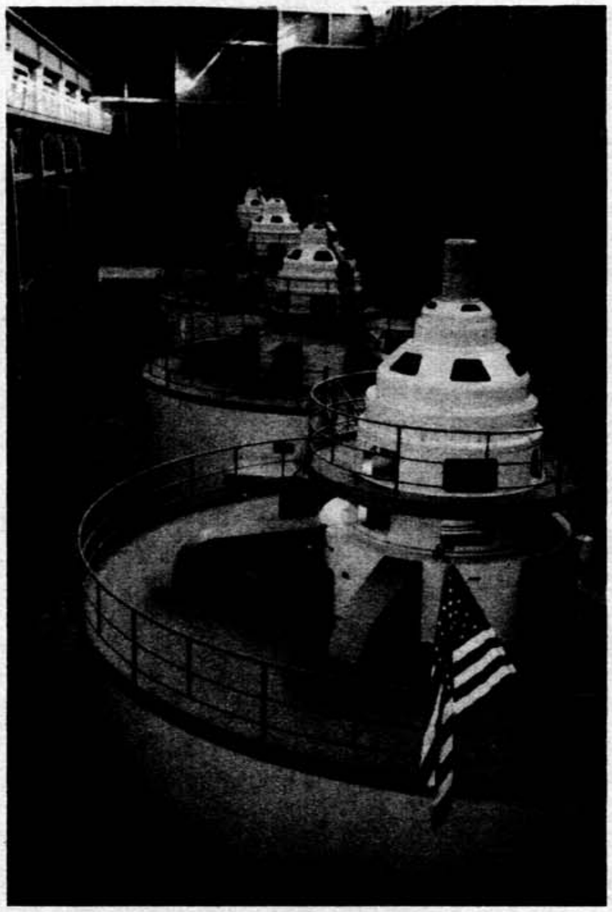
WATER OPERATION AND MAINTENANCE

BULLETIN NO. 160

June 1992



IN THIS ISSUE

MAINTENANCE AND PERIODIC INSPECTION OF MECHANICAL EQUIPMENT AT HYDROELECTRIC PLANTS

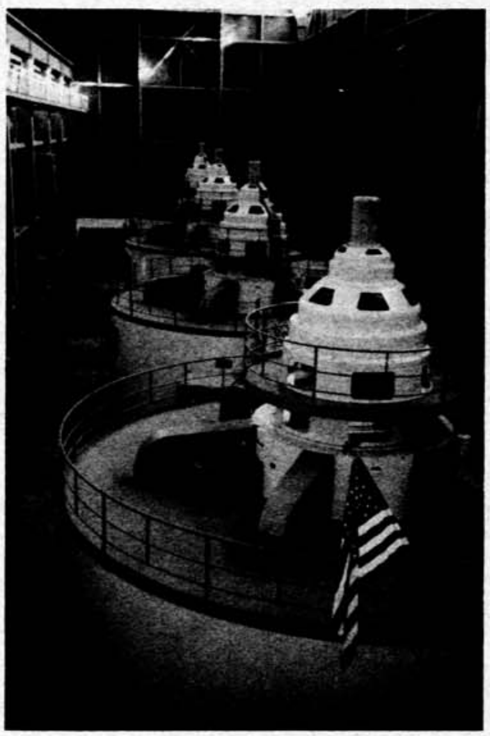
UNITED STATES DEPARTMENT OF THE INTERIOR
Bureau of Reclamation

The Water Operation and Maintenance Bulletin is published quarterly for the benefit of those operating water supply systems. Its principal purpose is to serve as a medium of exchanging information for use by Bureau personnel and water user groups for operating and maintaining project facilities.

While every attempt is made to insure high quality and accurate information, Reclamation cannot warrant nor be responsible for the use or misuse of information that is furnished in this bulletin.

* * * * *

Ferne Studer, Managing Editor
Bill Bouley, Technical Editor
Facilities Engineering Branch
Engineering Division
Denver Office, Code D-5210
PO Box 25007, Denver CO 80225
Telephone: (303) 236-8087



Cover photograph:

Palisades Dam Powerplant – Idaho/Wyoming

CONTENTS

WATER OPERATION AND MAINTENANCE BULLETIN

No. 160

June 1992

MAINTENANCE AND PERIODIC INSPECTION OF MECHANICAL EQUIPMENT AT HYDROELECTRIC PLANTS

CONTENTS — Continued

No.	FIGURES	Page

MAINTENANCE AND PERIODIC INSPECTION OF MECHANICAL EQUIPMENT AT HYDROELECTRIC PLANTS

By Roger A. Cline[1]

CHAPTER 1 — INTRODUCTION

1. General

The main reason for setting up a preventive maintenance program is to prevent unscheduled outages from failure of equipment. Depending on the circumstances, an unscheduled outage will be, at least, very inconvenient and can be extremely expensive. A well-designed program of preventive and routine maintenance should reduce equipment failures, extend the life of the equipment, and reduce the overall operating costs. Due to the wide variety of equipment in use today, an all encompassing maintenance guide applicable to every piece of equipment is not possible. This publication is intended to provide general information on the maintenance of some of the most common equipment found in powerplants. Manufacturer's literature and actual operating experience should be used for setting up a comprehensive maintenance program.

Probably the best place to start in setting up a maintenance program is the equipment manufacturer. The manufacturer should be the foremost authority on what is required to keep its equipment operating properly. Normally the manufacturer's operating manual will provide recommendations on lubricants, spare parts, maintenance procedures, and intervals between maintenance.

When preparing a maintenance schedule, keep in mind that the manufacturer's recommendations as well as the recommendations in this publication are general and are to be used only as a starting point. A particular piece of equipment may operate under much more severe conditions than the manufacturer expected. Conversely, the equipment may experience very mild service and not require as much attention as anticipated. This is why it is important to utilize personal experience and the equipment's history in preparing a maintenance schedule. An effective maintenance program requires tailoring the schedule to the equipment and the conditions under which it operates. Maintenance performed more frequently than required can cause undue wear and tear to the equipment being serviced as well as being a waste of time, while insufficient maintenance will cause premature equipment failure and a reduced service life. It should be noted that some equipment, most notably cranes and elevators, must be maintained on a regular basis to meet safety regulations.

An equipment maintenance record system is essential in establishing a successful preventive maintenance program. The record system should contain a description of the equipment and its location; manufacturer's data such as size, model, type and serial number; pertinent electrical and mechanical data; schedule for preventive maintenance and periodic inspections; data on repairs or maintenance performed including, actual

[1] Roger Cline is a Mechanical Engineer, Bureau of Reclamation, Facilities Engineering Branch, PO Box 25007, Denver CO 80225; telephone (303) 236-8964

work accomplished, material used, number of hours required to accomplish the work, and the cost of labor and materials.

In addition to the records for routine maintenance, after major overhauls or extraordinary maintenance, a more complete report should be written describing the work done and how it was accomplished. Pertinent photographs should be included in the report. This can be especially helpful in future maintenance if different personnel will be doing the work. These reports along with manufacturer's drawings and operation and maintenance material should be kept in a history file where they are readily accessible to maintenance personnel.

Well kept maintenance records are invaluable in any maintenance program. They provide the necessary information for establishing a preventive maintenance and inspection schedule and a spare parts inventory. Through the analysis of the data from these records, trends indicating deterioration of equipment may become apparent, allowing corrective action to be taken before failure occurs.

2. Inspection Checklists

The information contained in the following chapters is intended to provide general maintenance and inspection information for some of the most common equipment found in powerplants. This information, combined with actual operating experience and manufacturer's recommendations, should be used to develop specific inspection schedules and checklists.

The checklist should be concise, but descriptive enough to leave no question as to what information is required and how it should be obtained. For example, if a bearing temperature is to be checked, indicate where the thermometer is located and that the reading should be in degrees Fahrenheit, or degrees Centigrade. This should infer to the person performing the inspection that a simple checkmark indicating the temperature is okay is not acceptable. The checklist should also include the range of acceptable values or conditions for each item on the list. This will allow the person performing the checks to quickly recognize a problem, and notify maintenance personnel.

CHAPTER 2 — HYDRAULIC TURBINES, LARGE PUMPS, AND AUXILIARY PUMPS

1. Pumps

General.–Basically there are two general classifications of pumps: dynamic and positive displacement. These classifications are based on the method the pump uses to impart motion and pressure to the fluid.

Dynamic pumps.–Dynamic pumps continuously accelerate the fluid within the pump to a velocity much higher than the velocity at the discharge. The subsequent decrease of the fluid velocity at the discharge causes a corresponding increase in pressure. The dynamic pump category is made up of centrifugal pumps and special effect pumps such as eductor and hydraulic ram pumps.

Eductors, or jet pumps as they are sometimes called, use a high-pressure stream of fluid to pump a larger volume of fluid at a lower pressure. An eductor consists of three basic parts, the nozzle, the suction chamber, and the diffuser. The high-pressure fluid is directed through a nozzle to increase its velocity. The high velocity creates a low-pressure area that causes the low-pressure fluid to be drawn into the suction chamber. The low-pressure fluid is then mixed with the high-velocity fluid as it flows through the diffuser, and the velocity energy of the mixture is converted into pressure at the discharge. Eductors are commonly used in powerplants and dams to dewater sumps below the inlet of the sump pumps.

By far the most common type of dynamic pump is the centrifugal pump. The impeller of a centrifugal pump, the rotating component of the pump which imparts the necessary energy to the fluid to provide flow and pressure, is classified according to the direction of flow in reference to the axis of rotation of the impeller. The three major classes of centrifugal impellers are:

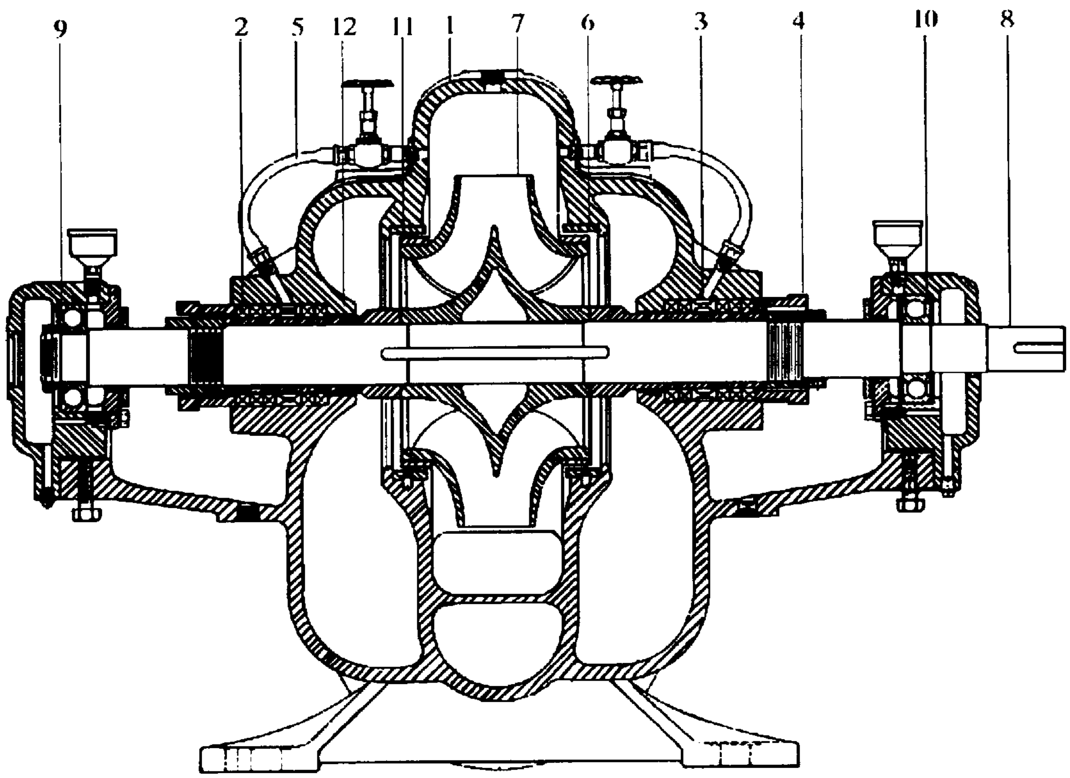
Axial-flow
Radial-flow
Mixed-flow

Impellers may be further classified by their construction. The impeller construction may be:

Open
Semi-open
Closed

An open impeller consists of vanes attached to a central hub. A semi-open impeller has a single shroud supporting the vanes, usually on the back of the impeller. The closed impeller incorporates shrouds on both sides of the vanes. The shrouds totally enclose the impeller's waterways and support the impeller vanes.

