**S9086-RH-STM-010**

**0910-LP-108-3282** **REVISION 4**

# NAVAL SHIP'S TECHNICAL MANUAL

# CHAPTER 503

# PUMPS

SUPERSEDES CHAPTER 503 OF 31 DEC 2000

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## REVISION RECORD

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# CHAPTER 503

# PUMPS

## SECTION 1

## SAFETY

### 503-1.1 SAFETY PRECAUTIONS

Operation of centrifugal and positive displacement pumps can be hazardous to both personnel and equipment unless appropriate safety precautions are taken. The general safety precautions related to the operation, maintenance and repair of centrifugal, rotary and reciprocating pumps are listed in the beginning of Section 2, Section 3, and Section 4, respectively. These safety precautions are generic in nature and should not be considered all encompassing. A thorough knowledge of and adherence to the requirements of those documents cited in paragraph 503-1.1.1 are required to prevent personnel injury or equipment damage.

503-1.1.1 Safety References:

a. General safety requirements and precautions can be located in the following references (Listed in order of precedence):
   1. Navy Safety Precautions for Forces Afloat (OPNAVINST 5200.19)
   2. General Specifications For Ships Of The United States Navy, Section 503 - Pumps
   3. General Specifications for Overhaul of Surface Ships (GSO),S9AAO-AB-GOS-010, Section 503 - Pumps
b. Specific safety precautions related to the operation, maintenance, and repair of pumps can be found in the following documents (Listed in the order of precedence):
   1. Operation
      (a) Engineering Operating Sequencing System (EOSS) Equipment Operating Procedures (EOPs)
      (b) Equipment Technical Manual (T/M)
   2. Maintenance and repair:
      (a) Preventive Maintenance System (PMS) Maintenance Requirement Cards (MRCs)
      (b) Technical Repair Standard (TRS)
      (c) Technical Manual (T/M)

# SECTION 2

# CENTRIFUGAL PUMPS

## 503-2.1 CENTRIFUGAL PUMP SAFETY PRECAUTIONS.

The following safety measures must be followed exactly to minimize hazards to personnel and equipment while operating centrifugal pumps:

a. If relief valves are fitted, ensure that they function at designated pressure.

b. Ensure that the steam valve to the turbine is closed when attempting to jack a pump by hand.

c. Boiler feed system pumps shall not be used for purposes other than those connected with boiler or feedwater service.

d. Do not tie down or otherwise render inoperative the overspeed trip, speed-limiting, or speed-regulating governor.

e. Ensure that overspeed trips, where fitted, are set to shut off steam to the unit when rated speed is exceeded by 10 percent.

f. Ensure that speed-limiting and speed-regulating governors are set to limit pump speed to rated speed under rated conditions, and that rated speed is not exceeded by more than 5 percent for any loading conditions.

g. Overspeed trip and speed-limiting governor settings shall be verified by forces afloat as authorized by NAVSEA, using validated PMS procedures.

h. When inspecting or overhauling vertical centrifugal pumps, never rely on a flexible coupling to support the total pump rotor weight. Always support the rotors with wire rope slings, block and tackle, or chocks before working on the pump. Pads or wrappings shall be used to protect the shafts or shaft sleeve areas of the supporting devices.

i. Before opening the steam end of turbine-driven pumps, ensure that drains are open and that steam and exhaust root valves are wired shut. If double valve protection is provided, wire both valves shut.

## 503-2.2 GENERAL DESCRIPTION.

503-2.2.1 Centrifugal pumps are used for the majority of nonviscous liquid services on ships. Advantages of a centrifugal pump include simplicity, compactness, weight saving, and adaptability to the highspeed prime mover. Disadvantages are lack of suction lift (until the pump has been primed), and sensitivity to variations in head and speed.

503-2.2.2 Centrifugal pumps are usually designed for a specific set of operating conditions (design point) and may perform unsatisfactorily when subjected to conditions that vary significantly from the design point. The principles of centrifugal pump operation, suitable application, and limitations of the various types, therefore, must be understood.

