated or specified conditions – The specified capacity, head, net positive suction head, and speed of the pump.

Sacrificial coating – A coating that provides corrosion protection wherein the coating material corrodes in preference to the substrate, thereby protecting the latter from corrosion.

Setting – The vertical distance in feet or meters from the datum or grade to the centerline of the first-stage suction.

Shot peening – The bombardment of a component surface with steel or ceramic shot. Produces a residual compressive stress in the surface and improves fatigue and stress corrosion performance.

Shutoff – The condition where the pump is primed and running but delivering no flow.

Spalling – The lifting or detachment of a coating from the substrate.

Specific Speed (S, NS, or Ns) – Defined as the speed in revolutions per minute at which a geometrically similar pump would operate in order to deliver 1 gallon per minute at a total head of 1 foot. Specific speed represents a correlation of pump capacity, head, and speed at optimum efficiency and is indicative of the shape and characteristics of the impeller. Specific speed numbers range from approximately 500 (representing radial flow impellers) to approximately 20,000 (representing axial flow impellers):

$$Ns = \frac{\text{Pump Speed (rpm) x [Capacity (gpm)]}^{0.5}}{\text{[head (feet)]}^{0.75}}$$

Speed – The rate of rotation of the pump shaft, expressed in revolutions per minute (rpm) or revolutions per second (rps or hertz).

Spray-fused coatings – A process in which the coating material is deposited by flame spraying and then fused into the substrate by the addition of further heat. This can be applied by flame, induction heating, or by laser.

Static Head – The vertical distance in feet or meters between the centerline of the pump discharge connection and the surface of the liquid in the discharge vessel, and between the centerline of the pump suction connection and the liquid surface of the suction vessel.

Static Liquid Level – The vertical distance in feet or meters from the datum or grade to the liquid level of the suction water reservoir surface under no-flow conditions.

Strain – A measure of the extent to which a body is deformed when it is subjected to a stress.

Stress – The force per unit area on a body that tends to cause it to deform. It is a measure of the internal forces in a body between particles of the material of which it consists as they resist separation, compression, or sliding.

Stress corrosion cracking – A cracking process that requires the simultaneous action of a corrodent and sustained tensile stress.

Submerged suction – A submerged suction exists when the centerline of the pump inlet is below tile level of the liquid in the suction reservoir.

Submergence – For vertical pumps, the distance from the liquid level in the suction reservoir to the lip of the suction bell.

Substrate – The parent or base material to which the coating is applied.

Suction Specific Speed (NSS or Nss) – An index number descriptive of the suction characteristics of a pump.

$$N_S = \frac{\text{Pump Speed (rpm) x [Capacity (gpm)]}^{0.5}}{\text{[NPSHr (feet)]}^{0.75}}$$

The numerical value of suction specific speed is primarily a function of the impeller inlet design. Lower numerical values are associated with better NPSH capabilities.

Surface preparation – Cleaning and roughening the surface to be sprayed, usually by grit blasting. This is to increase the adhesion of the coating to the substrate.

Thermal spraying – A process in which coating material is heated and accelerated from a spray torch toward the workpiece. The deposited material forms a coating on the surface.

Total Discharge Head (h_d) – The algebraic sum of the pressure in feet or meters of liquid measured at the pump discharge and the velocity head at that point.

Total Head (H); Total Dynamic Head (TDH); Total Developed Head (TDH) – The measure of energy increase per unit weight of the liquid, imparted to the liquid by the pump; the difference between the total discharge head and the total suction head.

Total Suction Head (h_s) – The algebraic sum of the pressure in feet or meters of liquid measured at the pump suction and the velocity head at that point.

Total Pump Efficiency (water to water) – The efficiency of the complete pump minus the driver, with all pump losses taken into account as follows:

$$\text{Efficiency} = \frac{\text{Specified Pump Head (feet) x Capacity (gpm)}}{\text{3960 x Brake Horsepower}}$$

Total Pump Length (TPL) – For a vertical pump, the distance from the bottom of the discharge head at grade to the bottom of the lowest part of the pump (including strainer if required). The sole plate, if used, is considered part of the foundation; the measurement is to the top of the sole plate.

Velocity Head (H_v) – The energy of a liquid as a result of its motion at a specific velocity; the equivalent head in feet through which the water would have to fall to acquire the same velocity. In other words, the head necessary to accelerate the water:

$$H_v = \frac{v^2}{2g}$$

where:

g	=	acceleration of gravity
v	=	velocity of the liquid

The velocity head is usually insignificant and can be ignored in most high head systems but might be significant in low-head systems.

A.2 SI Unit Conversion Factors for Units Used in This Report

1 gallon = 3.78 liters

1 inch = 2.54 cm

1 lb = .45 kg

1°F = (°C x 9/5) + 32

1 gpm = 0.063 liters/second

1 psi = 6.89 kPa

1 psig = 6.89 kPa

1 ft^3 = 28.3 liters

1 ft^2= 929 cm^2

1 fpm = 0.51 cm/sec

1 foot = 0.3 meters

1 foot pound = 0.138 kg-meters

1 horsepower = 746 watts

1 ounce-inch = 28.35 gram-inch
28.35 gram-inches = 72.009 g-cm

B
EXPLANATION OF NPSH AND THE EFFECTS OF SPECIFIC GRAVITY AND VISCOSITY

The term *NPSH* is simple but denotes a complex value that is all-important in the successful operation of a centrifugal pump.

By accepted definition, NPSH means *net positive suction head* above the vapor pressure of the pumped liquid, available at the pump suction flange, and referred to the centerline of the impeller. It should always be given in feet of pumped liquid. It is actually a measure of the amount of energy available in the pumped liquid to produce the required absolute entrance velocity into the eye of the first-stage impeller. If a particular pump requires more energy (or NPSH) than is available at a given capacity, the pressure of the impeller inlet vane tips will fall below the vapor pressure of the pumped liquid and cavitation will result.

B.1 Effects of Specific Gravity

The capacity and total head in feet of liquid developed by a centrifugal pump are fixed for every point on the curve and are always the same for the same speed. Neither capacity nor total head will be affected by a change in the specific gravity of the liquid being pumped. However, because the developed pressure is psig (pounds per square inch) and the brake horsepower to drive the pump is a function of the specific gravity of the liquid, both will be affected in direct proportions by any change in specific gravity. Therefore, a change in specific gravity will affect the discharge pressure and is dangerous in that it might overload the pump's driver.

B.2 Effects of Viscosity

The pump is designed to deliver rated capacity at rated head for a liquid with a particular viscosity. When contemplating operation at some viscosity other than for which the pump was originally designed, the changed conditions should be evaluated.

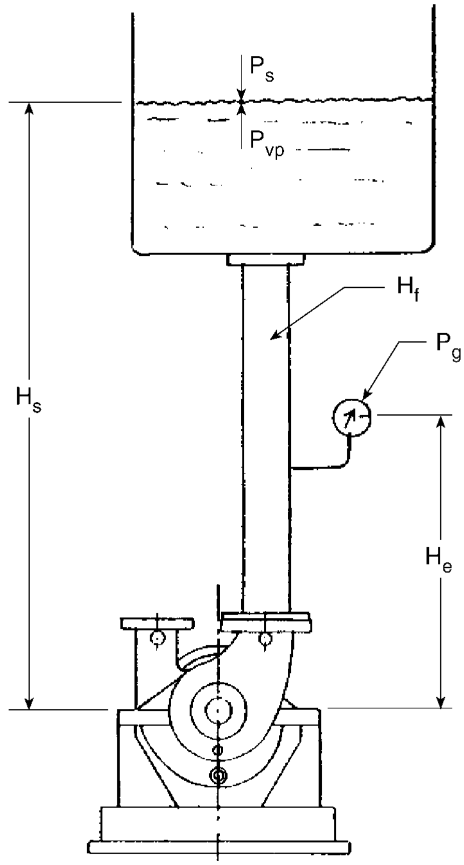


Figure B-1
Simple Pumping System Schematic

In the figure:

P_s = Pressure in the suction vessel (or atmospheric pressure if the vessel is open) in pounds/sq. inch absolute.

P_{vp} = Vapor pressure of pumped liquid at the pumping temperature in psia.

H_s = Static elevation in feet of the liquid surface above (+) or below (-) the centerline of the impeller.

H_f = Friction loss in the suction line from the vessel to the pump suction flange. Entrance loss into the suction line, should be included.

Then:

NPSH= 2.31 (P_s – P_{vp}) Sp. Gravity + H_s-H_f

Sp. gravity is that of the pumped liquid at the pumping temperature.

It can readily be seen from the figure that if the liquid is boiling, P_s and P_{vp} are equal and the first term in the formula becomes zero.

If the liquid surface is below the centerline of the impeller, the value H_s becomes negative and must be subtracted from the first term. For an open vessel, P_s is atmospheric pressure at the elevation of the installation.

If a gage reading is obtained in the suction line:

$$\text{NPSH available (ft. of liquid)} = \frac{2.31\ (P_g + P_a - P_{vp})}{\text{Sp. Gravity}} + H_e + H_v - H_f$$

P_a = Atmospheric pressure
H_f = Friction loss from gage connection to pump suction
H_v = Velocity head at point of gage connection (varies with capacity)
P_g = Gage pressure, psig

If P_g is negative and the connecting hose is empty, as it should be for a vacuum reading, H_e is measured from the point at which the connection is made to the suction line.

Examples:

1. Pump water at 160°F from an open vessel with liquid level 12 feet above impeller centerline, and losses in suction line of 2 feet. Elevation of installation is 5000 ft. above sea level.

   Then P_S = 12.22 psia (std. atmospheric pressure at 5000 ft. elevation).
   P_{vp}= 4.74 psia (vapor pressure of water at 160°F)
   Sp. Gr. of water at 160°F = 0.979. (See Figure B-2 for specific gravity of water)

   $$\text{NPSH available} = \frac{2.31\ (12.22 - 4.74)}{0.979} + 12 - 2 = 27{,}65 \text{ ft.}$$

2. Pump water at 100°F from an open vessel, water surface 10 feet below impeller centerline, and losses in suction line 1.5 feet. Installation at sea level.

   $$\text{NPSH available} = \frac{2.31\ (14.7 - .95)}{0.995} + (-10) - 1.5 = 20.4 \text{ ft.}$$

3. Pump crude oil at 60°F from an open tank with liquid level 5 feet above impeller centerline, vapor pressures 1.0 psia, and suction line losses of 8 feet. Specific gravity = 0.86. Installation at sea level.

   $$\text{NPSH available} = \frac{2.31\ (14.7 - 1.0)}{0.86} + 5 - 8 = 33.8 \text{ ft.}$$

4. Pump propane at 100°F from a tank under pressure, 8 feet of liquid level above impeller centerline, and suction line losses of 2 feet.