Centrifugal pumps are also classified by the means in which the velocity energy imparted to the fluid by the impeller is converted to pressure. Volute pumps use a spiral or volute shaped casing to change velocity energy to pressure energy. Pumps which use a set

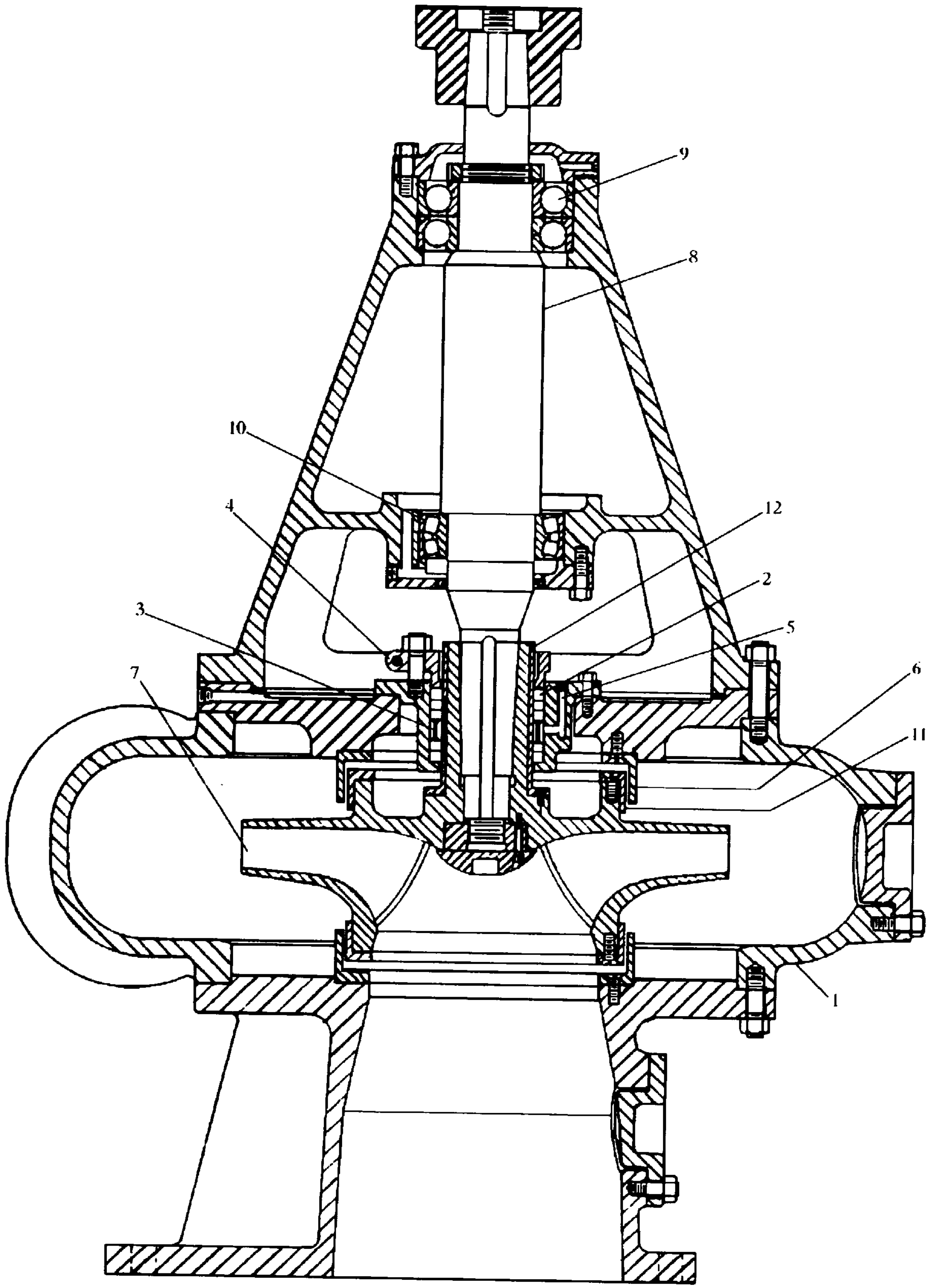


Double Suction Horizontal Volute Pump

PARTS LIST FOR HORIZONTAL AND VERTICAL PUMP DRAWINGS			
STATIONARY PARTS		ROTATING PARTS	
1	Pump Case	7	Impeller
2	Packing	8	Pump Shaft
3	Lantern Ring	9	Thrust Bearing
4	Packing Gland	10	Line Bearing
5	Packing Water Supply	11	Rotating Wear Ring
6	Stationary Wear Ring	12	Shaft Sleeve

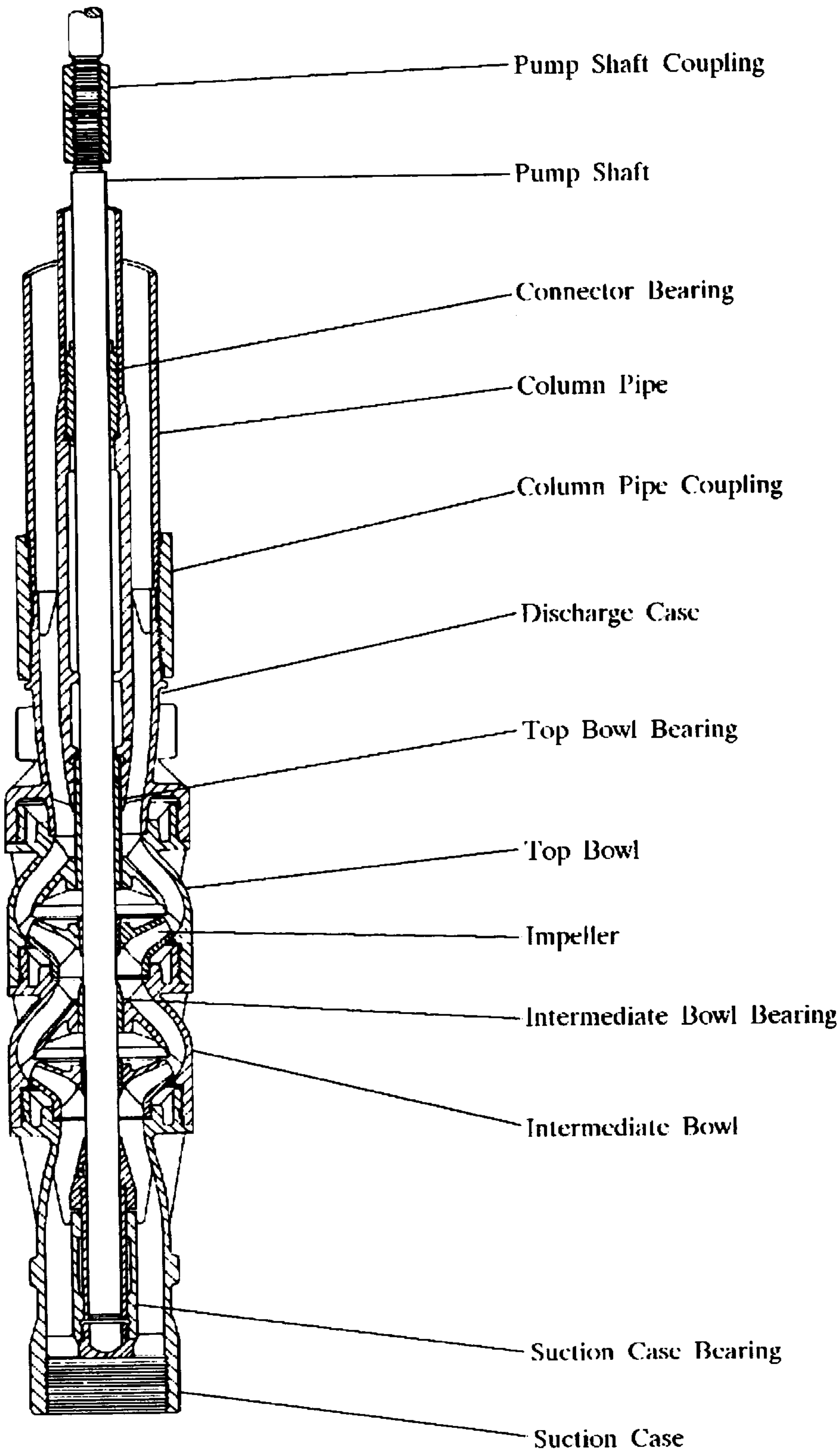
(Courtesy of Dresser Pump Division)

Figure 1. - Double suction horizontal volute pump.



Vertical Volute Pump
(Courtesy of Dresser Pump Division)

Figure 2. - Vertical volute pump.



Two Stage Vertical Turbine Pump
(Courtesy of Dresser Pump Division)

Figure 3. - Two stage vertical turbine pump.

of stationary diffuser vanes to change velocity to pressure are called diffuser pumps. The most common diffuser-type pumps are vertical turbine pumps and single-stage, low-head, propeller pumps. Large volute pumps may also have diffuser vanes, but while these vanes may direct the waterflow, their main purpose is structural and not energy conversion.

Centrifugal pumps are further classified as either horizontal or vertical, referring to the orientation of the pump shaft. In comparison to horizontal pumps, vertical pumps take up less floor space, the pump suction can be more easily positioned below the water surface to eliminate the need for priming, and the pump motor can be located above the water surface to prevent damage in the event of flooding. Vertical pumps can be either dry-pit or wet-pit. Dry-pit pumps are surrounded by air while wet-pit pumps are either fully or partially submerged. The dry-pit pumps are commonly used in medium- to high-head, large-capacity pumping plants. These large dry-pit pumps are generally volute pumps with closed, radial-flow impellers.

There is a variety of wet-pit pump designs for differing applications. One of the most common types is the vertical turbine pump. The vertical turbine pump is a diffuser pump with either closed or semi-open, radial-flow or mixed-flow impellers. Vertical turbine pumps, while originally designed for deep well applications, have a wide variety of uses, including irrigation pumping plants and sumps in powerplants and dams. This type of pump is normally constructed of several stages. A stage consists of an impeller and its casing, called a bowl. The main advantage of this type of construction is that system pressure can be varied by simply adding or reducing the number of stages of the pump. The use of vertical propeller pumps is normally limited to low-head, high-capacity use.

Horizontal pumps are classified according to the location of the suction pipe. The suction can be from the end, side, top, or bottom. Also common in horizontal pumps is the use of double-suction impellers. In a double-suction impeller pump, water flows symmetrically from both sides into the impeller which helps to reduce the axial thrust load.

Positive displacement pumps.–Positive displacement pumps enclose the fluid through the use of gears, pistons, or other devices, and push or "displace" the fluid out through the discharge line. Displacement pumps are divided into two groups: reciprocating, such as piston and diaphragm pumps; and rotary, such as gear, screw, and vane pumps. Since positive displacement pumps do "displace" the fluid being pumped, relief valves are required in the discharge line, ahead of any shutoff valve or any device that could conceivably act as a flow restriction.