## 503-2.3 CENTRIFUGAL PUMP TERMINOLOGY.

503-2.3.1 The following terminology applies to centrifugal pumps:

a. Bearing  A bearing is a part which supports or positions the shafts on which a rotor is mounted. A bearing may be internal (bearing being lubricated by the liquid being pumped) or external (bearing isolated from the liquid being pumped). The bearings may be either an anti-friction (roller or ball bearing) or fluid film type (sleeve and journal).

b. Bearing Bushing  The removable portion of a sleeve bearing in contact with the journal.

c. Bearing Housing  A body in which the bearing is mounted.

d. Casing  The portion of the pump which includes the impeller chamber and volute or diffuser.

e. Casing Wearing Ring  A stationary replaceable ring to protect the casing at a running fit with the impeller. Wearing rings provide an easily and economically renewable leakage joint between the impeller and the casing. Wear occurs due to pressure differential across the leakage joint.

f. Cyclone Separator  A mechanical device designed to eliminate foreign matter from the seawater (seal flushing water) supplied to the mechanical seal faces. The device is integrally connected to the pump through tubing and fittings on the suction and discharge piping and on the stuffing box.

g. Deflector (Flinger)  A flange or collar around a shaft which rotates with it to prevent passage of liquid, grease, oil, or heat along the shaft.

h. Diffuser  A piece, adjacent to the impeller exit, which has multiple passages of increasing area for converting velocity to pressure.

i. Gland  A follower which compresses packing in a stuffing box or retains the stationary element of a mechanical seal.

j. Grease Cup  A receptacle for containing and supplying lubricant.

k. Impeller  The bladed member of the rotating assembly of the pump which imparts the centrifugal force to the liquid being pumped.

l. Lantern Ring  An annular device used to establish a liquid seal around a shaft, to lubricate the stuffing box packing and to prevent the leakage of air into the stuffing box during suction lift conditions.

m. Mechanical Seal  A mechanical device located in the pump stuffing box, consisting of a stationary element and a rotating element, each with a smooth flat face that prevents the flow of a liquid or gas into or out of the pump casing. Details regarding the operation, installation, leakage, and testing are provided in Section 5.

n. Nozzle  The portion of the pump casing between the pump and piping connection that is designed to convert fluid velocity to pressure.

o. Packing  A pliable lubricated material used to provide a seal around that portion of the shaft located in the stuffing box.

p. Polymeric Compound  Polymer-based material consisting of a thermosetting base (usually epoxy) with an addition of metal, ceramic, or other material.

q. Shaft Coupling  A mechanism used to transmit power from the drive shaft to the pump shaft or to connect two pieces of shafting.

r. Shaft Sleeve  A cylindrical piece fitted over the shaft to protect the shaft through the stuffing box, and which may also serve to locate the impeller on the shaft.

s. Specific Speed  A dimensionless number which is defined as the number of revolutions per minute at which a geometrically similar impeller would run if it were of such a size as to discharge one gallon per minute against one foot of head.

t. Stuffing Box  A portion of the pump casing through which the shaft extends and in which a mechanical seal or packing and a gland may be placed to prevent leakage.

u. Stuffing Box, Auxiliary  Also called a ″Back-up″ Stuffing Box. A recessed portion of the gland and cover of a mechanical seal sub-assembly designed to accommodate two or more rings of emergency backup packing.

v. Throat Bushing  A renewable lining for a pump casing at a shaft penetration or throat. Its purpose is to: provide a solid surface against which packing seats, provide a fairly tight clearance to the rotating shaft sleeve so the packing doesn't extrude into the fluid system through the bushing-to-shaft sleeve gap, and be replaceable in case the tight clearance opens up due to wear.

## 503-2.4 CENTRIFUGAL PUMP TYPES.

503-2.4.1 Centrifugal pumps are most commonly typed by their general mechanical configuration as follows ( Figure 503-2-1):

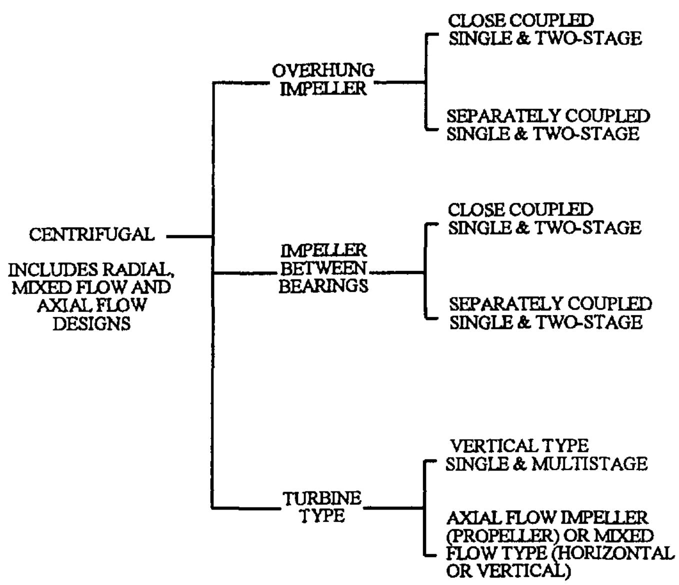



Figure 503-2-1. Centrifugal Pump Types by Mechanical Configuration.