   P_S = 170 psia P_{vp}= 170 psia NPSH available = 0 + 8 - 2 = 6 ft.

5. Example, using gage reading on suction line. Pump water at 310°F, P_9 =70 psig, elevation 4000 ft., H_e=5 ft., H_f (between gage connection and pump flange) = 0.1 feet, capacity 1000 gpm, 8 in. line.

   Then: P_a = 12.69 psia Sp. gr. = 0.914
   P_{vp} = 77.68 psia $H_v = (6.28)^2 / 64.4 =$ 0.61 ft.

   $$\text{NPSH available} = \frac{2.31\ (70 + 12.69 - 77.68)}{0.914} + 5 + 0.61 - 0.1 = 18.17 \text{ ft.}$$

Temp. °F	Sp. Gr.	Temp. °F	Sp. Gr.	Temp. °F	Sp. Gr.	Temp. °F	Sp. Gr.
210	.961	290	.925	370	.880	450	.826
2	.961	2	.924	2	.879		
4	.960	4	.923	4	.878	460	.818
6	.959	6	.922	6	.877		
8	.958	8	.921	8	.875	470	.810
220	.957	300	.920	380	.874	480	.802
2	.956	2	.919	2	.873		
4	.955	4	.918	4	.872	490	.794
6	.955	6	.917	6	.870		
8	.954	8	.916	8	.869	500	.786
230	.953	310	.914	390	.867	510	.775
2	.952	2	.913	2	.866		
4	.951	4	.912	4	.865	520	.767
6	.950	6	.911	6	.863		
8	.949	8	.910	8	.862	530	.757
240	.948	320	.909	400	.861	540	.746
2	.948	2	.908	2	.860		
4	.947	4	.907	4	.858	550	.737
6	.946	6	.906	6	.857		
8	.945	8	.905	8	.856	560	.725
250	.944	330	.904	410	.855	570	.716
2	.943	2	.903	2	.853		
4	.942	4	.901	4	.852	580	.704
6	.941	6	.900	6	.850		
8	.940	8	.899	8	.849	590	.692
260	.939	340	.898	420	.847	600	.680
2	.938	2	.897	2	.846		
4	.937	4	.896	4	.845	610	.666
6	.937	6	.895	6	.843		
8	.936	8	.893	8	.842	620	.650
270	.935	350	.892	430	.840	630	.634
2	.934	2	.891	2	.839		
4	.933	4	.889	4	.838	640	.618
6	.932	6	.888	6	.836		
8	.931	8	.887	8	.835	650	.599
280	.930	360	.886	440	.833	660	.578
2	.929	2	.885	2	.832		
4	.928	4	.884	4	.830	670	.554
6	.927	6	.883	6	.829		
8	.926	8	.881	8	.828	680	.526

Figure B-2
Specific Gravity of Water

C
CALCULATION OF PUMPING COST

The term *efficiency*, as used in pumping, would be of no practical value if it could not be reduced to terms of actual pumping costs, expressed in dollars.

With the efficiency of the pump and motor known, proportionate cost of power can be predetermined on a basis common to all pumps, regardless of size or capacity. By using units of capacity and head, comparisons can be made in pumps having different capacities.

The power cost of pumping varies inversely with overall plant efficiency (O.P.E.). Thus, the power cost per gallon for each foot of developed head on a pump of 30% overall plant efficiency is double that of a pump with 60% overall plant efficiency (assuming power rate is the same in both cases).

To pump one gallon of water in one minute (1 gpm) against one foot of head with 100% overall plant efficiency, requires 0.000189 kilowatts. Pumping 1000 gallons per minute (1000 gpm) per foot of head at 100% overall plant efficiency requires 0.189 kilowatts.

The following formulas can be used for determining power coats of pumping under any conditions:

(a) Cost per 1000 gallons (not gpm) for each foot of head $= \dfrac{0.00315 \times R}{\text{O.P.E.}}$

or

$= \dfrac{0.189 \times R}{\text{P.E.} \times \text{M.E.} \times 60}$

Where: 0.189 = Theoretical kW as stated above
R = Power cost per kW-hr. (KWH)
P.E. = Pump efficiency
M.E. = Motor efficiency
O.P.E. = Overall plant efficiency
60 = Minutes

Example: Find the cost per 1000 gallons (not gpm) per foot of head, in a pumping plant whose overall plant efficiency is 60% (0.60), where the power rate is nine mils ($0.009) per kW-hr. Substituting in formula (a):

Cost per 1000 gallons per foot head $= \dfrac{0.0315 \times 0.009}{0.60} = \dfrac{0.00002835}{0.60} = \$0.00004725$

If the pump is lifting water over a 120 foot head, then the cost per 1000 gallons (not gpm) delivered would be equal to $0.00004725 x 120 = $0.00567.

Pumping costs per any given condition of capacity or head can be determined by using the following formula:

(b) Cost per hour = $0.000189 x gpm x total head x power rate overall plant efficiency

Example: Find the cost per hour of a pump delivering 500 gpm over a 120 foot bend, overall plant efficiency of pump 60% (0.60), power rate nine mils ($0.009) per kW-hr

Substituting in formula (b) we have:

$$\text{Cost per hour} = \frac{\$0.000189 \times 500 \times 120 \times 0.009}{0.60}$$

Cost per hour = 17 cents

D
CALCULATION OF MOTOR BRAKE HORSEPOWER

B. H. P. for A. C. motor = $\dfrac{\text{Volts x Amperes x Cos @ x } \bar{\ } \text{n x m}}{746}$

B. H. P. for D. motor =Volts x Amperes x 1. 34 x m

n = Number of phases
Cos @ = Power factor of motor
m = Motor efficiency

Motor speed RPM = $\dfrac{\text{Frequency x 120}}{\text{number of poles}}$
(No load)

Slip Frequency is close to zero at no load and increases with load to the point where motor speed equals nameplate speed when the motor reaches rated load.

Number of Poles	**Frequency (Cyles)**		
	25	**50**	**60**
2	1500	3000	3600
4	750	1500	1800
6	500	1000	1200
8	375	750	900
10	302	600	720
12	250	500	600
14	214	428	514
16	188	375	450

E
PUMP AFFINITY LAWS

The affinity laws express the mathematical relationship between the several variables involved in pump performance. They apply to all types of centrifugal and axial flow pumps. They are as follows:

1. With impeller diameter, D, held constant:

A. $$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

B. $$\frac{H_1}{H_2} = \left[\frac{N_1}{N_2}\right]^2$$

C. $$\frac{BHP_1}{BHP_2} = \left[\frac{N_1}{N_2}\right]^3$$

Where:

Q = Capacity, GPM

H = Total Head, Feet

BHP = Brake Horsepower

N = Pump Speed, RPM

2. With speed, N, held constant:

A. $$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

B. $$\frac{H_1}{H_2} = \left[\frac{D_1}{D_2}\right]^2$$

C. $$\frac{BHP_1}{BHP_2} = \left[\frac{D_1}{D_2}\right]^3$$

When the performance (Q_1, H_1, and BHP_1) is known at some particular speed (N_1) or diameter (D_1), the formulas can be used to estimate the performance (Q_2, H_2, and BHP_2) at some other speed (N_2) or diameter (D_2). The efficiency remains nearly constant for speed changes and for small changes in impeller diameter.

You might occasionally need to estimate the hydraulic performance of a centrifugal pump at a speed or impeller diameter not shown on the catalog performance curves. To approximate the performance of the pump, the affinity laws should be used as follows:

The pump capacity, or flow rate, *Q*, varies with the ratio of RPM:

$$Q_2 = Q_1 \text{ x } N_2 / N_1$$

The pump head, *H,* varies with the ratio of RPM^2:

$$H_2 = H_1 \text{ x } (N_2 / N_1)^2$$

The pump brake horsepower, *BHP,* varies with the ratio of RPM^3:

$$BHP_2 = BHP_1 \text{ x } (N_2 / N_1)^3$$

Example: A customer has an application where he prefers to use a v-belt drive with a maximum diameter impeller, instead of operating at a standard motor speed and trimming the impeller diameter. You have a catalog curve giving the operating characteristics of the pump at 1750 RPM. You want to determine the pump curve at 1535 RPM. The calculations are as follows:

$1535/1750 = 0.8771$

$(1535/1750)^2 = 0.7694$

$(1535/1750)^3 = 0.6749$

Data from 1750 RPM Curve				**Calculated Data for 1535 RPM**		
Q(GPM)	H(Ft.)	BHP	Eff.	Q x .8771 Q(GPM)	H x .7694 H(Ft.)	BHP x .6749 BHP
0	150	9.0	0%	0	115	6.1
100	135	9.7	35%	88	104	6.5
200	120	11.0	55%	175	92	7.4
300	105	12.1	66%	263	81	8.2
400	80	15.2	53%	351	62	10.3

F

APPROXIMATE COMPARISON OF VARIOUS HARDNESS SCALES

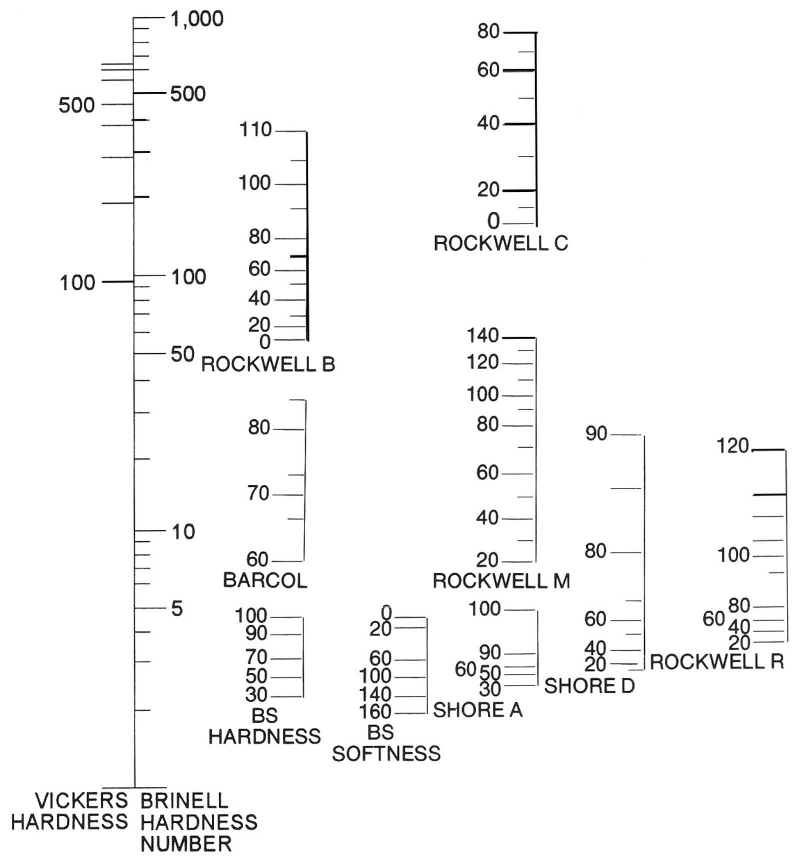


Figure F-1
Hardness Scales

To use the scale, extend a horizontal line across the chart at the value of interest.