Reciprocating piston or plunger pumps are suitable where a constant capacity is required over a variety of pressures. Piston and plunger pumps are capable of developing very high pressures, although capacities are somewhat limited. These pumps provide a pulsating output which, depending on the application, may be objectionable. The use of reciprocating pumps in hydroelectric powerplants is limited.

Rotary positive displacement pumps are used in a variety of applications, one of the most common being hydraulic systems. Some of the most common rotary pumps used in hydraulic systems are gear, vane, radial piston, and axial piston pumps. Screw pumps, with a single helical screw or meshing multiple screws, are most commonly used for

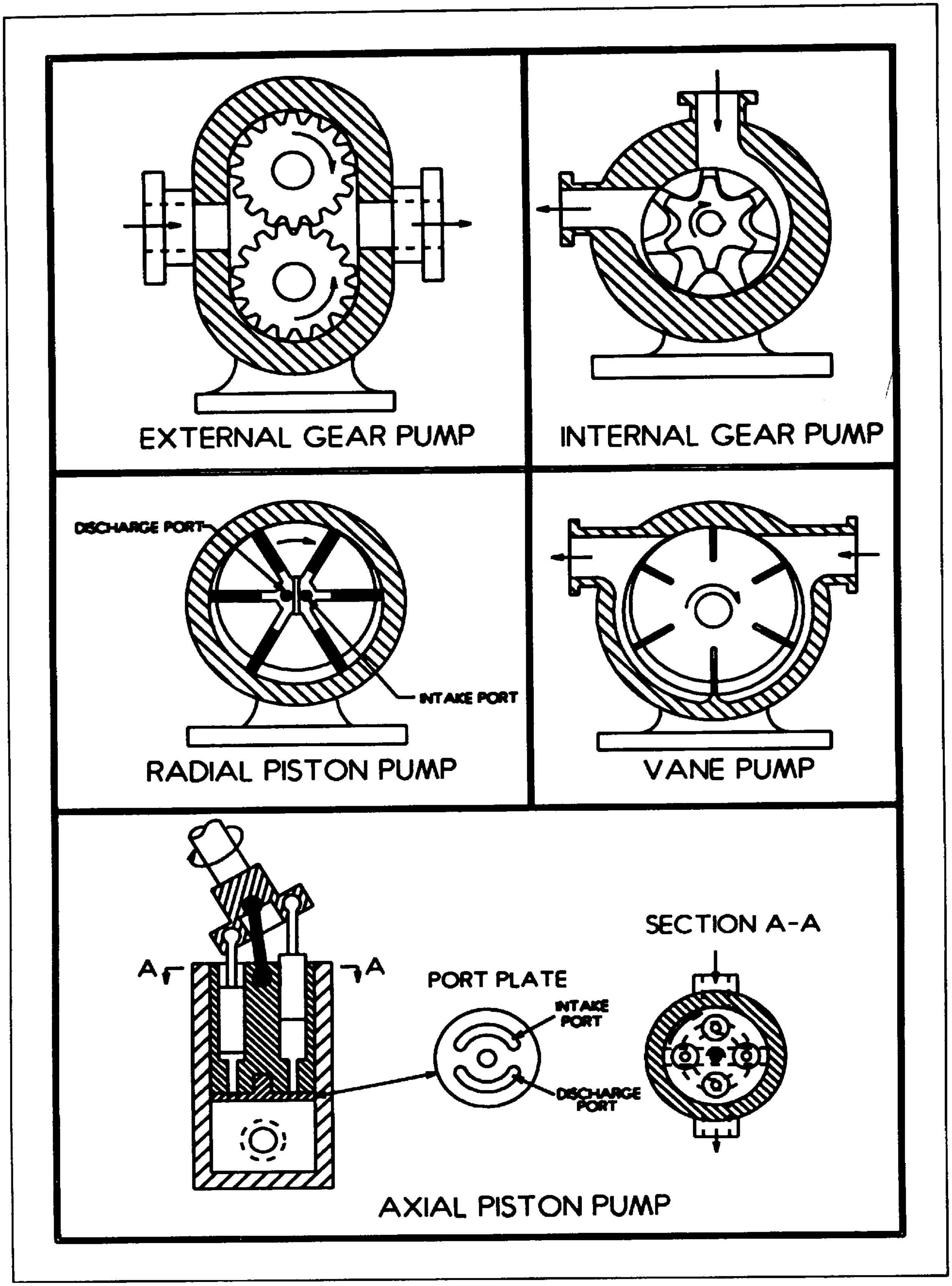


Figure 4. - Positive displacement pumps.

fluid transfer although they are sometimes used in hydraulic system applications such as governor oil pumps.

Gear pumps are relatively simple in design relying on the meshing of the mating gears and the fit of the gears in the pump casing to pump the fluid. External gear pumps utilize two meshing gears, usually spur or herringbone types, in a close fitting casing. The fluid is pumped as it is trapped between the rotating gears and the casing, and moved from the suction of the pump to the discharge. An internal gear pump utilizes an external gear which rotates eccentrically within and thereby drives an internal gear to pump the fluid.

Vane pumps consist of a case and a single eccentric rotor with multiple vanes sliding in slots in the rotor. Centrifugal force keeps the vanes in contact with the interior of the pump casing. As the rotor rotates, the fluid is drawn into the pump by the gradually increasing volume between the vanes, and is pushed out through the discharge as the volume gradually decreases.

The radial piston pump is similar in construction to the vane pump in that it has a single rotor, eccentric to the pump housing, but instead of vanes, it has radial pistons. The pistons are held against the pump housing by centrifugal force and the fluid is pumped by the reciprocating action of the pistons in their bore. The fluid ports are in the center of the rotor.

The axial piston pump rotor consists of a round cylinder block with multiple cylinders, parallel to the cylinder block axis. The cylinder block rotates at an angle to the axis of the drive shaft and the fluid is pumped by reciprocating action of the pistons in the cylinder block.

2. Hydraulic Turbines

Hydraulic turbines are classified as either reaction turbines or impulse turbines referring to the hydraulic action by which the pressure or potential energy is converted to rotating or kinetic energy. The reaction turbines include the Francis and the propeller types while the impulse turbines are represented by the Pelton-type turbine.

Impulse turbines convert all available head into kinetic or velocity energy through the use of contracting nozzles. The jets of water from the nozzles act on the runner buckets to exert a force in the direction of flow. This force, or impulse as it is referred to, turns the turbine. Impulse turbines are primarily used for heads of 800 feet or more although they are also used in some low-flow, low-head applications.

Waterflow to an impulse turbine is controlled by a hydraulic servomotor, controlled by a governor, which moves the needle of the nozzle. A moveable deflector plate, controlled by the governor, is positioned in front of the nozzle to rapidly deflect some of the water away from the turbine during a load rejection.

The head pressure in a reaction turbine is only partially converted to velocity. The reaction turbine obtains its power by a combination of the impulse force from the velocity of the water and a reaction force opposite in direction of waterflow exiting the runner.

VERTICAL FRANCIS TURBINE

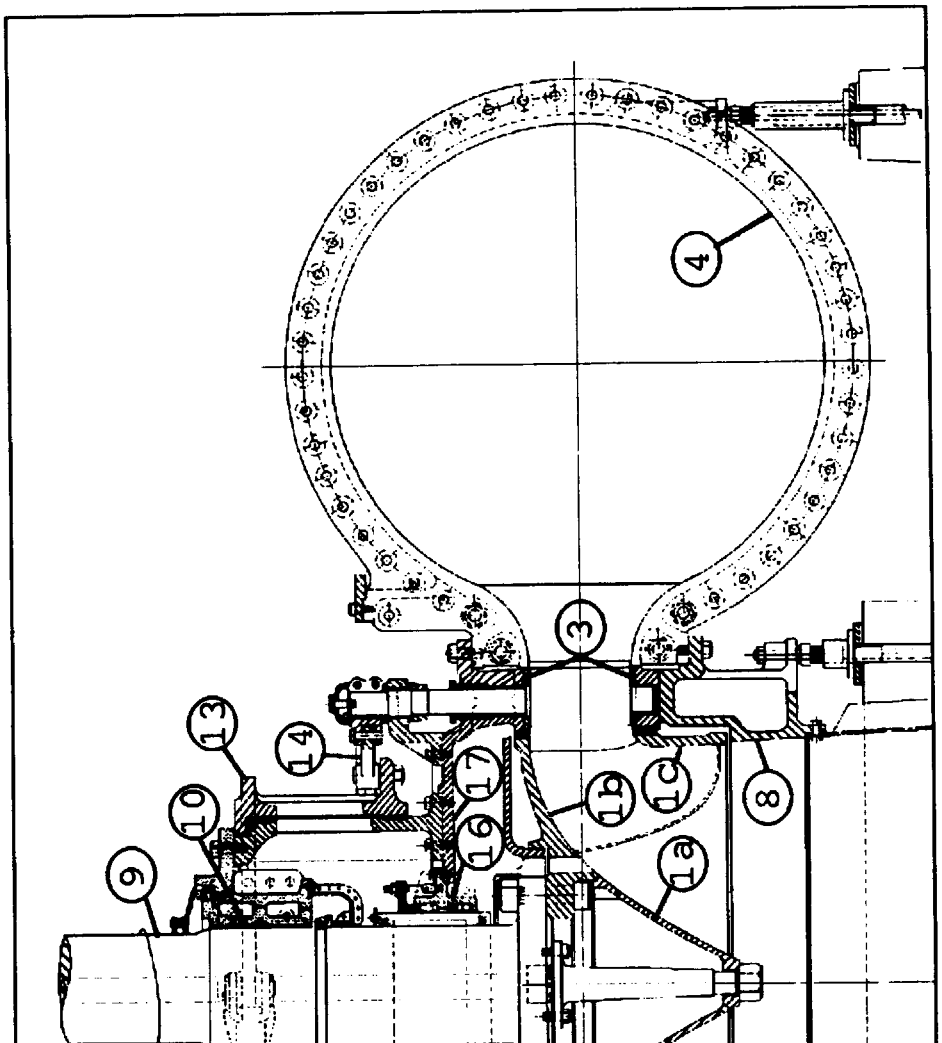


Figure 5. – Vertical Francis turbine.

The Francis turbine is very similar in construction to a volute pump with a closed impeller. Water entering the spiral or scroll case is directed to the turbine runner by the guide vanes and the wicket gates. The wicket gates, controlled by the governor through hydraulic servomotors, control waterflow to the turbine.

A propeller turbine is similar in appearance to a boat propeller. Water is directed and controlled in much the same manner as with the Francis turbine. A variation of the propeller turbine is the Kaplan turbine which features adjustable blades that are pivoted to obtain the highest efficiency possible at any load.