a. Overhung Impeller Type  The impeller (or impellers) is mounted on the end of a shaft which is cantilevered or overhung from its bearing supports. These pumps are further categorized as either close-coupled, where the impeller is mounted directly on the driver shaft; or separately coupled where the impeller is mounted on a separate pump shaft supported by its own bearings.

b. Impeller Between the Bearings Type  The impeller (or impellers) is mounted on a shaft with bearings at both ends. The impeller is mounted between the bearings. These pumps are further categorized into single-stage and multistage configurations.

c. Turbine Type  Pumps which are built mostly with internal, liquid lubricated bearings and diffuser casings, which are convenient for multistage construction, and which discharge through a supporting column pipe.

503-2.4.2 Centrifugal pumps are also grouped by the hydraulic geometry of their impeller designs as follows ( Figure 503-2-2):

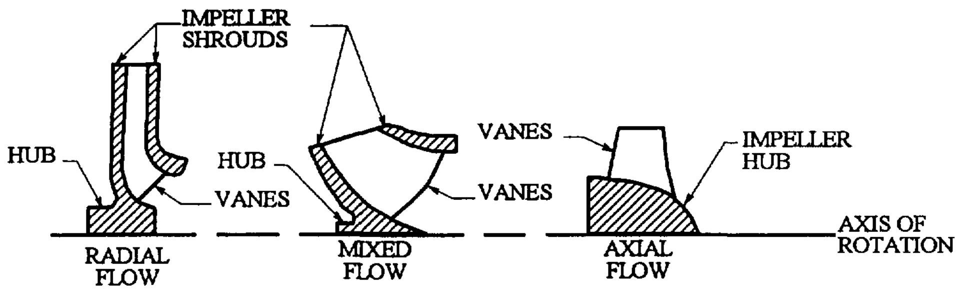



Figure 503-2-2. Hydraulic Geometry of Centrifugal Pump Impellers.

a. Radial Flow The liquid enters the impeller axially at the hub and flows radially to the periphery. This type of impeller develops pressure principally by the action of centrifugal force. These impellers usually have a specific speed below 4200 for single suction impellers and below 6000 for double suction impellers. Radial flow pumps produce a low flow rate capacity at a high head.

b. Axial Flow (Propeller Pump) A single inlet impeller with the flow entering axially and discharging nearly axially. An axial flow impeller develops most of its head by the propelling or lifting action of the vanes on the liquid. These type impellers usually have a specific speed above 9000. Axial flow pumps produce a high flow rate capacity at a low head.

c. Mixed Flow A single inlet impeller with the flow entering axially and discharging in an axial and radial direction.

503-2.4.3 Specific Speed To better understand pump grouping by their impellers hydraulic geometry, a discussion of specific speed is required. Specific speed is a correlation of pump capacity, head, and speed at optimum efficiency, which classifies the pump impellers with respect to their geometric similarity.

Specific speed is a dimensionless number usually expressed as:

$$N_S = N (Q)^{1/2} / H^{3/4}$$

Where:

$N_s$ = Pump specific speed

N = Rotative speed (revolutions per minute)

Q = Flow (gallons per minute) at optimum efficiency

H = Total head in feet per stage

The specific speed of an impeller is defined as the revolutions per minute at which a geometrically similar impeller would run if it were of such a size as to discharge one gallon per minute against one foot of head.

Specific speed is indicative of the shape and characteristics of an impeller. It has been found that the ratios of major dimensions vary uniformly with specific speed. Specific speed is useful to the designer in predicting proportions required and to the application engineer in checking suction limitations of pumps.

503-2.4.4 Vacuum Pumps. Vacuum pumps are a special case which do not fall within the categories or boundaries of either pumps or air compressors. These pumps are commonly used for the priming of centrifugal pumps and are occasionally used as air ejectors or air compressors. The majority of vacuum pumps are typically of the liquid ring wet vacuum design and are sometimes referred to as centrifugal displacement pumps or rotary-centrifugal compressors. The vacuum pump is covered in this manual to ensure completeness.