G
VERTICAL PUMP TROUBLESHOOTING CHART

Table G-1 is reproduced courtesy of the Johnston Pump Company.

Table G-1
Vertical Pump Troubleshooting Chart

TROUBLE INDICATOR	POSSIBLE CAUSES AND RECOMMENDED REMEDIES
1. Insufficient Pressure	A. Speed too slow—Check to assure the turbine receives full steam pressure. Check if driver is directly across the line and receiving full voltage. Check the pump rev/min is consistent with manufacturer's recommendations. B. Improper impeller adjustment—Use an ammeter to check the accuracy of the impeller adjustment. Follow the manufacturers recommended clearance. C. Loose impeller (rarely occurs)—Refit impeller. Change material with the same expansion factor. Overcome by adding keyway to collet mounting. D. Plugged impeller—Pull the pump. Inspect impeller and bowl passages. E. Wear rings worn—Inspect and replace. F. Entrained air in pump—Clean screen and trash racks. Check return line size and location in sump or tank. Increase suction bell diameter so the pumps required submergence is less than the actual submergence available. G. Leaking column joints or bowl castings—Pull pump and inspect. H. Wrong rotation—Check CCW rotation when viewed from above. Check engagement of motor coupling.
2. Insufficient Capacity	A. Speed too slow—See 1A. B. Improper impeller adjustment—See step 1B. C. Impeller loose (rarely occurs)—See step 1C. D. Impeller or bowl partially plugged—Pull pump and inspect for obstruction. E. Leaking joints—See 1G. F. Strainer partially clogged—See 3D. G. Suction valve throttled—Check and open valve. H. Low Water Level (Deep well)—Increase column length and add a stage or different bowl assembly. I. Wrong rotation—See step 1H.
3. No Liquid Delivered	A. Pump suction broken (water level below bell inlet)—Check for adequate submergence. B. Suction valve closed—Open valve. C. Impeller plugged—Pull pump and inspect for obstruction. D. Strainer clogged—Backflush. Install finer mesh strainer or larger strainer. Pull pump and remove obstruction. E. Wrong rotation—See step 1H. F. Shaft broken—Pull pump and replace shaft. G. Impeller loose—See step 1C.
4. Using Too Much Power	A. Speed to high—Check voltage on motor. B. Improper impeller adjustment—See step 1B. C. Improper impeller trim—Pull pump and inspect. Modify existing impeller to provide actual system requirements. D. Pump out of alignment or shaft bent—Check alignment of pump and driver, also foundations. E. Lubricating oil too heavy—Check quantity and quality of lubricants. Refer to manufacturers recommendations. F. Pumping sand, silt, or foreign material—Test liquid for viscosity and specific gravity.
5. Vibration	A. Motor imbalance—electrical—Carefully observe and analyze the operating history of the pump. Determine cause utilizing vibration frequency analyzer and/or pump disassembly. Consult motor and pump manufacturer. B. Motor bearings not properly seated—See step 5A. C. Motor drive coupling out of balance—See step 5A. D. Misalignment of pump, castings, discharge head, column or bowls—See step 5A. E. Discharge head misaligned by improper mounting or pipe strain—See 5A. F. Bent shafting—See step 5A. Check, straighten or replace. G. Worn pump bearings—See step 5A. Check and replace.

Table G-1 (continued)
Vertical Pump Troubleshooting Chart

TROUBLE INDICATOR	POSSIBLE CAUSES AND RECOMMENDED REMEDIES
5. Vibration	H. Clogged impeller or foreign material in pump—Backflush. Pull, check for damage and replace. I. Improper impeller adjustment—See step 1B. J. Vortex problems in sump—Increase pump submergence. Apply sump modification such as; install vertical splitter; install suction umbrella; relocate pump in sump, lower the inlet velocity in sump. K. Resonance—System frequency at or near pump speed—Loosen anchor bolts. Change pipe loading.
6. Abnormal Noise	A. Motor noise—Check for bearing failure. Install RTD'S in bearing housing. Monitor bearing oil level. B. Broken column bearing retainers—Check and replace. C. Broken shaft or shaft enclosing tube—Check and replace. D. Impellers dragging on bowl case—Re-adjustment impellers per manufacturers recommendations. E. Cavitation due to low submergence or operation beyond maximum capacity rating—Insufficient NPSH available. Evaluate system head conditions and reduce pump capacity, if possible. Change pump impeller to lower NPSH design. Replace bowl assembly with a different model capable of operating with the system NPSH available. F. Foreign material in pump—Backflush or pull and inspect.
7. Seal Leaks Steadily	A. Faces not flat—Check for incorrect installation dimensions. B. Blistered carbon graphite seal faces—Check for gland plate distortion due to over-torquing of gland bolts. Improve cooling flush line, if overheated. Check gland gasket for proper compression. Clean out any foreign particles between seal faces and re-lap faces. Check for cracks and chips at seal faces during installation. Replace primary and mating rings, if damaged. C. Secondary seals nicked or scratched during installation—Replace secondary seals. D. Worn out or damaged O-rings—Check for proper seal selection with seal manufacturer. E. Compression set of secondary seal (hard and brittle)—Check for proper lead-in on chamfers, burrs, etc. F. Chemical attack (soft and sticky)—Review with seal manufacturer for alternate materials selection. G. Spring failures—Replace parts. H. Erosion damage of hardware and/ or corrosion of drive mechanism—Review with seal manufacturer for alternate materials selection.
8. Seal Squeals during Operation.	A. Inadequate amount of liquid to lubricate seal faces—Flush line maybe required. Enlarge flush line and/or orifices in gland plate.
9. Carbon Dust Accumu-lating on Out-side of Gland Ring.	A. See step 8A. B. Liquid film evaporating between seal faces—Check for proper seal design with seal manufacturer if pressure in stuffing box is excessively high.
10. Seal Leaks Intermittently	A. See causes listed under "Seal leaks steadily."—See step 7 A through H. Check for squareness of stuffing box to shaft. Align shaft, impeller and bearing to prevent shaft vibration and/or distortion of gland plate and/or mating ring.
11. Short Seal Life	A. Abrasive particles in fluid—Prevent abrasives from accumulating at seal faces. Flush line maybe required. Use abrasive separator or filter. B. Seal running too hot—Increase cooling of seal faces (for example, by increasing flush line flow). Check for obstructed flow in cooling lines. C. Equipment mechanical misaligned—Align properly. Check for rubbing of seal on shaft.

H

REPAIR/REFURBISHMENT SPECIFICATION FOR APM CIRCULATING WATER PUMP

Specify the following items when specifying a replacement vertical pump:

- Quantity
- Rated capacity per pump, gpm
- Minimum total head at rated capacity, feet
- Maximum acceptable shutoff head, feet
- Maximum speed, rpm
- Elevation, top of pump base, feet
- Elevation, bottom flare of pump suction bell, feet
- Elevation, bottom of sump, feet
- Elevation, maximum water level in sump, feet
- Elevation, minimum water level in sump, feet
- Headroom available for dismantling pump, feet
- Design pressure, psig
- Design temperature, °F
- Impact test not required
- Maximum acceptable pump shaft power, bhp

Notes:

Shutoff head shall be a minimum of 25% greater than indicated total head at rated capacity

Rated total head shall be maximized by the pump design within the constraint of the maximum acceptable pump shaft power.

Contractor shall obtain dimensions of existing pump foundation and discharge piping for each of the 12 pump locations. Dimensions will include flange physical location and flange bolt hole orientation. These dimensions shall be used to fabricate replacement discharge heads that will be interchangeable for all pump locations. Contractor shall provide drawing showing these dimensions.

Pump shafts and line shafts shall be rolled or forged stainless steel conforming to ASTM A479 Grade XM-19 with bearing journal and throttle bushing surfaces hard-faced with tungsten carbide ceramic coating applied using the high-velocity oxygen fuel (HVOF) thermal spray process. Shafts shall be precision ground over their entire length to 63 micro-inch RMS finish or smoother. Shafts shall be straight to total indicated runout (TIR) not greater than 0.005 inch per shaft section.

Line shaft couplings shall be forged stainless steel conforming to ASTM A479 Grade XM-19. Line shaft couplings shall be precision ground throughout and shall be mechanically secured against axial movement in both directions on shafts. Line shaft couplings shall use keys to drive shafts and capture split-rings to join shaft ends.

Pump shaft and line shaft bearings shall be Greene Tweed AR-1 thermoset composite material or proven alternative materials with equal or greater wear resistance and dry-start capability. Bearings shall be press-fit and locked in place with locking screws of Type 316 material. Bearings shall be spaced so that the lateral critical speeds of the rotor are at least 20% greater than the pump rated speed with two times the as-new bearing clearances to prevent vibration during operating life.

The following items should be considered when upgrading an existing vertical pump:

- **Shaft critical speed below or close to running speed.** The following can be modified to affect the critical speed: shaft diameter, column length, bearing spacing, and shaft material.
- **Material changes to wear rings.** Installation of harder material will provide longer service life, but interference fit dimensions will need to be reviewed to determine if high hoop stress is generated. High interference fit can lead to cracked wear rings and pump failure.
- **Shaft coating installation in bearing wear areas can lead to longer shaft life.** Coating installation errors have caused coating to fail, leading to bearing failure. When specifying coating installation, verification of the process and audit of sub-vendor methods need to be considered. The HVOF coating application process results in high-density/low-porosity coating. Process gases—such as hydrogen, oxygen and air—are injected at high pressure into the combustion chamber of a torch and then lit. The flame reaches a supersonic speed, and coating powder is then injected. A dense coating is applied. HVOF is the preferred method for shaft coatings.