3. Cavitation Erosion, Abrasive Erosion, and Corrosion

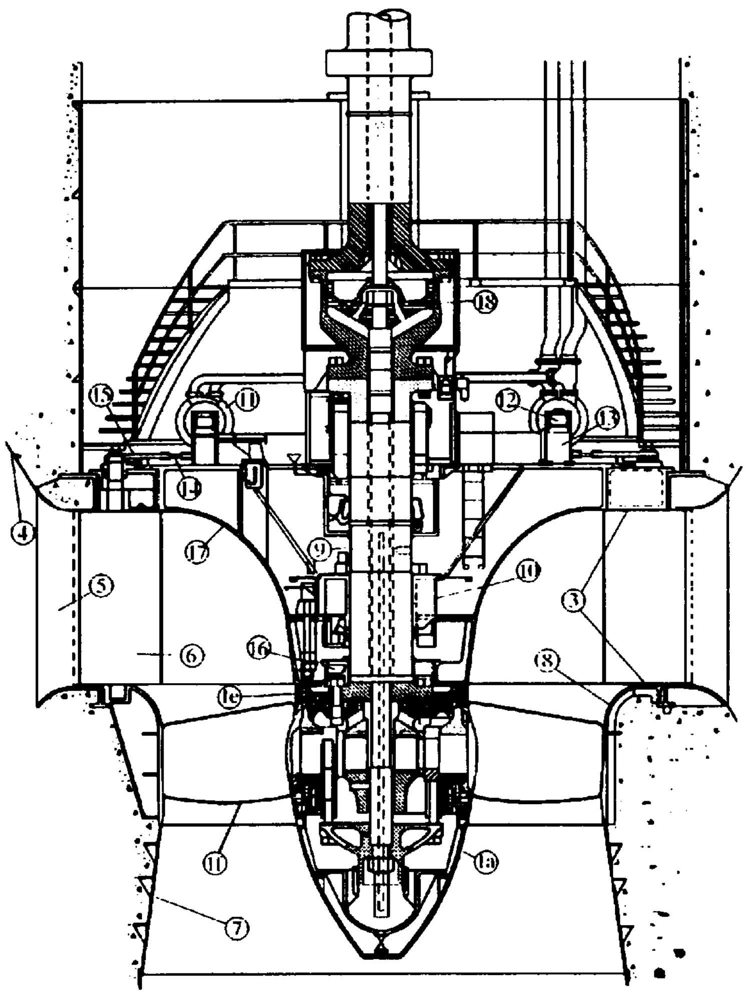
Pump impellers, turbine runners, and their related components may be damaged by a number of different actions, the most common being cavitation erosion, abrasive erosion, and corrosion. The appropriate repair procedure will depend on the cause of the damage.

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 which can be greater than 100,000 lb/in^2. When audible, cavitation makes a steady crackling sound similar to rocks passing through the pump or turbine. Cavitation erosion or pitting occurs when the bubbles collapse against the metal surface of the impeller and occurs most frequently on the low-pressure side of the impeller inlet vanes or turbine buckets. Cavitation cannot only severely damage the pump or turbine, but it can also substantially reduce the capacity and therefore lessen the efficiency.

Abrasive erosion is the mechanical removal of metal by suspended solids, such as sand, in the liquid being pumped or the water flowing through a turbine. The rate of wear is directly related to the velocity of the liquid, so wear will be more pronounced at the discharge of the nozzle of impulse turbines, near the exit vanes and shrouds of pump impellers, and near the exit area of reaction turbines 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.

Diagnosis of the problem can be difficult as the damage may 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.



VERTICAL KAPLAN TURBINE

PARTS LIST FOR FRANCIS AND KAPLAN TURBINE DRAWINGS			
1	Turbine Runner	7	Draft Tube
1a	Runner Cone	8	Discharge Ring
1b	Runner Crown (Francis)	9	Turbine Shaft
1c	Runner Band (Francis)	10	Turbine Guide Bearing
1d	Runner Bucket (Francis)	11	Wicket Gate Servomotors
1e	Runner Hub (Kaplan)	12	Servomotor Connecting Rod
1f	Runner Blade (Kaplan)	13	Wicket Gate Operating Ring or Shift Ring
2	Wearing Rings or Seal Rings (Francis)	14	Wicket Gate Link
3	Facing Plates or Curb Plates	15	Wicket Gate Arm
4	Spiral Case or Scroll Case	16	Packing Box or Stuffing Box (Mechanical Seals)
5	Stay Vane	17	Head Cover
6	Wicket Gate	18	Runner Blade Servomotor (Kaplan)

Figure 6. – Vertical Kaplan turbine.

Severe erosion or corrosion damage may 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 impeller or runner, or other design changes may 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 may be obtained by replacement.

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 method of repair for cavitation damage. Repair with non-fusing materials, such as epoxies and ceramics, is generally not successful because the low bond strength of these materials, usually less than 3,000 lb/in^2, is not capable of withstanding the high shock pressures encountered during cavitation. Prior to any weld repair, a detailed welding procedure should be developed. Welding performed incorrectly, can cause more damage, by distortion and cracking, than the cavitation did originally. Cavitation repair is discussed in more detail in FIST (Facilities Instructions, Standards, and Techniques) Volume 2-5, Turbine Repair[2].

Corrosion or erosion damage, if the pitting is deep enough, can also be repaired by welding. If the pitting is definitely not caused by cavitation, other coating or fillings may be acceptable. The epoxies and ceramics discussed earlier, if properly applied, can be helpful in filling in pitting damage caused by corrosion or erosion. In a corrosive environment, a coating of paint, after the original contour has been restored, can offer protection by forming a barrier between the metal and the electrolyte and preventing the electrochemical reaction.

Erosion-resistant coatings, in order to be effective, must be able to withstand the cutting action of the suspended abrasive. A coating of neoprene has been proven successful for sand erosion protection. There are other coatings available that have also been proven to be resistant to erosion, but many of these coatings can be difficult to apply and maintain, and may, because of coating thickness, restrict water passages somewhat. Erosion-resistant coatings should be chosen based on the design of the turbine or pump and the severity of erosion.

4. Wearing Rings

The purpose of wearing rings, or seal rings as they are also called, is to provide a renewable seal or leakage joint between a pump impeller or a turbine runner and its casing. As the name implies, these rings can wear over time and as the clearance increases, efficiency can decrease. As a general rule, when the wearing ring clearance exceeds 200 percent of the design clearance, the wearing rings should be replaced or renewed. If a design does not include replaceable wearing rings, it may be necessary to build up the wearing ring area by welding or other acceptable process and machining back to the original clearances; remachine the wear ring area and impeller or runner to accept replaceable wearing rings; or, on small pumps, replace the impeller and casing.

[2] FIST bulletins listed in "Publications for Sale" booklet, Attention D-7923A, PO Box 25007, Denver CO 80225-0007.

The location of the wearing rings varies depending on the design of the pump or turbine. Francis turbines and most closed impeller pumps have two wearing rings although some pump impellers may only have a suction side wearing ring. Propeller turbines, open impeller, and many semi-open impeller pumps do not have wearing rings, relying instead on a close fit between the runner or impeller vanes and the casing to control leakage.

5. Packing/Mechanical Seals

Packing.-The most common method of controlling leakage past a pump, turbine, or wicket gate shaft is by the use of compression packing. The standard packing or stuffing box will contain several rings of packing with a packing gland to hold the packing in place and maintain the desired compression. Some leakage past the packing is necessary to cool and lubricate the packing and shaft. If additional lubrication or cooling is required, a lantern ring also may be installed.

Over time the packing gland will have to be tightened to control leakage. To prevent burning the packing or scoring the shaft when these adjustments are made, most compression packings contain a lubricant. As the packing is tightened, the lubricant is released to lubricate the shaft until leakage past the packing is reestablished. Eventually, the packing will be compressed to a point where no lubricant remains and replacement is required. Continued operation with packing in this condition can severely damage the shaft.

When packing replacement is necessary, remove all of the old packing. If the packing box is equipped with a lantern ring, this also must be removed along with all of the packing below it. With the packing removed, special attention should be given to the cleaning and inspection of the packing box bore and the shaft or shaft sleeve. To provide an adequate sealing surface for the new packing, a severely worn shaft or shaft sleeve should be repaired or replaced. Likewise, severe pitting in the packing box bore should be repaired. In order for the packing to seal against a rough packing box bore, excessive compression of the packing is required. This over-compression of the packing will lead to premature wear of the shaft or shaft sleeve.

The shaft runout at the packing box should be checked with a dial indicator. In most cases, total indicated runout should not exceed 0.003 inch, although the shaft runout on large vertical units can usually be somewhat greater than this. If the runout is excessive, the cause should be found and corrected. Bent shafts should be replaced and misalignment corrected.

There are a number of different types of packing available; so when choosing new packing, care should be taken to ensure that it is the correct size and type for the intended application. Be sure to inform the packing distributor of all the relevant conditions, such as shaft size and rotational speed, the packing will operate under. Installing the wrong packing can result in excessive leakage, reduced service life, and damage to the shaft or sleeve.

The new packing should be installed with the joints staggered 90 degrees apart. It is sometimes helpful to lubricate the packing prior to installation. The packing manufacturer should be consulted for recommendations for a lubricant and for any special instructions that may be required for the type of packing being used. With all of the packing and the lantern ring in place the packing gland should be installed finger tight.

There should be generous leakage upon the initial startup after the installation of new packing. The packing gland should be tightened evenly and in small steps until the leakage is reduced sufficiently. The gland should be tightened at 15- to 30-minute intervals to allow the packing time to break-in. The temperature of the water leaking from the packing should be cool or lukewarm, never hot. If the water is hot, back off the packing gland.

Mechanical seals.–Mechanical seals are used in both pump and turbine applications. Mechanical seals allow very little leakage and can be designed to operate at high pressures. Properly installed mechanical seals will have a long service life and require little maintenance.

Basically, a mechanical seal on a small pump consists of a stationary and a rotating member with sealing surfaces perpendicular to the shaft. The highly polished sealing surfaces are held together by a combination of spring and fluid pressure and are lubricated by maintaining a thin film of the fluid sealed between the surfaces.

There is a wide variety of mechanical seals available for small pump applications, each having its own distinct installation procedure; therefore, it is important to follow the seal manufacturer's installation instructions as closely as possible. The manufacturer should also provide information of the allowable shaft runout and endplay for the particular seal.

Mechanical seals used in hydraulic turbines and large pumps consist of sealing segments, usually made of carbon, held against the shaft by spring tension and lubricated by a thin film of water. These seals usually require grease lubrication prior to startup if the unit is shut down for extended periods.

Since mechanical seals are precisely made and rely on very tight tolerances in order to operate successfully, a great deal of care must be taken during the installation. Just a small amount of dirt or other contaminants on the polished sealing surfaces can allow leakage past the seal and reduce the seal's life.

6. Bearings

General.–The purpose of the bearings is to locate and support the shafts of a pump or turbine. The bearings can provide radial support (line or guide bearings), axial support (thrust bearings), or both. The most common types of bearings are fluid film and antifriction bearings.

Fluid film bearings.–Fluid film bearings derive their load-carrying capacity through the formation of an "oil wedge" as the shaft or thrust runner rotates. The formation of this "oil wedge" is similar to the fluid wedge that forms under a speeding boat, raising its bow out of the water. The force of the wedge in a bearing must be sufficient to balance the load to the bearing surfaces.

Fluid film, or plain bearings, are normally used on turbines and large pumps and can be in the form of sleeve bearings, either solid or split, tilting pads, or pivoted thrust shoes. These bearings usually consist of a cast iron or steel bearing shell with a tin- or lead-based babbitt lining. Bronze bushings are used for line shaft bearings in vertical wet-pit pumps and on some horizontal pumps.

The thrust and upper guide bearings of large vertical generators are insulated from the frame to prevent circulating current from passing through the bearing. The bearing can be quickly damaged or destroyed if not adequately insulated. Test terminals are usually provided to check the insulation. For more information on bearing insulation testing, refer to FIST Volume 3-11, Generator Thrust Bearing Insulation and Oil Film Resistance.

Antifriction bearings.–The antifriction bearings, through the use of rolling elements, utilize the low coefficient of rolling friction as opposed to that of sliding friction of the fluid film bearing, in supporting a load. The most common types of antifriction bearings are "ball" and "roller" bearings, referring to the shape of the bearing's rolling elements. These bearings are also classified as "radial," "radial-thrust," or "thrust" bearings according to the type of load they are meant to support.