The rotor consists of a series of blades projecting from a hollow cylindrical hub through which the shaft has been pressed. The blades are shrouded at the sides and thus form a series of chambers. The pump body (casing) typically has either a circular shape or an elliptical shape. For circular pump casings, the rotor is positioned eccentrically within the circular pump casing.

When the rotor is spinning, seal water, located within the casing of the pump, is forced by the centrifugal action of the rotor blades to follow the internal surface of the casing of the pump. By following the contours of the pump casing, the chamber or pocket formed by the blades, the side shrouding, and the sealing water expands as each chamber moves past the inlet or suction port. The expansion of these chambers creates a low pressure area which allows the air in the suction piping to enter the chamber. As the chambers continue around the contours of the pump casing, the chambers compress as they pass the discharge port. This action results in the sealing water acting as a positive displacement piston which hydraulically compresses the air in the rotor chamber.

503-2.4.5 Canned Motor Pumps. Canned motor pumps are close-coupled pumps with the motor rotor and stator sealed in separate cans. Canned motor pumps are completely sealed to contain all pumped fluid. Therefore, they do not have mechanical seals or packing. The pumped fluid is used to lubricate the bearings and cool the motor. Canned motor pumps are used where containment of the pumped fluid is essential.

## 503-2.5 OPERATION PRINCIPLES.

503-2.5.1 The centrifugal pump imparts energy to a liquid through centrifugal force which is produced by rotating an impeller within a casing. When the liquid in the impeller is forced away from the eye (center) of the impeller, a reduced pressure is produced and consequently more liquid flows. A steady flow through the impeller is produced unless something causes the vacuum at the inlet to be broken, the flow to the impeller eye to be disrupted, or the flow at the discharge to be restricted by a pressure greater than the pressure head-developed by the rotating impeller. The liquid leaves the impeller at a high velocity and is collected by a progressively expanding spiral casing (volute) where the velocity is reduced and converted to pressure head. In a diffusion pump, stationary guide vanes called diffusion vanes surround the impeller. The change of flow direction and conversion of velocity head to pressure head occurs in the diffusion vanes.

503-2.5.2 Diagrammatic sectional views of volute and diffusion centrifugal pumps are shown in Figure 503-2-3 and Figure 503-2-4.

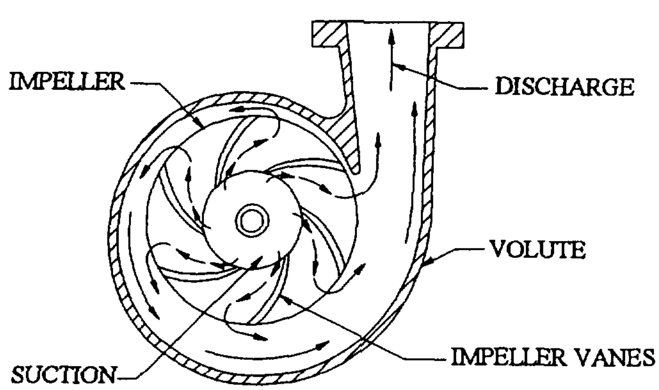



Figure 503-2-3. Volute Centrifugal Pump.

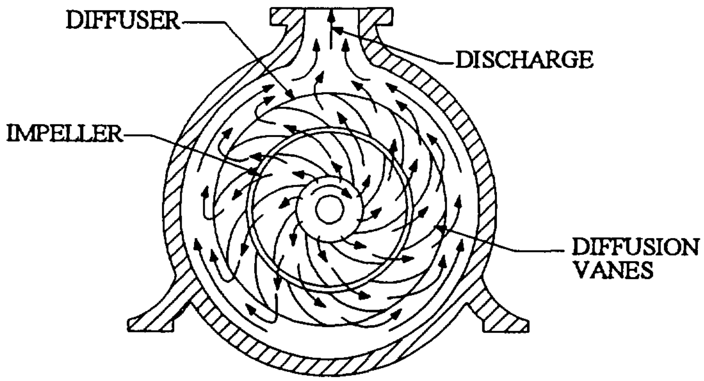



Figure 503-2-4. Diffusion Centrifugal Pump.