Enclosed is an example of a Repair Specification written for a 53 APM circulating water pump with a semi-enclosed impeller and pullout design. Also included is a Bid Evaluation Worksheet that can be used to review vendor proposals for pump repairs.

1. APPLICABLE PUMP(S)

1.1.	Application:	
1.2.	Manufacturer:	Ingersoll-Rand
1.3.	Model Number:	53 APM, 1 Stage
1.4.	Performance:	130,300 gpm, 28 psig, 350 rpm
1.5.	Serial Number(s):	
1.6.	Weight:	Pump – 49,000 lbs Motor – 55,550 lbs

2. RECEIPT INSPECTIONS, DISASSEMBLY, AND CLEANING

Due to disassembly required to remove the pump from the plant, the pump will be shipped with the support head (172), discharge head (361) separate from the pump pull out assembly. Pump should be shipped back to the plant site in the same partially disassembled condition.

2.1. Receipt Inspection
- **2.1.1.** Catalog all loose parts.
- **2.1.2.** List any shipping damage.

2.2. Disassembly
- **2.2.1.** Record total rotor end float.
- **2.2.2.** Record pump shaft stick up relative to casing (1).

2.3. Cleaning
- **2.3.1.** Clean stationary components as required to remove rust or scale and to prepare critical surfaces for non-destructive examination (NDE).
- **2.3.2.** Grit blast rotating components, except shaft, as required to prepare critical surfaces for NDE.
- **2.3.3.** Clean shaft using stainless wire brush.

3. DIMENSIONAL INSPECTIONS AND NDE

3.1. Support Head (Item 172) with Discharge Head Liner (Item 421)
- **3.1.1.** Record diameter of motor, stuffing box extension (264), and discharge head (361) registers. Repair as required to meet fit criteria.
- **3.1.2.** Record perpendicularity and concentricity of motor, and discharge head (361) registers, and inner column fit in discharge head liner (421) relative to the stuffing box extension (264) register. Repair as required.

3.2. Discharge Head Liner (Item 421)
- **3.2.1.** The discharge head liner (421) should be treated as an integral part of the support head. However, if the two must be separated record diameter of support head (172) register. Repair as required to meet fit criteria.
- **3.2.2.** Record perpendicularity and concentricity of support head (172) register relative to the inner column support diameter. Repair as required.

3.3. Stuffing Box Extension (Item 264)

3.3.1. Record ID of stuffing box extension bearing (138A).

3.3.2. Record diameter of support head (172) register and packing area. Repair as required to meet fit criteria.

3.3.3. Record perpendicularity and concentricity of support head (172) register and packing area. Repair as required.

3.3.4. Clean and inspect split gland (16).

3.3.5. Replace inner column o-ring (456)

3.3.6. Replace bearing (12) with rubber cutlass type or customer approved equivalent, meeting running clearance criteria and fit criteria in the stuffing box housing.

3.4. Discharge Head (Item 361)

3.4.1. Record diameter of the support head (172) and outer column (423) registers. Repair as required to meet fit criteria with mating register.

3.4.2. Record perpendicularity and concentricity of the support head (172) and outer column (423) registers. Repair as required.

3.5. Inner Column (Item 424)

3.5.1. Record diameter of casing (1) register. Repair as required to meet fit criteria with mating register.

3.5.2. Clean end of inner column for insertion into the discharge head liner (421).

3.5.3. Prepare surfaces and apply coal tar epoxy.

3.6. Casing (Item 1)

3.6.1. Record ID of the casing bearings (138A & 138B), quantity 2.

3.6.2. Record diameter of inner column (424) and shroud (89) registers. Repair as required to meet fit criteria with mating register.

3.6.3. Record OD of casing fit with outer column (423).

3.6.4. Record perpendicularity and concentricity of inner column (424) and shroud (89) registers relative to the casing bearing centerline. Repair as required.

3.6.5. Perform liquid penetrate examination of casing vanes at inlet to approximately 6 inches back along vane.

3.6.6. Replace bearings (138A & 138B) with rubber cutlass type or customer approved equivalent, meeting running clearance criteria and fit criteria in the casing (1).

3.7. Bearing Support (Item 110)

3.7.1. The bearing support (110) should be treated as an integral part of the casing (1). However, if the two must be separated record diameter of the casing (1) register. Repair as required to meet fit criteria.

3.7.2. Record perpendicularity and concentricity of casing (1) register relative to the bearing centerline. Repair as required.

3.8. Shroud (89)

3.8.1. Record diameter of the casing (1) register. Repair as required to meet fit criteria with mating register.

3.8.2. Inspect taper fit for degradation of contact surface. Repair as required.

3.8.3. Record perpendicularity and concentricity of the casing (1) register relative to taper for outer column (423) fit.

3.9. Impeller (Item 3)

3.9.1. Sand blast impeller and perform NDE for cracks.

3.9.2. Advise on impeller condition. Any impeller welding to repair vane or cracks shall require stress relief of the impeller.

3.9.3. Machine impeller vanes to restore contour with shroud (89). With impeller resting in housing, gap between impeller and shroud along each vane should not exceed .020" clearance

3.9.4. Record impeller bore diameter. Repair as required to meet fit criteria with shaft.

3.9.5. Balance lower shaft (10A) and impeller (3) as an assembly to a maximum residual unbalance criteria of 4W/N.

3.9.6. Clean and inspect impeller lock collar. Replace as required

3.10. Upper Shaft (Item 10B), and Lower Shaft (Item 10A)

3.10.1. Perform magnetic particle or liquid penetrate examination of threaded areas, keyways, and shoulders of the shaft. Visually examine entire shaft for cracks. Report any cracks, degradation or indications.

3.10.2. Remove shaft sleeves (135A, 135B & 135C) and record OD of shaft fit areas.

3.10.3. Record shaft TIR and OD at shaft sleeve, impeller and coupling fit locations. Restore shaft diameters as required.

3.10.4. Inspect shaft sleeve nut (127). Replace as required.

3.10.5. Replace shaft sleeves (135A, 135B & 135C) with ones meeting fit and running clearance criteria. Shaft sleeve OD shall be surfaced hardened with tungsten carbide coatings applied by HVOF process.

3.11. Shaft Coupling (Item 164)

3.11.1. Perform magnetic particle or liquid penetrate examination of shaft coupling. Report and crack indications.

3.11.2. Record ID of shaft fit locations. Repair as required to meet fit and concentricity criteria.

3.11.3. Balance coupling to a maximum residual unbalance criteria of 4W/N.

3.11.4. Inspect split rings (252A & 252B). Replace as required.

4. REPAIR AND ASSEMBLY CRITERIA

4.1. Establish true centerline compatibility between rotor and stator at the motor-to-discharge flange location. Ensure that the maximum relative displacement of the stator to the rotor, at any point for the entire length of the pump, does not exceed the minimum clearance of the closest running fit. This will eliminate non-straightness of the rotor bundle, non-straightness of the stator bundle, rotor imbalance, and rotor-to-stator misalignment. Meeting the acceptance criteria for critical tolerances and following the recommended manufacturing practices should provide a minimum life mean time between repair (MTBR) of 10 years.

4.2. Shaft TIR maximum of .0015”

4.3. Shaft OD to shaft sleeve ID shall have maximum fit clearance 0.002”

4.4. Balance bowl shaft and impeller as an assembly to a maximum residual unbalance criteria of 4W/N or W/N when possible.

4.5. The loss of damping resulting from increased clearances leads to increased vibration, further increasing clearances. To halt this cycle, pay careful attention to critical component fit-ups, concentricity, perpendicularity, and straightness to minimize non-centerline compatibility between the rotor and stator.

4.6. Fit criteria: .0005–.002” clearance for male and female registers. Pad weld alignment registers a minimum of 8 locations to restore unacceptable fits. Head-to-column fit is critical; inspect head closely for cracks and other defects that could affect fit of column to head. Dimensional checks of the head may be required.

4.7. Male and female registers and flanges shall be repaired as required to obtain concentricity, perpendicularity, and parallelism within .0015”

4.8. Impeller shall be repaired or machined as required to remove nominal damage and to obtain fit to shaft of not greater than .002” loose on the diameter. Major impeller repair, if required, may be quoted separate after results of detail inspections.

4.9. Shroud and impeller vane edges shall be machined as required to obtain proper contour. Maximum 0.020” gap between impeller vanes and shroud.

4.10. Bearing ID to journal sleeve OD clearance of .014” to .027” (running clearance).

4.11. Bearing OD fit in housings shall be .000” to .002” clearance. Bearing shall be positively secured in housings. The root cause for life degradation in many vertical multistage pumps is premature opening of radial bearings and/or wear ring clearances.

4.12. Provide all new bolting and fasteners

4.13. Coat wetted non-machined surfaces with coal tar epoxy or customer approved equivalent. Do not coat impeller housing surface, which establishes impeller lift (running clearance) or running position. Prime and paint non-wetted surfaces.

4.14. Assemble the pump, recording critical dimensions and package for shipment.

4.15. Record “as-left” or “as-repaired” condition of all items required to be documented during disassembly and inspection.

4.16. Any criteria not specifically addressed within shall default to manufacturer specifications.

5. WORK SCOPE & PROPOSAL

5.1. The intent of this bid specification is that the dimensional inspections in Section 3.0 be performed as part of the base work scope. Dimensional inspection does not include checking perpendicularity, and concentricity of the parts. After review of the dimensional inspection results a determination will be made to repair or leave the component "as is". In the event of an acceptable dimensional inspection, the option to check the perpendicularity and concentricity will be considered. If it is decided to repair a component, the component shall be required to meet all criteria required by this specification.

5.2. Quote as an option the use of Thordon bearing material as a replacement for rubber bearing material.

5.3. Provide a line item for the base/minimum work scope defined by Sections 2, 3 and 4. Base work scope shall include consumables such as gaskets, & packing.

5.4. Provide line item adders for repair of registers and fits identified in Sections 3 to requirements of Section 4.

5.5. Provide report documenting "as found" and "as left" condition. Record critical dimensions of all parts. Include diameters, perpendicularity, parallelism, concentricity, total indicated runout (TIR) and fit clearances as applicable.