An antifriction bearing is a delicate, precision made piece of equipment; and a great deal of care should be taken during installation. The bearing manufacturer will usually provide instructions and precautions for the installation of a particular bearing and these instructions should be followed closely. Cleanliness is probably the most important thing to take into consideration in handling antifriction bearings. Any dust or dirt can act as an abrasive and quickly wear the bearing's rolling elements; therefore, it is important to work with clean tools and clean hands and to clean the bearing housings, covers, and shaft prior to installation. The new bearing should not be cleaned or wiped prior to installation unless it is recommended by the manufacturer. Bearings should be pressed onto shafts using adapters that apply even pressure to the inner race only. Never hammer a bearing onto a shaft.

7. Shaft Couplings

Couplings are used to connect the turbine or pump shaft to its respective generator or motor shaft and to transmit rotary motion and torque from the driver to the driven device. There are basically two types of couplings, rigid and flexible.

Rigid couplings require precise alignment and are most commonly used in vertical units where the entire weight is supported by thrust bearings in the motor or generator. Flanged and threaded couplings are the most widely used rigid couplings. Flanged couplings are used on large vertical units and consist of precisely machined flanges on each shaft, connected by a series of coupling bolts around the perimeter of the flanges. Threaded couplings, used to connect the line shafts of vertical turbine pumps, are cylindrically shaped with internal threads matching the external threads on the line shafts. The shafts to be coupled are simply screwed tightly into either end of the coupling.

Flexible couplings are designed to accommodate slight misalignment between shafts and, to some extent, dampen vibration. The amount of misalignment allowable is completely dependent of the design of the particular coupling. Since there are a number of flexible coupling designs, tolerances for misalignment should be obtained from the coupling manufacturer. The flexibility of the couplings can be provided through clearances between mating parts, as in gear and chain couplings, or through the use of a flexible material in the coupling, as in flexible disk and compression couplings. Horizontal pumps usually employ some sort of flexible coupling to connect the pump to its driver.

If properly aligned, most couplings should require very little maintenance outside of periodic inspection and, in some cases, lubrication. Over time, the alignment between the pump and its driver can deteriorate, increasing stress on the coupling which can lead to a shorter life.

8. Shaft Alignment

General.–Misalignment is a common and sometimes serious problem. Poor alignment can cause premature wear or failure of bearings, overheating of shaft couplings, and in extreme cases, cracked or broken shafts. The procedure for alignment depends on the type of equipment and its design.

Large vertical units, suspended from a thrust bearing in the motor or generator, require plumbing the shaft and making all guide bearings concentric. The procedure for aligning these units is discussed in detail in FIST Volume 2-1, Alignment of Vertical Shaft Hydro Units.

The line shafts of vertical turbine pumps are held in alignment by line shaft bearings in the pipe column. The proper alignment of the line shaft depends on the proper assembly of the pipe column and the bearing retainers. Depending on the design, the pump motor to line shaft coupling may be aligned by the face and rim alignment method or the reverse indicator alignment method described below. Refer to the pump manufacturer's instructions for specific directions for assembly and alignment.

Horizontal pump alignment.–Horizontal pumps are usually coupled to the pump driver with a flexible coupling. The amount of misalignment a flexible coupling can tolerate is dependent on its design. The coupling's manufacturer should provide installation instructions indicating the allowable tolerances for a particular design. A horizontal pump can usually be aligned acceptably by either the face and rim alignment method or reverse indicator alignment method. In most cases, the pump driver is aligned to the pump, as the pump is usually connected to rigid piping and is more difficult of move.

Preliminary checks for alignment of horizontal pumps.–

a. At least 0.125 inch of nonrusting shims should be installed under each leg of the motor to allow for adjustments that may be required during the alignment procedure.

b. Compensation should be made for any "soft" or "dead" foot condition. A "soft foot" condition is comparable to a short leg on a four legged table. To check for a "soft foot," make sure all four feet are securely bolted to the baseplate. With a dial indicator, check the rise of each foot as its holddown bolt is loosened. Retighten the holddown bolt after the rise is recorded, so that only one bolt is loose at a time. If one foot rises more than the other three, that foot is the "soft foot." For example, if one foot rises 0.005 inch while the other three rise only 0.002 inch, a 0.003-inch shim should be added to the "soft foot."

c. The holddown boltholes should be checked for sufficient clearance to allow for movement during the alignment procedure.

d. The mounting brackets and extension bars used for the indicators should be constructed to minimize sag. Sag is the effect of gravity on the indicator extension bar and can greatly affect the accuracy of the readings when using the reverse indicator alignment method or rim readings of the face and rim alignment method. The sag of an indicator bar can be determined by securely attaching the bar to a section of rigid bar stock or a shaft mandrel. The bar stock or mandrel can be supported and rotated by hand or between centers on a lathe. With the indicator bar positioned on top, zero the indicator and rotate the bar stock 180 degrees. The indicator reading will be twice the actual amount of bar sag. To correct alignment readings for sag, add twice the amount of bar sag to the bottom indicator reading. The bar sag is always expressed as a positive number regardless of indicator convention.

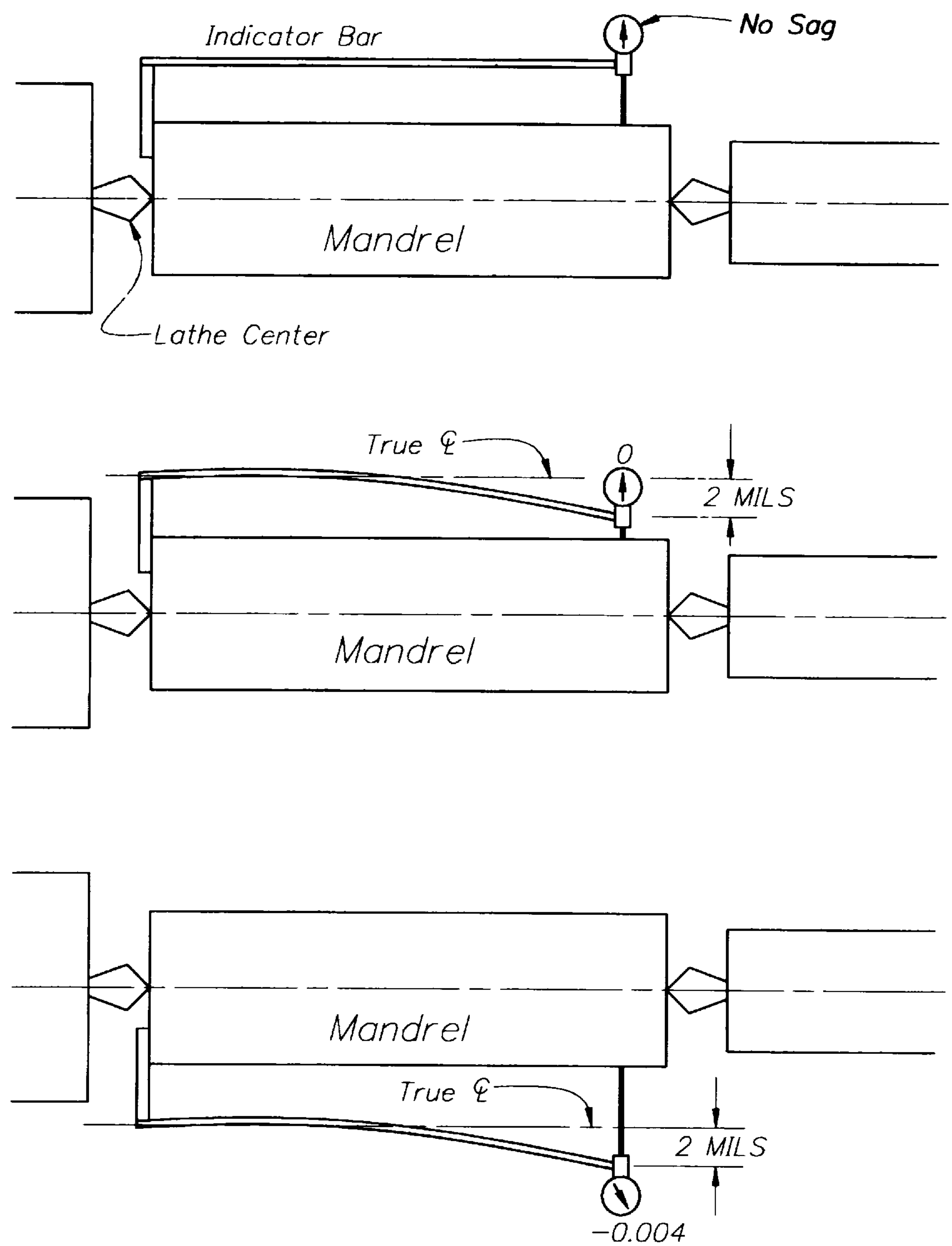
Important: The procedures described below for the face and rim and the reverse indicator alignment methods assume that movement toward the indicator moves the indicator needle in the positive direction, while movement away from the indicator moves the needle in the negative direction.

If the indicator used has the opposite sign convention; that is, movement toward the indicator moves the needle in the negative direction, the following corrections will have to made. In using the face and rim alignment method, simply make all movements in the opposite direction indicated in the procedure. When using the reverse indicator alignment worksheet, the value in column 2 should be subtracted from the bottom reading in column 1, instead of adding, to obtain the corrected value in column 3. In determining the direction of the motor shaft from the pump shaft in column 6, circle the direction corresponding to the sign opposite that in column 5. For example, if the values in column 5 for the pump indicator are +5 and -1, the directions in column 6 would be below and left.

Face and rim alignment method.–The face and rim method of alignment utilizes a dial indicator attached to one of the coupling flanges to check for angular (dogleg) and parallel (offset) misalignment. Indicator readings can be taken by rotating just one shaft, but in order to compensate for an untrue surface on the face or rim of the coupling flange, both shafts should be rotated together in the direction of normal rotation. If it is not possible to rotate both shafts, the indicator should be attached to the shaft that is rotated. The procedure is the same whether one or both shafts are rotated. The data obtained during the face and rim alignment procedure provide fairly accurate values for the movement or shims required to correct parallel misalignment, but some trial and error may be required to correct angular misalignment.

a. Angular alignment.–

(1) With the indicator attached to one coupling half and the indicator button resting near the outer edge of the other coupling's flange face, rotate the shaft with the indicator so that the indicator is at the top or 12 o'clock position and zero the indicator.



CHECKING INDICATOR BAR FOR SAG

Figure 7. – Checking indicator bar for sag.

(2) Rotate both shafts 180 degrees and read indicator. If the reading is positive, the rear legs of the motor must be raised or the front legs lowered. A negative reading requires lowering the rear legs or raising the front legs.

(3) Looking from the pump end, rotate both shafts so that the indicator is at the 3 o'clock position and zero the indicator.

(4) Rotate both shafts 180 degrees and read indicator. If the reading is positive, the rear of the motor must be moved to the right or the front of the motor moved to the left. A negative reading requires moving the rear of the motor to the left or the front to the right.

(5) Angular alignment is acceptable when dial indicator readings are zero at all positions or within the tolerances specified by the coupling manufacturer.

b. Parallel alignment.-

(1) Reposition indicator so that the button rests on rim of coupling flange. Rotate shaft with indicator so that the indicator is on top or at the 12 o'clock position and zero indicator.

(2) Rotate both shafts 180 degrees and read indicator. Add twice the amount of actual bar sag to obtain the corrected reading. If the indicator is attached to the pump coupling, a positive corrected reading requires raising the motor by half of the corrected reading. For example, if the reading is +0.010, a 0.005-inch shim should be added to each motor leg. A negative corrected reading requires lowering the motor. If the indicator is attached to the motor coupling, a positive reading requires lowering the motor and a negative reading requires raising the motor.

(3) Looking from the pump end, rotate both shafts so that the indicator is at the 3 o'clock position and zero the indicator.