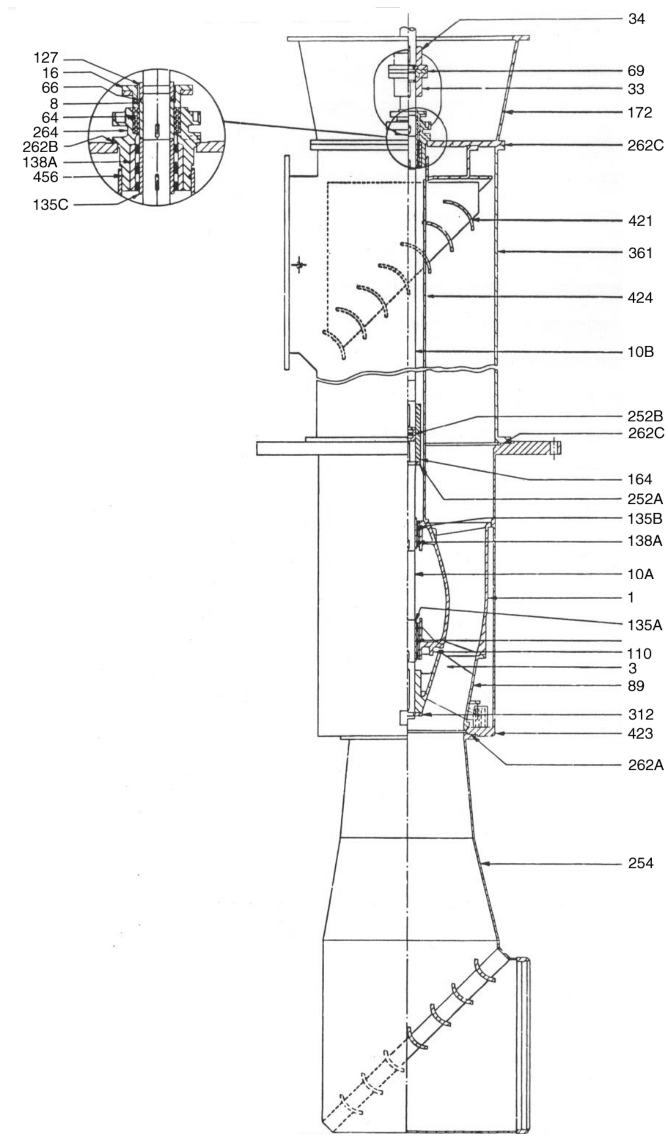
5.6. Provide pricing for the following parts:

Item	Description	Material
8	Stuffing Box Shaft Sleeve	Stainless Steel - AISI Type 410
10A	Lower Shaft	Carbon Steel - ASTM A576, Grade 1020, BHN 140
10B	Upper Shaft	Carbon Steel - ASTM A576, Grade 1020, BHN 140
16	Split Gland	Bronze - ASTM B505 CA932, BHN 60
66	Shaft Packing Sleeve	Stainless Steel - 420(F), HVOF Tungsten Carbide Coated
69	Adjusting Nut	Carbon Steel
127	Shaft Sleeve Nut	Bronze - ASTM B505 CA932, BHN 60
135A	Shaft Journal Sleeve	Stainless Steel - 420(F), HVOF Tungsten Carbide Coated
135B	Shaft Journal Sleeve	Stainless Steel - 420(F), HVOF Tungsten Carbide Coated
135C	Shaft Journal Sleeve	Stainless Steel - 420(F), HVOF Tungsten Carbide Coated
138A	Rubber Cutless Bearing	Steel or Bronze Backed Cutless Rubber
138B	Rubber Cutless Bearing	Steel or Bronze Backed Cutless Rubber
164	Shaft Coupling	Stainless Steel - AISI Type 410
252A	Split Ring	Carbon Steel - ASTM A576, Grade 1020, BHN 140
252B	Split Ring	Carbon Steel - ASTM A576, Grade 1020, BHN 140
312	Lock Collar	Stainless Steel - AISI Type 410 or 416
424	Inner Column	Carbon Steel

5.7. Complete Bid Evaluation Worksheet provided at the end of this specification.

6. Drawings

Item	Description
1	Casing
3	Impeller
8	Stuffing Box Shaft Sleeve
10A	Lower Shaft
10B	Upper Shaft
16	Split Gland
33	Driven Half Coupling
34	Driver Half Coupling
64	Stuffing Box Packing
66	Shaft Sleeve Packing
69	Adjusting Nut
89	Shroud
110	Bearing Support
127	Shaft Sleeve Nut
135A	Bearing Support Journal Sleeve
135B	Casing Journal Sleeve
135C	Stuffing Box Journal Sleeve
138A	Casing & Stuffing Box Bearing
138B	Bearing Support Bearing
164	Shaft Coupling
172	Support Head
262A	Suction Pipe Gasket
262B	Stuffing Box Gasket
262C	Support Head – Outer Column Gasket
264	Stuffing Box Extension
312	Lock Collar
361	Discharge Head
421	Discharge Head
423	Outer Column
424	Inner Column
456	Inner Column O-Ring



CLEARANCES

Bearings (138A/B) to journal sleeve (135A/B/C)	.014" - .027"

TORQUE VALUES

Suction pipe (254) to outer column (423)	392 - 435 Ft. Lbs.
Shroud (89) to casing (1)	279 - 310 Ft. Lbs.
Casing (1) to bearing support (110) capscrews	135 - 150 Ft. Lbs.
Outer column (423) to discharge head (361)	990 - 1100 Ft. Lbs.
Casing (1) to inner column (424)	72 - 80 Ft. Lbs.
Stuffing box extension (264) to support head (172)	391 - 435 Ft. Lbs.
Lock collar (312) to impeller (3) capscrews	176 - 195 Ft. Lbs.
Motor to discharge head (361)	392 - 435 Ft. Lbs.
Split ring (252B) to shaft coupling (164) bolting	22 - 24 Ft. Lbs.

Torque Values listed above are selected to achieve the proper amount of pre-stress in the threaded fastener. Maintenance personnel must insure that threads are in good condition (free of burrs, galling, dirt, etc.) and that commercial thread lubricant is used. Torque should be periodically checked to assure that it is at the recommended value.

APPROXIMATE WEIGHTS

PART	WEIGHT
Support Head (172)	10,000 Lbs.
Discharge Head Liner (421)	3,500 Lbs.
Discharge Head (361)	13,100 Lbs.
Shroud (89)	1,325 Lbs.
Bearing Support (110)	1,025 Lbs.
Casing (1)	11,000 Lbs.
Inner Column (424)	1,060 Lbs.
Lower Shaft (10A)	1,125 Lbs.
Upper Shaft (10B)	2,600 Lbs.
Shaft Coupling (164)	385 Lbs.
Impeller (3)	2,500 Lbs.
Stuffing Box Extension (264)	460 Lbs.
Gland (16)	40 Lbs.
R.A. Coupling (33/34)	740 Lbs.
Miscellaneous	140 Lbs.
Pump (Minus Outer Column And Suction Pipe)	49,000 Lbs.
Water In Pump (Above Casing)	46,200 Lbs.
Motor (Nominal Plus 10% Possible)	55,550 Lbs.
TOTAL	150,750 Lbs.

Bid Evaluation Worksheet	
Disassembly/Inspection/Repair/Assembly (labor)	
Materials & Parts (bearings, bolting, gaskets, consumables, etc.)	
Transportation	
Total for Base Work Scope	
3.1.1 Repair fits and/or perpendicularity & concentricity of Support Head registers	
3.1.2 Record perpendicularity & concentricity of Support Head registers	
3.2.1 Repair fits and/or perpendicularity & concentricity of Discharge Head Liner registers	
3.2.2 Record perpendicularity & concentricity of Discharge Head Liner registers	
3.3.2 Repair fits and/or perpendicularity & concentricity of Stuffing Box registers	
3.3.3 Record perpendicularity & concentricity of casing registers	
3.4.1 Repair fits and/or perpendicularity & concentricity of Discharge Head registers	
3.4.2 Record perpendicularity & concentricity of Discharge Head registers	
3.5.1 Repair fits and/or perpendicularity & concentricity of Inner Column registers	
3.6.2 Repair fits and/or perpendicularity & concentricity of Casing registers	
3.6.4 Record perpendicularity & concentricity of Casing registers	
3.7.1 Repair fits and/or perpendicularity & concentricity of Bearing Support registers	
3.7.2 Record perpendicularity & concentricity of Bearing Support registers	
3.8.1 Repair fits and/or perpendicularity & concentricity of Shroud registers	
3.8.2 Repair taper fit of Shroud into outer column	
3.8.3 Record perpendicularity & concentricity of Shroud registers	
3.9.4 Repair Impeller bore	
3.10.3 Repair shaft fit locations for Shaft Sleeves, Impeller, & Coupling	
3.11.2 Repair Shaft Coupling fits	
Total for 1 Each of the Adders	
Item No. 6 – Stuffing Box Shaft Sleeve	
Item No. 10A – Lower Shaft	
Item No. 10B - Upper Shaft	
Item No. 16 – Split Gland	
Item No. 66 – Shaft Packing Sleeve	
Item No. 69 – Adjusting Nut	
Item No. 127 – Shaft Sleeve Nut	
Item No. 135A – Shaft Journal Sleeve	
Item No. 135B – Shaft Journal Sleeve	
Item No. 135C - Shaft Journal Sleeve	
Item No. 138A – Rubber Cutless Bearing	
Item No. 138B – Rubber Cutless Bearing	
Item No. 164 – Shaft Coupling	
Item No. 252A – Split Ring	
Item No. 252B – Split Ring	
Item No. 312 – Lock Collar	
Total for 1 each of the Parts	
Exceptions or Comments to Bid Specification	

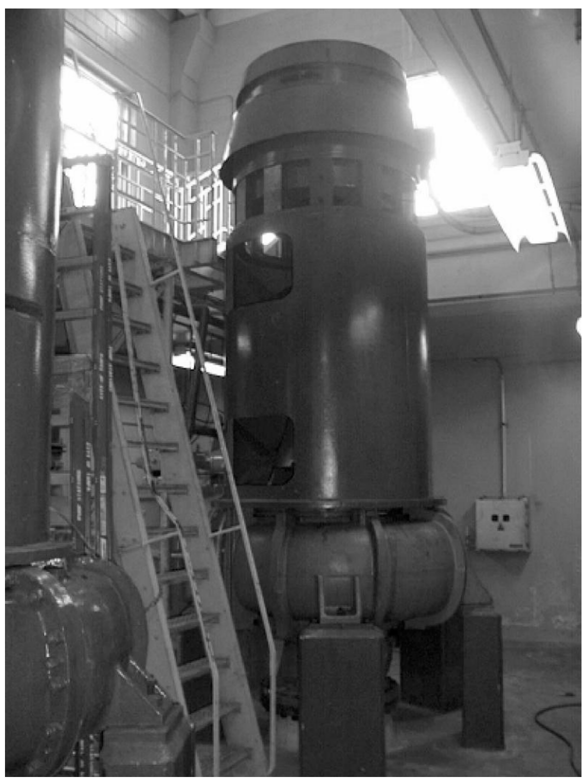
I

VERTICAL PUMP ALIGNMENT USING LASER TECHNOLOGY

The following case study by Alan Luedeking, of Ludeca, Inc., has been provided to illustrate vertical pump laser alignment. The case study has been modified only with regard to commercial-type statements.

I.1 Case Study

A 640HP vertical flange-mounted Fairbanks-Morse condenser pump, considered critical equipment at a nuclear power plant in South Florida, needed to be quickly realigned after the motor was repaired and reinstalled. The alignment had been scheduled for the next day, but was pushed forward and assigned to the weekend night shift when another condenser pump had to be brought down due to excessive vibration readings.

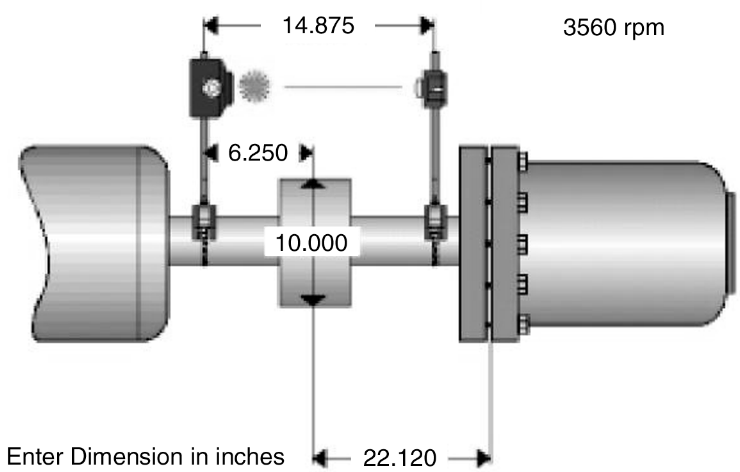


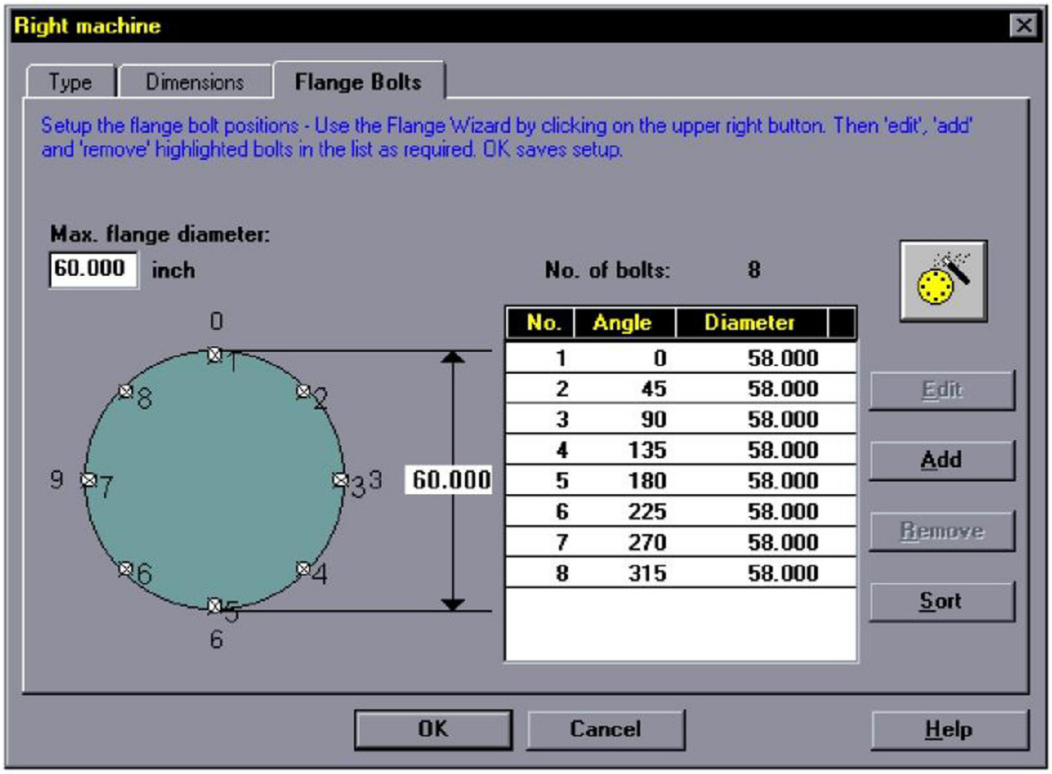
When the ROTALIGN® PRO system normally used by the day crew was found to be locked away in an office and the key could not be found, the alternative would have been to take time-consuming dial indicator readings with subsequent laborious calculations to compute corrections. The excessive downtime required was very undesirable in this case, because only two other condenser pumps were up, and an unexpected failure of either one of them would have required a major reduction in power output. It was decided to try using an OPTALIGN© PLUS system that was available.

After alignment readings were taken and results obtained, a controversy arose over performing the shimming corrections. The situation was as follows.

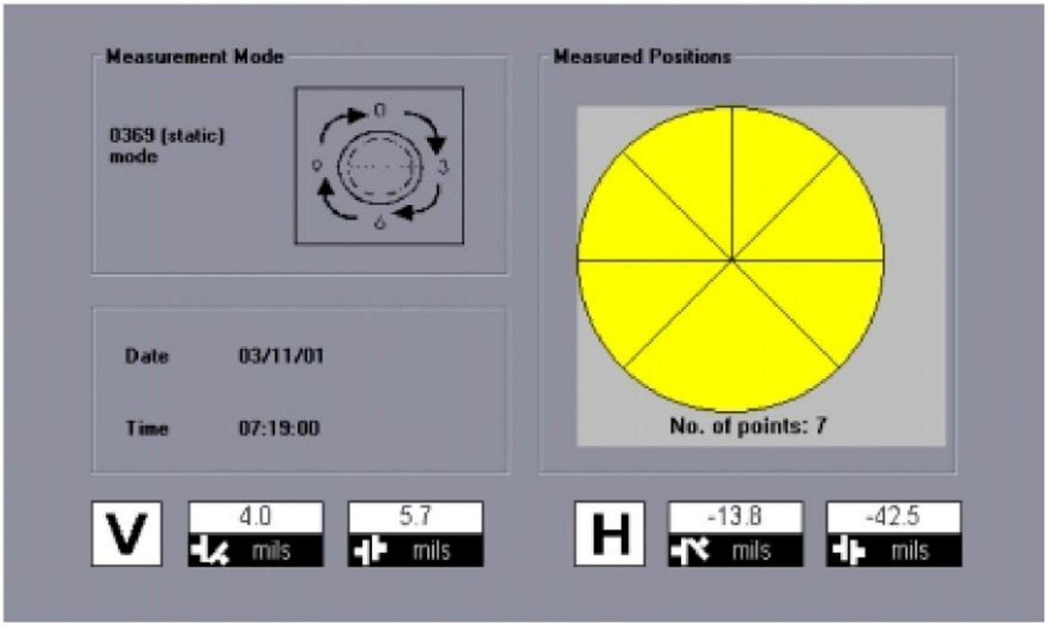
The OPTALIGN© PLUS was set up as usual, with the prism on the motor shaft.

Vertical Machine mode was selected and all necessary dimensions were entered (see below).

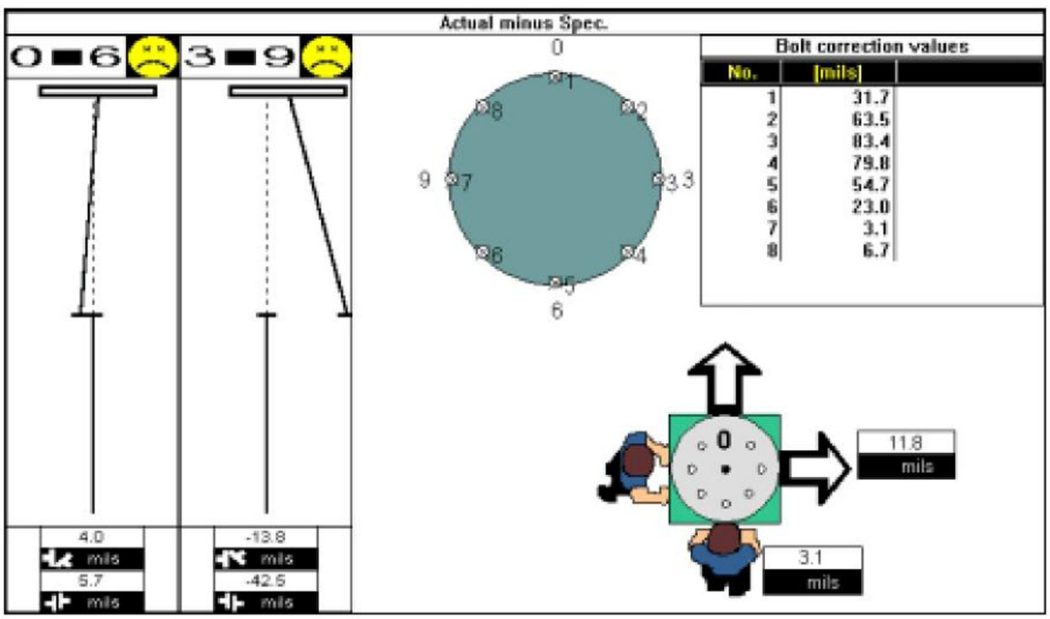




Next, “Static Mode” (0,3,6,9) was selected and readings taken.



Now results were obtained.