(4) Rotate both shafts 180 degrees and read indicator. If the indicator is attached to the pump coupling, a positive reading requires moving the motor to the right and a negative reading requires moving the motor to the left. If the indicator is attached to the motor coupling, a positive reading requires moving the motor to the left and a negative reading requires moving the motor to the right.

(5) Parallel alignment is acceptable when dial indicator readings are zero at all positions or within the tolerances specified by the coupling manufacturer.

(6) After any movement of the motor, repeat steps a.(1) through a.(2) to check angular alignment.

Reverse indicator alignment method.-The reverse indicator method of alignment can be used when it is possible to rotate both shafts. This method utilizes two dial indicators, one attached to each shaft, taking a reading on the opposite shaft. Indicator brackets are available that allow the indicator to be attached directly to shaft, indicating off the indicator bar. This arrangement reduces bar sag and eliminates inaccuracies caused by poor surface condition of the shaft. From the data obtained by the reverse indicator

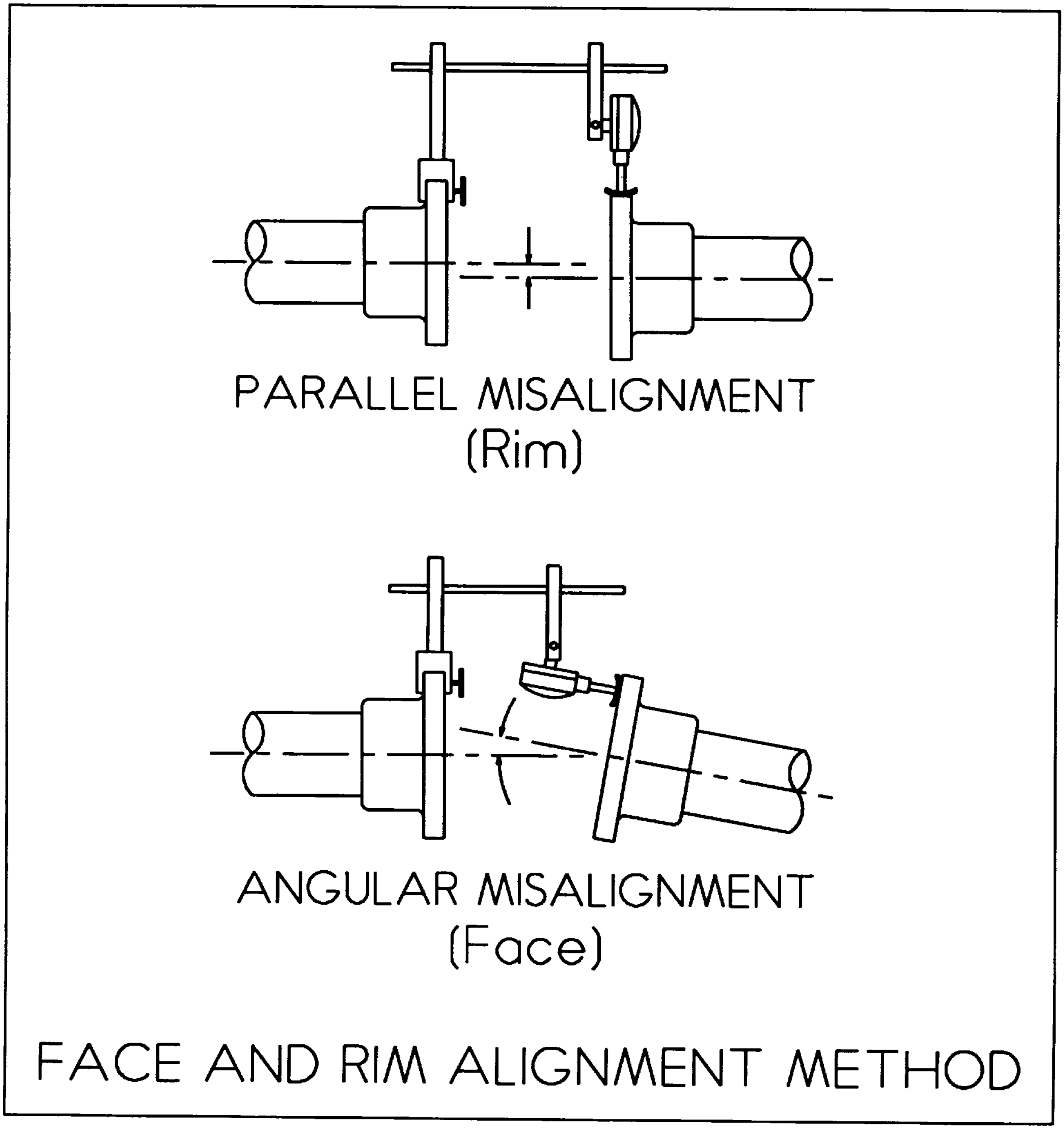


Figure 8. – Face and rim alignment method.

method, it is possible to determine, either analytically or graphically, the movement or shims necessary to align the shafts. A graphical method is presented below.

a. Record indicator readings.-

(1) Attach indicator bars and indicators to shafts and position shafts so that the pump indicator, which is the indicator nearest the pump, is on top and the motor indicator is on the bottom. By increasing the span between the indicators, the accuracy of the readings can usually be increased, although bar sag may also increase. Zero both indicators at this position.

(2) Rotate both shafts, preferably in the direction of normal rotation, and record the indicator readings at 90-degree intervals. For consistency, right and left readings should be designated for both shafts looking from the pump end toward the motor end. Both indicators should read zero at 360 degrees. If not, zero indicators and retake readings. **It is very important to record whether a reading is positive or negative and to keep track of each value's sign while performing the addition and subtraction in the following steps.**

(3) To correct for bar sag, add twice the actual amount of sag to the bottom readings.

(4) Subtract the top reading from the corrected bottom reading and the left reading from the right reading and divide the differences by 2. These values will be used for plotting the position of the shafts.

b. Plot data.-

(1) Two graphs will be needed. One for the horizontal plane (top view) and one for the vertical plane (side view). The horizontal scale of both graphs will represent the horizontal distance from the plane of the pump indicator to the plane of the rear motor feet. Since the pump shaft will not be moved, it will be used as the horizontal reference in determining the position of the motor shaft. The vertical scale will represent the misalignment of the motor shaft.

(2) Establish the horizontal scale, marking with vertical lines, the relative position of both indicators and the front and rear motor feet. Draw two horizontal lines representing the pump shaft reference line for the horizontal and vertical planes. A vertical scale of 0.001 inch per division is usually satisfactory.

(3) Plot the values from step a.(4). These values represent the vertical distance from the pump shaft line to the motor shaft line at each of the indicator locations. The top-bottom readings are used in the vertical plane plot and the left-right readings are used in the horizontal plane plot. The sign convention is different for the two indicators. If the values for the pump indicator are positive, the plot will be above and left of the pump shaft reference line. The plot will be below and right of the pump shaft reference line for positive motor indicator readings.

(4) Draw a line from the pump indicator point through the motor indicator pump point extending to the rear motor feet line. This line represents the position of the motor shaft. The vertical distances from the motor shaft line to the pump shaft

Reverse Indicator Alignment Worksheet

		Column 1 Actual Reading	Column 2 Correction to bottom Reading for Bar Sag (Twice Actual Amount)	Column 3 Column 1 + Column 2	Column 4 Bottom - Top Right - Left	Column 5 1/2 Column 4 Distance of Motor Shaft Line from Pump Shaft Line	Column 6 Direction of Motor Shaft Line from Pump Shaft Line Circle Direction Corresponding to Sign of Valve in Column 5
Pump Indicator	Bottom						+ Above
	Top	0	0				- Below
	Right		0				+ Left
	Left		0				- Right
	Top						
Motor Indicator	Bottom	0					+ Below
	Top		0				- Above
	Right		0				+ Right
	Left		0				- Left
	Bottom						

1. Zero indicators with pump indicator at top position and motor indicator at the bottom. (Pump indicator is indicator nearest pump)
2. Left and right for both indicators is determined by looking from pump end towards motor end.
3. The second top reading for the pump indicator and the second bottom reading for the Motor Indicator should be zero. If not, repeat all readings.

Indicator Bar Sag = ____________ A = ____________ B = ____________ C = ____________

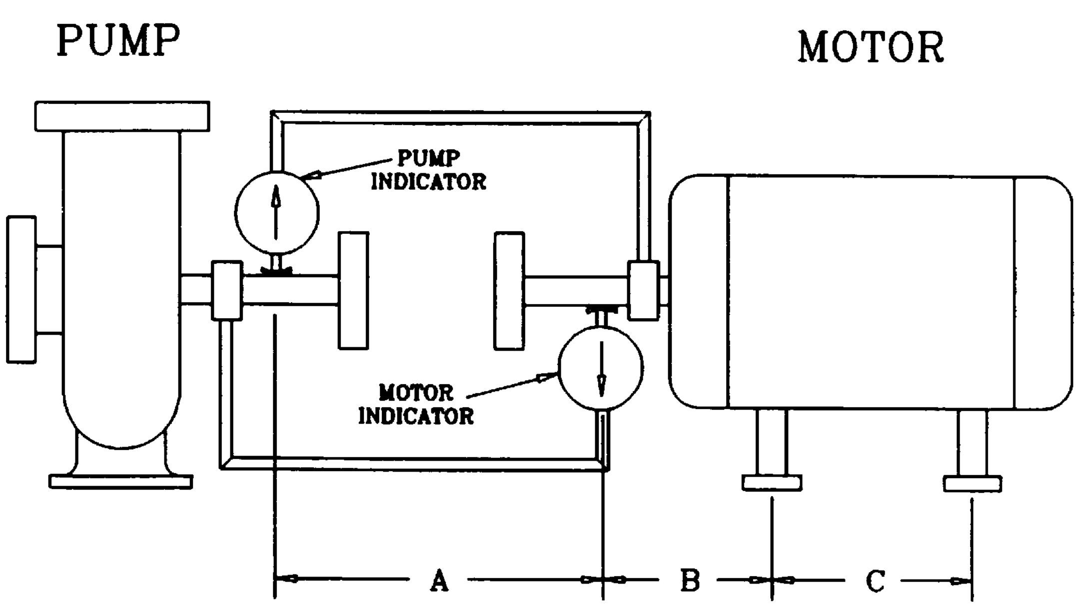


Figure 9. – Reverse indicator alignment worksheet.

Reverse Indicator Alignment Worksheet

		Column 1 Actual Reading	Column 2 Correction to bottom Reading for Bar Sag (Twice Actual Amount)	Column 3 Column 1 + Column 2	Column 4 Bottom - Top Right - Left	Column 5 1/2 Column 4 Distance of Motor Shaft Line from Pump Shaft Line	Column 6 Direction of Motor Shaft Line from Pump Shaft Line Circle Direction Corresponding to Sign of Valve in Column 5
Pump Indicator	Bottom	12	+2	- 10	-10	-5	+ Above
	Top	0	0	0			- Below
	Right	-5	0	-5	+2	+1	+ Left
	Left	-7	0	-7			- Right
	Top	0					
Motor Indicator	Bottom	0	+2	+2	+12	+6	+ Below
	Top	-10	0	-10			- Above
	Right	-4	0	-4	+2	+1	+ Right
	Left	-6	0	-6			- Left
	Bottom	0					

1. Zero indicators with pump indicator at top position and motor indicator at the bottom. (Pump indicator is indicator nearest pump)
2. Left and right for both indicators is determined by looking from pump end towards motor end.
3. The second top reading for the pump indicator and the second bottom reading for the Motor Indicator should be zero. If not, repeat all readings.

Indicator Bar Sag = 0.001" A = 16" B = 4" C = 40"

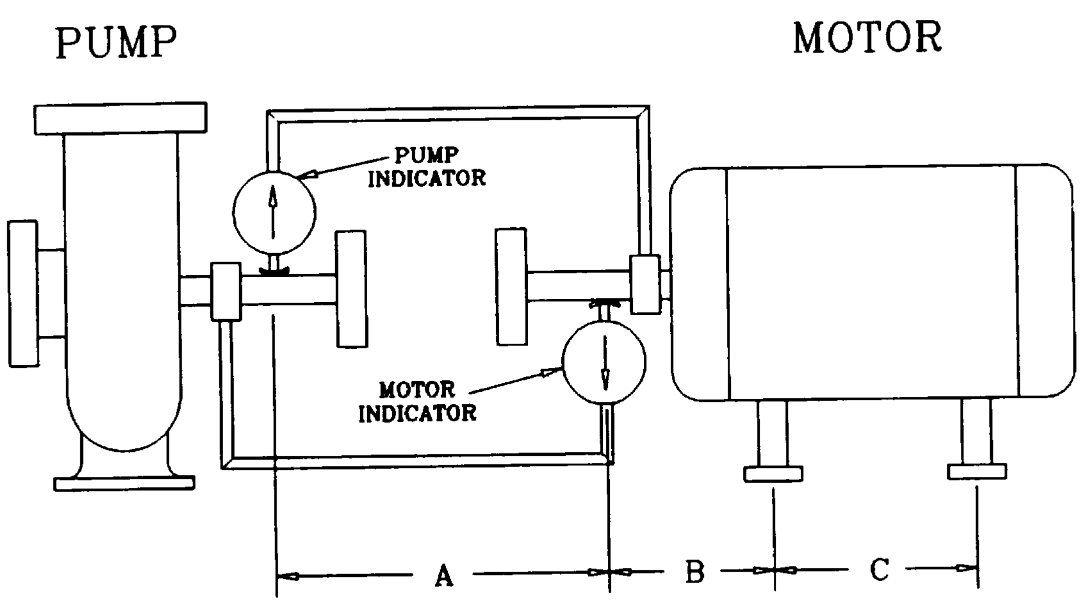


Figure 10. - Reverse indicator alignment worksheet (showing readings added).

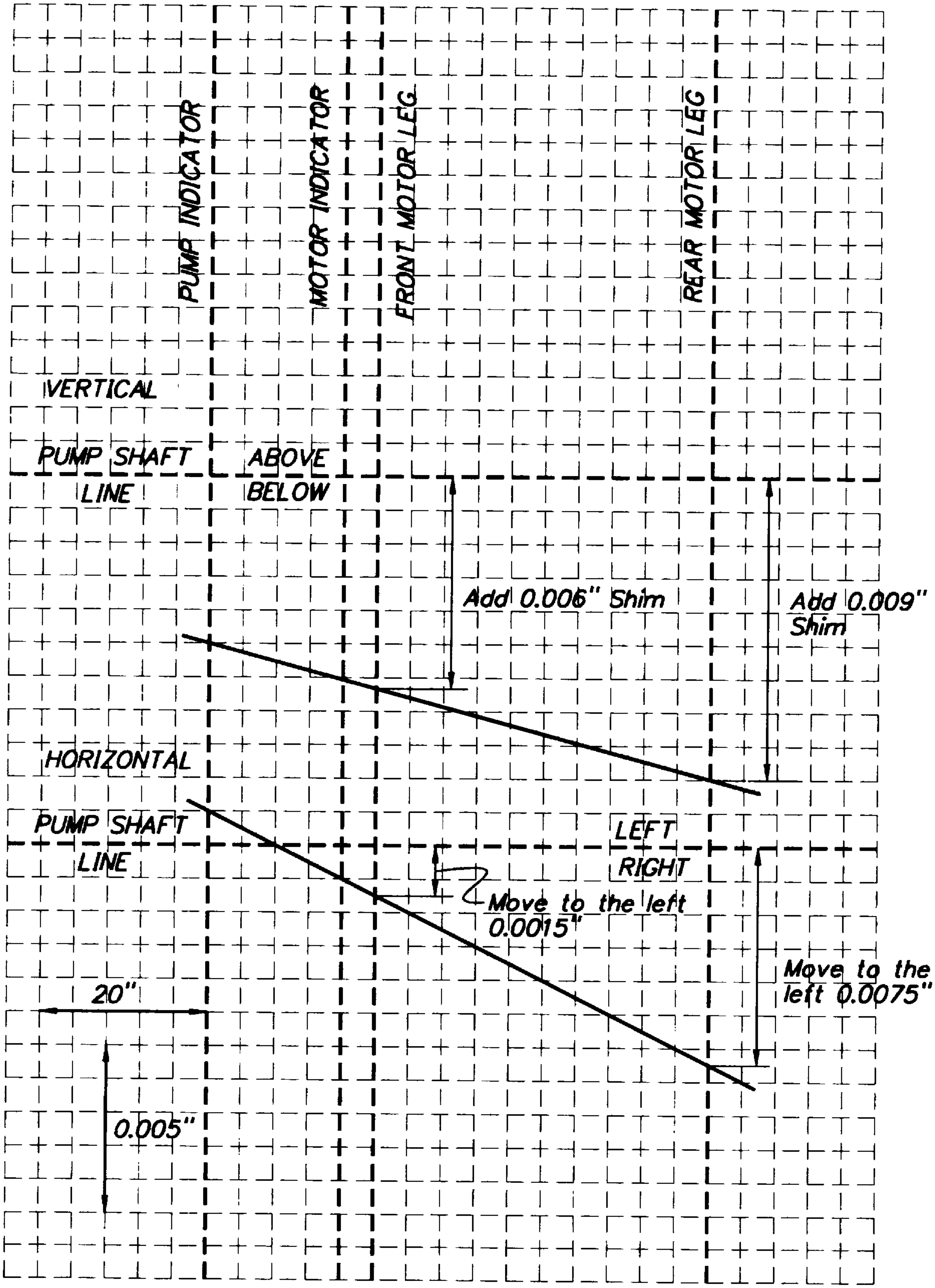


Figure 11. - Reverse indicator alignment graph.

line at the two motor feet lines are the required movements of the motor feet to align the motor to the pump. On the vertical plane plot, these distances represent the required amount of shims to be added or removed. On the horizontal plane, these distances represent the amount of lateral movement required at the motor feet.

(5) After any shimming or movement of the motor, repeat steps a.(1) through b.(4) to verify the alignment.

9. Vibration Monitoring and Analysis

General.–Vibration monitoring and analysis can be a useful part of a preventive maintenance program. There is a variety of vibration monitoring systems available. Some use permanently mounted sensors to continually monitor vibration levels while other systems require readings to be taken periodically with hand-held accelerometers. The maintenance supervisor should compare the potential benefits of a vibration monitoring system, such as preventing major failures and reducing outages, to the overall cost before deciding which system to use or whether to use any system at all.

Proximity probe system.–Proximity probe systems are generally installed to continually monitor shaft runout, alarming, or shutting a unit down before extensive damage occurs to the bearings or other components. A proximity probe is a non-contacting-type sensor which provides a direct-current voltage directly proportional to shaft position relative to the probe. The proximity probe can be looked upon as sort of an electronic dial indicator. A typical proximity probe system normally utilizes two probes per bearing location, radially mounted and 90 degrees apart. The monitors for the probes are centrally located and are provided with relays for alarm and shutdown with continuous indication of shaft runout in mils.

The primary purpose of a proximity probe system is to provide unit protection; but when hooked up to a strip chart recorder or a spectrum analyzer, it can also be useful in vibration analysis or unit balancing. In order to perform vibration analysis, a basic understanding of the characteristics of machine vibration and some knowledge of use of the test equipment is required. Several of the manufacturers of proximity probe systems provide seminars and training for vibration analysis.

Accelerometer systems.–There are a number of hand-held accelerometer-based vibration monitoring systems available varying greatly in complexity and capability. Accelerometers are lightweight vibration sensors that, as the name implies, provide an electrical output proportional to the acceleration of the vibration of the machine being checked. The method readings are taken depends on the design of the accelerometer. Some require holding a probe against the bearing housing or shaft while the reading is being taken while others use a magnet to hold the accelerometer in place.

An accelerometer system requires periodic readings to be taken at different points on each machine. The data from these readings are stored in a portable recording instrument or plotted directly on what is known as a signature card. The data in the recording instrument, in many systems, can be downloaded into a personal computer. The data can then be manipulated in various manners to compare it to data from previous readings at the same points, to determine if there is any increase in the vibration levels indicating an impending failure.

The data on the signature card, and in many cases in the computer comparison, are in the form of a spectrum plot. A spectrum plot is an X-Y plot with the X-axis representing the vibration frequency, usually in cycles per minute or cycles per second (Hertz); and the Y-axis represents vibration amplitude either in acceleration, velocity, or displacement. A spectrum plot features amplitude spikes or peaks corresponding to operating frequencies of components of the equipment being tested. The initial plot provides a "signature" of the vibration for that particular piece of equipment. An increase in the amplitude of vibration at any of the various frequencies in subsequent plots may indicate an impending failure. By determining what component operates at the frequency corresponding to the amplitude peak, corrective action can be taken.

An accelerometer system is most useful when there is a relatively large number of rotating machines at a particular site or if one maintenance crew is maintaining several sites. Regardless of the number of machines involved, in order to fully utilize such a system, someone must be available to analyze the data. This analysis is not necessarily complicated, but it will take some time; and if maintenance personnel are already stretched thin, the vibration analysis may get neglected.

10. Lubrication

General.–The primary purpose of a lubricant is to reduce friction and wear between two moving surfaces, but a lubricant also acts as a coolant, prevents corrosion, and seals out dirt and other contaminants. In order for a lubricant to perform as intended, careful attention must be given to its selection and application as well as its condition while in use. FIST Volume 2-4, Lubrication of Powerplant Equipment, provides more information on lubricants and their use. The equipment manufacturer should provide specific information on the type lubricant and periodic maintenance recommended for a particular application.

Oil.–Oil lubrication can take many forms, from a simple squirt oilcan to a complex circulating system. Regardless of the method by which the oil is applied, the intent is the same, and that is to keep a lubricant film between moving surfaces. For successful lubrication, it is critical that the proper oil be chosen, properly applied, and kept clean and uncontaminated.

While it is beneficial to have as few types of oil in stock as possible, there is no one all-purpose oil that can be used in all applications. Various additives such as emulsifiers, rust and corrosion preventers, detergent, and dispersants are added to oils to enhance their performance for a given application. Characteristics that may be desirable in one case, may be very undesirable in another. For example, emulsifiers added to motor oil allow the oil to hold any water in an emulsion until the engine's heat can boil it away. In bearing lubrication, where there is not sufficient heat to evaporate the water, the oil must be capable of readily separating from water.