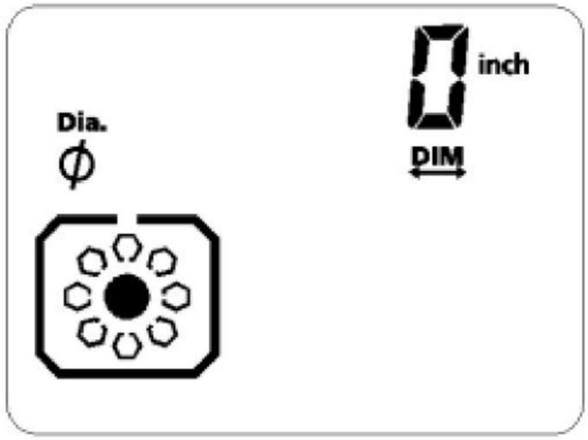
The following picture shows the OPTALIGN© PLUS connected to a laptop computer, with the OPTALIGN© PLUS PC DISPLAY software running. This feature allowed everybody to participate and see every step of the alignment job as it proceeded.

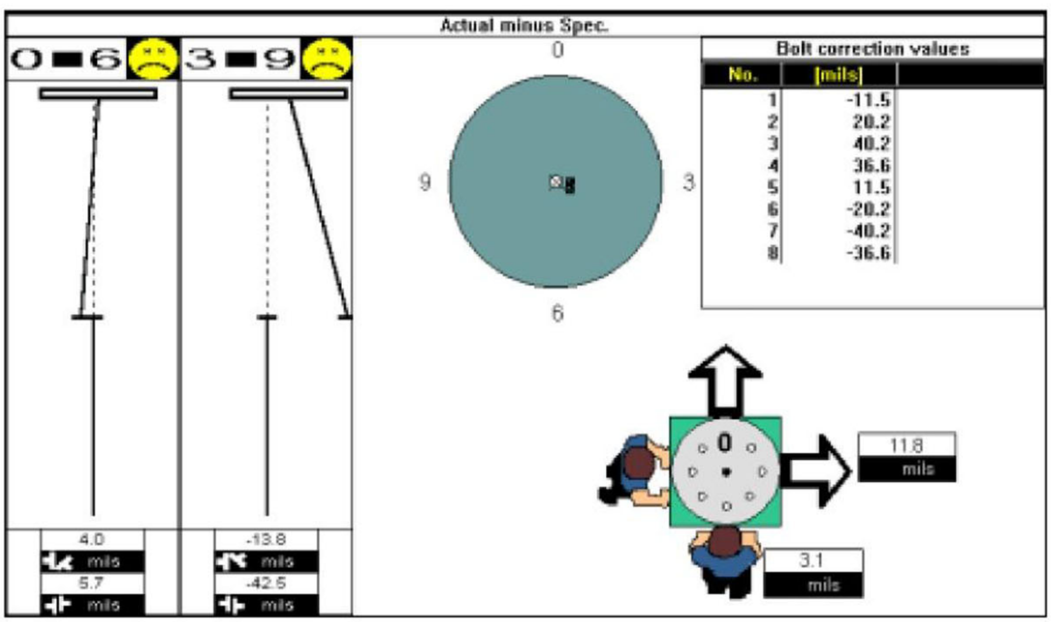
The problem arose when the customer objected to the fact that all shimming results for angularity correction were displayed as positive values, because there was a concern that performing these corrections would result in changing the axial position of the pump impeller, possibly lifting it off of its conical seat at the bottom of the pump pit and imposing an axial load on the motor's bearings. These results were therefore considered unacceptable. Ideally, the customer desired shimming corrections that would not result in any axial movement of the shafts, while still accomplishing the alignment.

The customer was aware that their ROTALIGN® PRO system offered a capability of obtaining flange shimming corrections as half positive and half negative values, in order to avoid axial movement—the *Plus/Minus* solution or *minimized shimming* solution. They were very desirous of having this same capability in the OPTALIGN© Plus. Although not mentioned in the operating manual, this capability exists! The following simple procedure was devised to obtain these results:

1. All dimensions are entered normally, including flange dimensions and bolt positions.
2. Take readings using 0,3,6,9 mode.
3. Obtain results.

4. Now, return to the Dimensions screen and reset the flange diameter to 0!
   Note: It is not possible to enter a flange diameter of 0 beforehand, because the OPTALIGN© PLUS will not allow you to subsequently enter the correct bolt circle diameter for each bolt.
5. Obtain shimming corrections again.



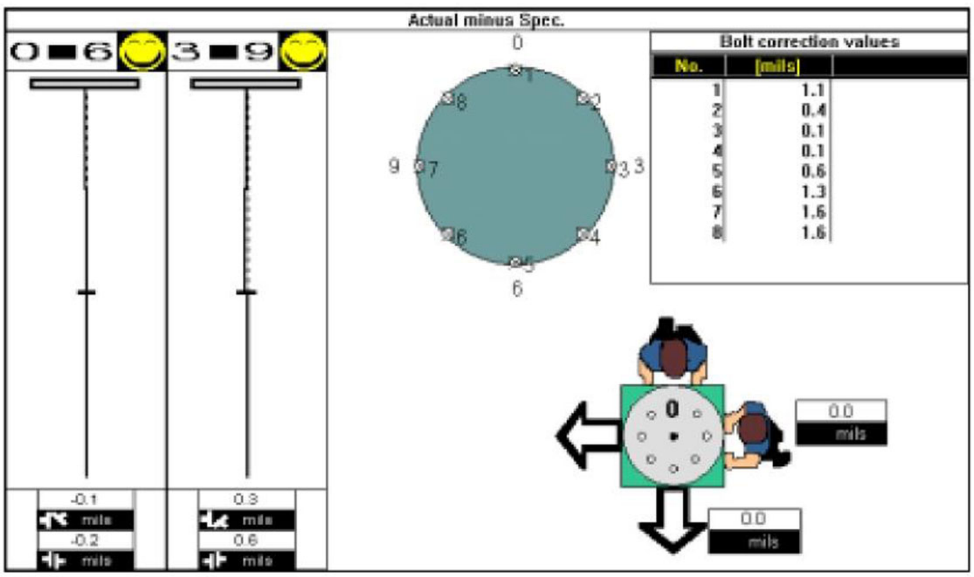


Note: The OPTALIGN© Plus Commander Alignment Editor does not allow you to do this, however, the OPTALIGN© Plus tool does; then the Editor will not display the 8 bolt numbers after you have done this, but all the results still appear normally.

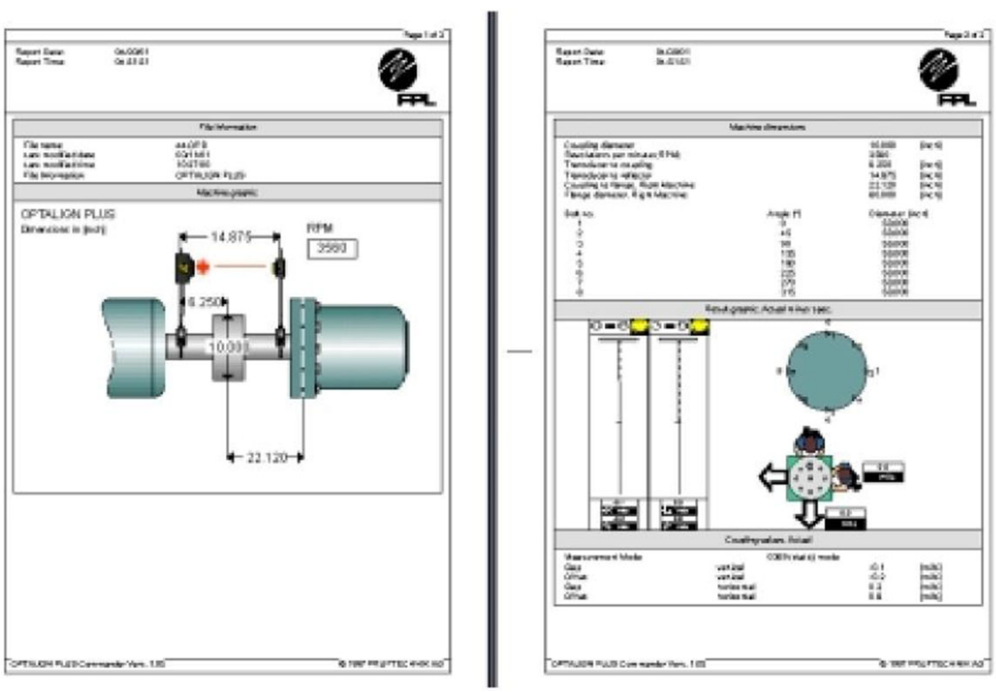
You now have your Plus/Minus shimming corrections!

With this simple workaround the *Plus/Minus* solution was obtained. (It was also mentioned that by resetting the flange diameter to equal the bolt circle diameter, a “0+” solution could also be obtained!)

The motor was successfully and quickly aligned, with an excellent final alignment result attained. The flange diameter was then reset to the correct dimension and final results were recorded and saved.



A color printed report.



J
IMPELLER INSPECTION GUIDELINES

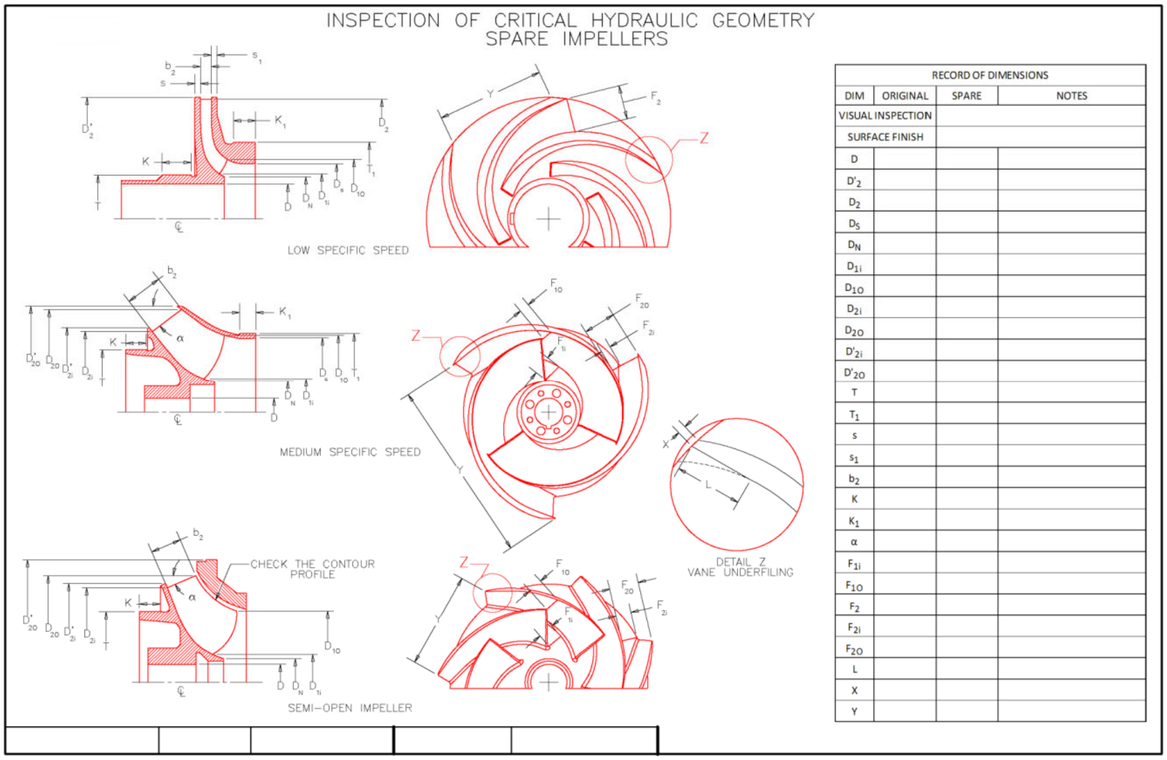


Figure J-1
Impeller Inspection Guidelines
Used with permission from HydroAire, Inc.