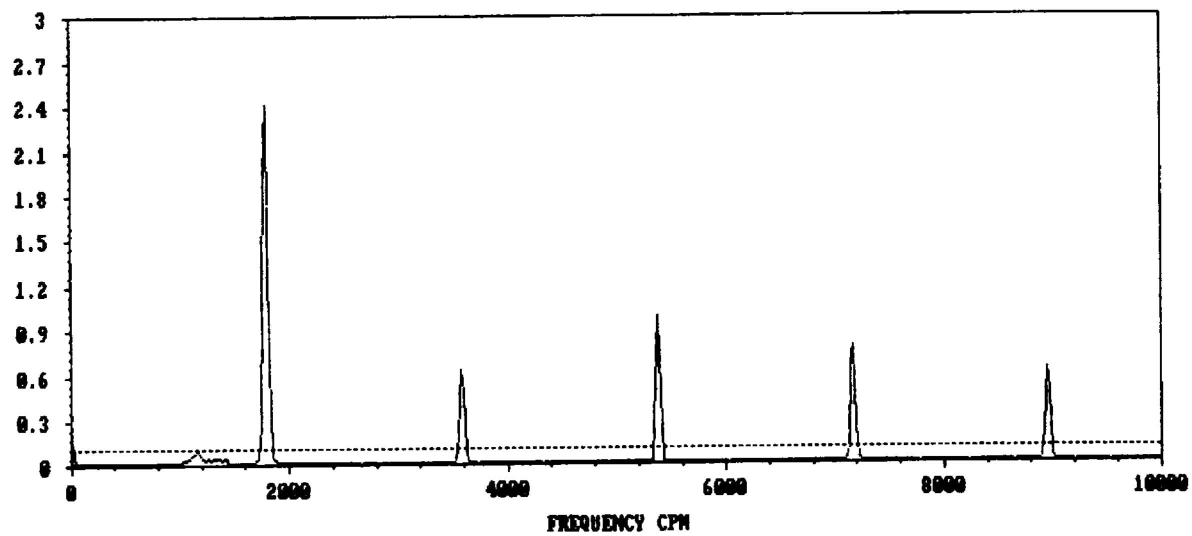
Cleanliness is also extremely important. All seals should be installed and in good condition. Dirt, water, or other contaminants not only can cause premature wear of the bearings, they also can cause the depletion of some of the oil's additives. Samples of the oil

24-JUN-91 **Palomar Technology** **Page 1**

SINGLE SPECTRUM PLOT

SET: POWER GENERATION TYPE: FFT DATE: 08-NOV-90 13:01:32
POINT ID: TEST DESC:
WINDOW: HANNING LINES: 400 AVER: 1 FREQ: 0 - 10000 CPM
DETECT: PEAK TO PEAK RPM: 1800 THRESHOLD: 0.1000 UNITS: mils

FREQ: 0.00 AMP: 0 ORDER: 0.000 DEG: ---



IDENTIFICATION OF SPECTRAL PEAKS ABOVE THRESHOLD

NO.	AMP.	FREQ.	ORDER	NO.	AMP.	FREQ.	ORDER
1.	0.1349	25.0	0.014				
2.	2.3978	1800.0	1.000				
3.	0.6228	3575.0	1.986				
4.	0.9849	5375.0	2.986				
5.	0.7759	7175.0	3.986				
6.	0.6270	8950.0	4.972				

SPECTRAL ENERGY SUMMARY

OVERALL	SYNC	SUBSYNC	NONSYNC
3.049	2.908	0.2776	0.8761

Figure 12. – Spectrum plot.

from large bearings should be periodically tested for viscosity, acidity, water content, and the presence and identity of any foreign material. There may be local laboratories that can perform the tests; but if possible, the oil's manufacturer should perform the tests. The manufacturer can usually perform the tests mentioned and also determine whether any of the additives have been depleted. In any testing program, it is important to keep complete and accurate records of the tests. A significant change in any of the oil's characteristics from previous tests may indicate a problem, although the oil may still be acceptable for service.

The oil from large bearings should be periodically drained and filtered; and the oil reservoir should be cleaned thoroughly. The frequency of filtering should be based on experience or the results of the oil tests. Filtering more frequently than is necessary is a waste of time, while waiting too long to filter the oil will shorten the oil's life and damage the equipment being lubricated.

The oil from small bearings should be periodically drained, and the reservoir or case should be cleaned and filled with new oil. Care should be taken when filling a bearing oil reservoir so as not to under- or overfill. In many cases, overfilling an oil reservoir can cause as much damage as an underfilling.

Another possible source of contamination is the mixing of incompatible oils. Different types of oils or even similar oils from different manufacturers should never be mixed. Additives in different oils may not be compatible and when mixed, may have an adverse reaction, rendering the additives, and possibly the oil itself useless.

Grease.–Grease is a lubricant consisting of a lubricating oil combined with a thickening agent. The base oil makes up, depending on the grease, 85 to 95 percent of the grease and performs the actual lubrication. The thickening agent, usually some type of soap, determines many of the characteristics of a grease such as, heat resistance, water resistance, and cold weather pumpability. Various additives may also be added to improve performance.

Overheating and subsequent failure of grease-lubricated bearings caused by over-lubrication is a common problem. The idea that more is better coupled with the fact that it usually is difficult to determine the actual amount of grease in a bearing housing, causes many bearings to be "over-greased."

Ideally, a grease-lubricated bearing should be "packed" by hand so that the bearing housing is approximately one-third full of grease. When grease is applied using a grease gun, the relief plug, if so equipped, should be removed so that as the new grease is applied, all of the old grease is purged from the bearing housing. The unit should be operated approximately 30 minutes before the plug is replaced to allow excess grease to escape. If the bearing housing does not have a relief plug, grease should be added very infrequently to prevent over-lubrication.

Many of the soap bases used in making grease are incompatible. Mixing two different types of grease will many times result in a mixture inferior to both of the component greases. As a general rule, different greases should not be mixed. If it becomes necessary to change the type of grease used on a piece of equipment, the bearing housing should be completely disassembled and thoroughly cleaned to remove all the old grease. If this cannot be accomplished, as much of the old grease as possible should be flushed out by the new grease during the initial application; and the greasing frequency should be increased until it is determined that all of the old grease has been purged from the system.

In wicket gate greasing systems, a grease must be chosen that is water resistant, is somewhat adhesive, and has extreme pressure characteristics, as well as being pumpable. A grease that is impervious to water and has excellent lubricating qualities is useless if it does not get to the bearing. The consistency must be thin enough to be pumped through the grease lines, but thick enough to stay in the bearings once it is there. Some compromise in the desired qualities is required to obtain a workable grease.

11. Inspection Checklist

Items of inspection	
Hydraulic turbines and large pumps	
a. *Runner or impeller*	*A NS*
b. *Spiral case and draft tube or pump casing and suction inlet*	*A NS*
c. *Wearing rings*	*A NS*
d. *Main shaft packing*	*W NS*
e. *Mechanical seals*	*W NS*
f. *Wicket gates and facing plates*	*A NS*
g. *Servomotors, shift ring, and wicket gate linkage*	*A NS*
h. *Bearings*	*D A NS*
i. *Shaft and coupling*	*A*
j. *Generator or motor rotor*	*A*
k. *Air coolers*	*A*
l. *Unit brakes*	*A*
m. *Inspection reports*	*A*
Auxiliary pumps	
n. *Pump impeller or rotor and casing*	*A NS*
o. *Shaft and coupling*	*W A*
p. *Packing*	*W NS*
q. *Mechanical seals*	*W*
r. *Bearings*	*W*
s. *Pressure-relief valves*	*A*
t. *Eductors*	*NS*

D - *Daily inspection.*
W - *Weekly inspection.*
A - *Annual inspection.*
NS - *Not scheduled (extraordinary maintenance; usually 5-year or longer intervals).*

Hydraulic turbines and large pumps

a. *Runner or impeller.-*

Annual inspection.-Examine runner or impeller thoroughly for cavitation or other damage. Use a nondestructive test to check for cracks in runner buckets or impeller vanes. Refer to FIST Volume 2-5, Turbine Repair, for repair recommendations and techniques.

Not scheduled.-Remove runner or impeller and inspect and repair areas not normally accessible.

b. *Spiral case and draft tube or pump casing and suction inlet.*-

Annual inspection.-Check condition of interior coating and repair as required. Weld repair cavitation damaged areas. Inspect riveted and welded joints for leaks and corrosion and repair as required. Check mandoors for leaks and condition of door hinges. The draft tube or suction tube liner should be checked for voids between the liner and the concrete and grouted if necessary. Any leaks between the concrete and the spiral case, pump casing, draft tube, or suction tube should be monitored; and if excessive or if an increase is noted, the source of the leak should be found and repaired.

Not scheduled.-If condition of interior coating is such that spot repairs are no longer effective, sandblast and repaint entire surface. Draft tube liners severely damaged by cavitation may be repaired by cutting out the damaged area and welding stainless steel plates in place that have been rolled to the proper diameter.

c. *Wearing rings.*-

Annual inspection.-Check top and bottom wearing ring clearances at four points, 90 degrees apart.

Not scheduled.-Remove runner or impeller and replace or renew wearing rings when clearance exceeds 200 percent of design clearance. Wearing rings that are an integral part of the runner or impeller, or the casing in some cases, may be built up by welding and be remachined. Replaceable wearing rings, in most cases, should not be built up by welding, as the heat of welding can induce stresses or distort the rings. If the wearing rings are replaced, the stationary rings should be supplied with an undersized inner diameter and bored concentric to the center of the unit.

d. *Main shaft packing.*-

Weekly inspection.-Check flow and pressure of packing cooling water. Check for excessive heat and for leakage past the packing. Tighten the packing gland as leakage becomes excessive and grease the packing box if and when required.

Not scheduled.-Remove old packing and lantern ring and thoroughly clean packing box. Check packing sleeve for excessive wear and repair as required. Install new packing, staggering adjacent rings so that joints do not coincide.

e. *Mechanical seals.*-

Weekly inspection.-Check for excessive leakage. Properly installed mechanical seals should require very little attention. Follow manufacturer's recommendations for lubrication during extended outages. When excessive leakage does occur, it normally is an indication that new seals are required.

Not scheduled.-Disassemble seal and thoroughly clean seal components and shaft sleeve. Check shaft sleeve for excessive wear and repair as required. Replace segments or other components as required.

f. *Wicket gates and facing plates.*-

Annual inspection.-Measure clearance between gates at the top, middle, and bottom with feeler gauges with gates closed and the servomotor pressure released. Check clearance between wicket gates and upper and lower facing plates. Check gates and facing plates for cavitation damage, corrosion, or other damage. Repair or repaint as required. Check leakage past packing and tighten as required.

Not scheduled.-Disassemble and check wicket gate bushings, thrust washers, stems, and packing sleeve for wear or corrosion. If bushings are out of tolerance, replace and line bore, making sure the bushings are bored concentric and plumb. Check upper and lower facing plates for scoring, corrosion, or other damage and repair as required. Take measurements to verify that facing plates are level and parallel to one another. If necessary, replace facing plates or machine existing plates level and parallel to one another. Measure height of wicket gate and compare it to the distance between the upper and lower facing plates. If out of tolerance, build up wicket gate ends and machine back to specified dimensions. Check gate-to-gate sealing surfaces and build up and remachine as required. Replace shaft packing.

g. *Servomotors, shift ring, and wicket gate linkage.*-

Annual inspection.-Observe servomotor, shift ring, and wicket gate linkage as they are moved through full range of motion in both directions. Look for any lateral movement of shift ring indicating worn bearing pads and for any backlash in the wicket gate linkage. Check for leakage past servomotor packing glands and tighten as required. Check servomotor shaft for scoring and repair or schedule repairs as required. Check amount of squeeze on the wicket gates. Remove at least 10 percent of the wicket gate shear pins and visually inspect for any signs of fatigue cracking. If cracking is evident, inspect remaining shear pins and replace any that are questionable. Keep records of when shear pins are replaced.

Not scheduled.-Disassemble and check condition of shift ring bearing pads and wicket gate linkage bushings and pins. Replace if out of tolerance. Disassemble servomotors and check pistons and cylinder for scoring or signs of misalignment and realign as required. Bore or polish scored cylinder and renew piston by sleeving or other method. Replace piston rings. Check servomotor shaft for scoring and repair by machining and hard chrome plating or other method if necessary. Replace packing as required. Set squeeze on wicket gates by procedure in FIST Volume 2-3, Mechanical Governors for Hydroelectric Units.

h. *Bearings.*-

Daily inspection.-Check the bearing temperature and lubricant level. Check flow and pressure of cooling water. Check flow and pressure of turbine guide bearing oil pump.