K

LISTING OF KEY INFORMATION

The following list provides the location of “Key Point” information in this report.

Key O&M Cost Point

Emphasizes information that will result in reduced purchase, operating, or maintenance costs.

Page Number	Key Point
2-20	Provisions for renewable wear rings might increase the initial cost but can improve the pump refurbishment process during the life of the pump, thereby reducing total maintenance costs.
4-2	A *ring and bushing* job is the least expensive repair cost but can potentially be the most costly total cost when the service life of the pump is reduced, or the pump catastrophically fails on startup or in service and results in a plant trip.
4-3	Based on years of developing specifications and monitoring the repairs in progress, it is possible to write a repair specification that defines a base work scope, and to be able to handle the unknowns that are uncovered during the repair process.
4-3	If there is something learned during a repair, revise the specification to capture that information.
4-7	The casing vane NDE recommendation is intended primarily for cast iron casings. Cast iron casings crack in this area due to stress cracks (sometimes microscopic) that are induced during manufacturing and that propagate over time. In most cases, the casings can be salvaged by grinding out the crack and reshaping the inlet vane. Cracks extending as much as 5 inches along the vane have been removed without affecting pump performance.
4-13	Coatings are usually applied to pumps used in environments such as river water, lakes, circulating water, and so on. Coatings are usually not applied in condensate and heater drain applications.
4-22	Except for severe cases, repair instead of replacement is the most economical solution.

Key Technical Point

Targets information that will lead to improved equipment reliability.

Page Number	Key Point
2-7	Adequate clearance between impeller and suction liner/shroud must be maintained to allow for thermal growth and shaft stretch due to thrust loads, and rotor and fluid weight.
2-25	If a pump experiences multiple mechanical seal failures, be sure to extend the analysis of the problem beyond the seal itself.
2-45	Most APM/APMA/APS performance curves have a discontinuity or hook. Typically minimum operating flow is 85% or greater than the pump best efficiency point (BEP) flow.
3-2	The most common vibration problems associated with vertical pumps are unbalance and misalignment.
3-5	Amplitude can be expressed in many forms such as displacement, velocity, and acceleration.
3-8	Frequency is the number of occurrences with respect to time. The frequency provides insight into the source of the vibration.
3-8	Phase angle provides the timing relationship of the vibration amplitude of any two points.
3-11	In most plants, vibration test personnel collect measurements to compare to formal or informal acceptance levels. If the vibration meets the acceptance criteria, then the pump is placed in service. If the pump does not perform satisfactorily, then the process of vibration diagnostics begins.
4-2	A precision balance of the vertical pump rotating components is important to low synchronous vibration and long-term pump service life.
4-5	Provide any special requirements or restrictions (hold points) that might be required, such as during pump disassembly and for inspection cleanliness.
4-5	Record concentricity and perpendicularity during disassembly only if repairs are not warranted by the dimensional inspection results but physical damage (for example, bearing wear, seal failures) indicates a problem or potential issue with the component.
4-6	An option for the above step is to include machining alignment registers on the underside of the motor support plate to aid in pump alignment at the plant. This option should only be performed, however, if the machining of all registers can be performed or verified in the vertical position and on the same setup.
4-6	Many times, the bearings, bushings, or case rings are damaged during disassembly, such that the ODs of the bearings, bushings, and case rings cannot be obtained. However, this should be recorded during assembly and verified to meet fit criteria.

Page Number	Key Point
4-8	NDE of suction head is intended for cast iron suction heads. Cast iron suction head bearing supports crack due to stress cracks (sometimes microscopic) being induced during manufacturing that propagate over time. If cracks are found, grind to remove them and evaluate the strength of supports.
4-8	"Weld repair impeller wear ring turns" applies to impellers that do not have replaceable wear rings. One option, to weld repair the impeller wear ring turn area, would be to undercut the ring turn and add a replaceable wear ring.
4-8	Shaft straightening is not a preferred option, due to the memory of most shaft materials. If possible, undercut coat journals and fits, and precision grind to get these components on the same centerline. If unbalance results, it should be corrected during rotor balancing.
4-9	APKD pump designs without setscrews to hold the suction head bearing journal sleeve in place have had failures. The setscrews prevent the sleeve from resting on the anti-rotation key and causing the key to shear. The sleeve continues to slide down on the shaft, making contact with the suction head. The original equipment manufacturer (OEM) offers a modified design that combines the impeller locknut and journal sleeve functions. The new design should work, but the use of the setscrews has proven to provide years of trouble-free service.
4-9	Burnishing the shaft for proximity probes would be a good idea even if permanent proximity probes are not installed at the facility, because troubleshooting might require the installation of temporary proximity probes.
4-10	A single, fitted coupling key is recommended so that the entire length of the coupling keyway is filled. The filled keyway reduces the unbalance inherent in the original design of the coupling with two keys. Also, the single key can aid the alignment of the adjoining shaft while the coupling is being installed.
4-11	Balance criteria are normally considered a total residual unbalance allowable. Divide the criteria by two for the per plane balance tolerance.
4-11	The calculation W/N, where W equals the weight of the component being balanced and N equals the operating speed at the plant, yields balance criteria in ounce-inches (oz-in). This calculation provides total residual unbalance criteria; divide by 2 to obtain unbalance per plane.
4-12	Restoration of concentricity, perpendicularity, and parallelism is a machining process. Welding is required only if fit does not meet the criteria.
4-12	An interference fit of up to 0.002 inch for male and female registers is often used in vertical pumps, particularly in condensate and heater drain pump service where bearing clearances are small and maximum reliable service life is desired.
4-12	An even number of pad welds (if not eight) should be required, so that there would be two pads 180° apart where the diameter for the register can be verified. A 100% weld is acceptable and might be preferable, depending on register size and material. Minimize welding and heat to cast iron parts.
4-14	One situation when a problem with sole plate levelness could be noted is when significant shimming between the pump and motor is required to obtain angular alignment. It is recommended that the criteria for base (sole) plate levelness be 0.001 inch per foot of diameter, with a maximum of 0.005 inch total.

Page Number	Key Point
4-14	The pump-to-motor alignment is about aligning the pump stuffing box centerline with the centerline of the motor bearings. Any of the procedures published, regardless of measurements being made at the pump coupling, shaft, or stuffing box, are designed to accomplish stuffing box and motor bearing centerline alignment. It is that fact that makes it imperative to establish the centerline of the pump bearings, wear rings, column registers, and discharge head registers, relative to the pump stuffing box centerline. It is imperative that the centerline of the pump bearings, wear rings, column registers, and discharge head registers, be ultimately corrected relative to the pump stuffing box centerline.
4-18	If the total travel does not make sense, double check the reading using a different person, method, or both.
4-20	Caution should be exercised when upgrading materials to preclude the creation of a potentially harmful galvanic environment.
4-21	Cavitation, erosion, or pitting occurs when vapor bubbles are transported into higher-pressure regions, causing them to form and collapse violently against the metal surface of the impeller. It occurs most frequently on the low-pressure side of the impeller inlet vanes.
4-21	The rate of abrasive wear is directly related to the velocity of the liquid, so wear will be more pronounced near the exit vanes and shrouds of pump impellers where the liquid velocity is highest.
4-21	Corrosion, unlike abrasive erosion, is generally independent of the liquid velocity. Pitting caused strictly by corrosion will be uniform over the entire surface.
4-36	In all vertical pump applications, the lift must be adequate to account for shaft stretch due to pump thrust loads and thermal expansion differences between the pump shafts and the pump columns.
4-52	The balancing machine measures only unbalance, never balance.
4-52	Unlike soft bearing machines, the use of calibration masses or shakers is not required to calibrate a hard bearing machine for a given rotor.
4-53	An unbalance value without a distance has no value or meaning.
4-61	The HVOF bond is a mechanical bond with the shaft; however, it appears superior to any other mechanical bonding process available on the market.
4-62	HVOF coatings are used in applications requiring the highest density and strength not found in most other thermal spray processes.
4-67	Remember, bulk strength is supplied by the substrate (cheap, strong and ductile). Surface properties are supplied by the coating (wear and corrosion, etc.).
4-70	Simply stated, a calculation of NPSH is an analysis of energy conditions on the suction side of a pump to determine if the liquid will vaporize at the lowest pressure point in the pump.

Key Human Performance Point

Denotes information that requires personnel action or consideration in order to prevent injury or damage or ease completion of the task.

Page Number	Key Point
3-1	Vibration testing is the science of using appropriate testing techniques, depending upon the results or outcome of previous data or information. This creates difficulty in establishing a standard test plan for a specific problem. The use of various techniques and technologies allows the evaluator to quickly focus on the key elements of a problem and bypass other non-vital information.
4-1	A key component in the repair process is making *the stuffing box centerline, the centerline of the pump* assumption valid.
4-4	It is important to verify shaft stick up during pump assembly, to ensure that no interference issues will arise between the pump and the motor shaft once installed at the plant.
4-70	The limited non-destructive testing available for thermally sprayed coatings should emphasize the need for a high standard of quality control over the process, to ensure a high level of confidence in the coated products.

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