## 503-2.6 CLASSIFICATION.

503-2.6.1 GENERAL CLASSIFICATIONS Centrifugal pumps are classified in several ways, including:

a. Service application (refer to paragraph 503-2.6.2).
b. Number of stages.
   1. Single-stage.
   2. Two-stage.
   3. Multistage.
c. Shaft position.
   1. Horizontal.
   2. Vertical.
   3. Inclined.
d. Impeller type.
   1. Single suction or double suction.
   2. Open, semiclosed, or closed.
   3. Radial flow, mixed flow, or axial flow (propeller).
e. Casing type.
   1. Volute, diffuser, or turbine.
   2. Axially split, radially split, or barrel.
f. Piping connection arrangement: side, end, top, or bottom, identified to both suction and discharge connections.
g. Materials: Nodular iron, bronze-fitted, all bronze, steel, stainless steel, titanium, nickel-aluminum-bronze, copper-nickel alloy (70-30), Monel, non-metalic composities, ceramics, and other special alloys.

**NOTE**

**Cast iron and bronze-fitted pumps are seldom used in shipboard service because of shock-proofing problems and rapid corrosion when handling seawater.**

h. Drive type: flexible coupled, rigid coupled, close coupled, or gear- or belt-driven.

i. Other classifications, such as special construction, direction of rotation (or reversible), and kind of power drive.

503-2.6.2 Service Application. The various centrifugal pumps used in ships have requirements, and operation, and installation problems peculiar to each application. The more distinctive pump types are listed in this paragraph according to application. Because ship classes vary, all the services listed may not be found on every ship; also, special machinery plants may require pumps not listed. Following are the primary service applications:

a. Steam Propulsion Systems
   1. Boiler feed pumps
   2. Feed booster pumps
   3. Condensate pumps
   4. Condensate booster pumps
   5. Circulating pumps

b. Fuel Systems
   1. Fuel transfer pump
   2. Cargo fuel pump
   3. JP5 transfer pump
   4. JP-5 service pump
   5. Cargo JP-5 pump

c. Auxiliary Systems
   1. Fire pumps
   2. Seawater service pumps
   3. A/C chill water pumps/cooling water pumps
   4. Distilling plant/W.H.B. pumps
   5. Potable pumps
   6. Bilge pumps

503-2.6.3 General Characteristics. Unlike positive displacement pumps, centrifugal pumps deliver a wide range of flows at relatively constant head (pressure). A centrifugal pump at constant speed will deliver liquid at any capacity from zero to a maximum capacity (determined by pump size, suction condition, and design factors). At constant speed the pressure, efficiency, and power required vary over the range of the pump capacity according to relationships that are expressed by a pump characteristic curve.

A typical pump characteristic curve is shown in Figure 503-2-5.

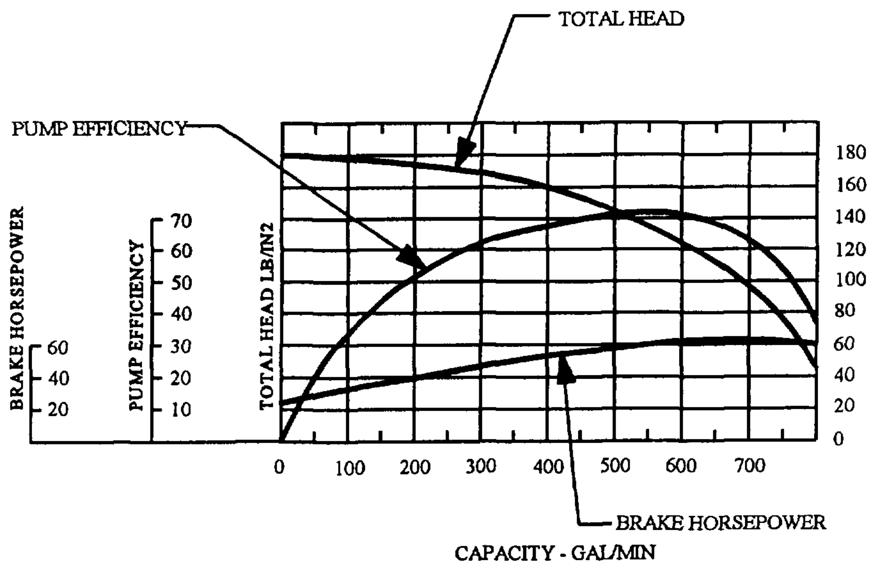



Figure 503-2-5. Typical Centrifugal Pump Characteristic Curve

503-2.6.3.1 In all centrifugal pumps, the affinity laws allow you to project capacity, head, and brake horsepower for small changes in pump speed. For a given pump the affinity laws are:

a. The capacity varies directly as the speed:
$Q_1/Q_2 = N_1/N_2$

b. The total head varies as the square of the speed:
$H_1/H_2 = [N_1/N_2]^2$

c. The brake horsepower varies as the cube of the speed:
$BHP_1/BHP_2 = [N_1/N_2]^3$
where:

Q = capacity (gpm)

H = total head (feet)

BHP = brake horsepower

N = pump speed (rpm)

1 = initial condition

2 = final condition

503-2.6.3.2 The power required to drive any centrifugal pump is determined as follows:

$WHP = Gal/Min \times H \times sg/3960$

where:

WHP = water horsepower output.

Gal/Min = capacity in gallons per minute.

H = total head in feet of liquid.

sg = specific gravity.

or

*WHP= Gal/Min x P/1714*

where:

P = total pressure in lb/in $^2$ ,

and the brake horsepower required = *WHP/pump efficiency*

503-2.6.3.3 Net Positive Suction Head. Net positive suction head (NPSH) is the total suction head (pressure) in feet of liquid measured at the pump centerline or impeller eye minus the vapor pressure (expressed in feet) of the liquid being pumped. Two types of NPSH must be considered in selecting pumps:

a. Available NPSH.

b. Required NPSH

503-2.6.3.3.1 Available NPSH. Available NPSH represents the amount of energy (head) available at the pump suction to get liquid to flow into the pump. In an open piping system, the available NPSH is the sum of atmospheric pressure (in feet of liquid absolute), plus static suction head, (or minus static suction lift) minus suction piping system friction (in feet of liquid), minus the liquid vapor pressure (in feet of liquid absolute), with a correction for the specific gravity of the liquid at the pumped temperature. In a closed system such as a steam condenser or a distilling plant, the pump takes suction from a source under vacuum. The absolute pressure (vacuum) is the vapor pressure of the liquid at the pumped temperature. Under these conditions, the available NPSH is the elevation difference (in feet) between the liquid supply level and the centerline of the pump impeller eye in a vertical plane, minus suction piping system friction (in feet of liquid).

503-2.6.3.3.2 Required NPSH. Required NPSH is determined by the pump manufacturer and depends on the type of impeller inlet, impeller design, pump flow, rotational speed, and nature of the pumped liquid. Required NPSH is usually based on an actual test of nearly identical pumps. For satisfactory operation of the pump, available NPSH must be equal to or greater than the required NPSH throughout the pump operating range to prevent detrimental cavitation.

503-2.6.3.4 Suction Lift. Suction lift exists when the total suction head is below atmospheric pressure. Suction lift is equal to the static lift plus the friction losses in the suction piping. When the liquid supply is above the pump centerline, suction lift may result if the friction losses are greater than the static head. The capability of a pump to operate at suction lift depends on factors such as viscosity, temperature, vapor pressure of the liquid being handled, and the various pump design features, including the suction passages, the impeller design, and the pump speed. If an application requires a high suction lift, pumps that have a larger impeller eye diameter and that run at a slower speed may be chosen. Pump capacity and efficiency are normally unaffected by varying suction conditions until the suction lift exceeds a certain value. At extreme values, the pump may cavitate.

503-2.6.3.5 Cavitation. Pumps cavitate when the absolute static pressure in liquid drops below the liquid's vapor pressure. When this condition exists, vapor bubbles form. As the liquid develops pressure within the impeller, the bubbles collapse. Severe cavitation will remove material from the impeller. Pump cavitation may result from improper pump design, excessive suction lift, improper pump selection for the specific suction conditions, and improper suction piping arrangement. Cavitation results in pump noise and vibration, reduced capacity and efficiency, and pitting of pump parts, especially the impeller. Cavitation damage is characterized by pitting on the back side of the impeller vane and on the inside of the back shroud, downstream of the suction edge of the vane. Where cavitation cannot be avoided, special materials resistant to cavitation pitting must be used for impellers and other parts. Condensate pumps usually operate in cavitation and the relationship between capacity and submergence makes them self-regulating as to capacity. A characteristic curve of a pump that operates almost continuously in cavitation is illustrated in Figure 503-2-6.

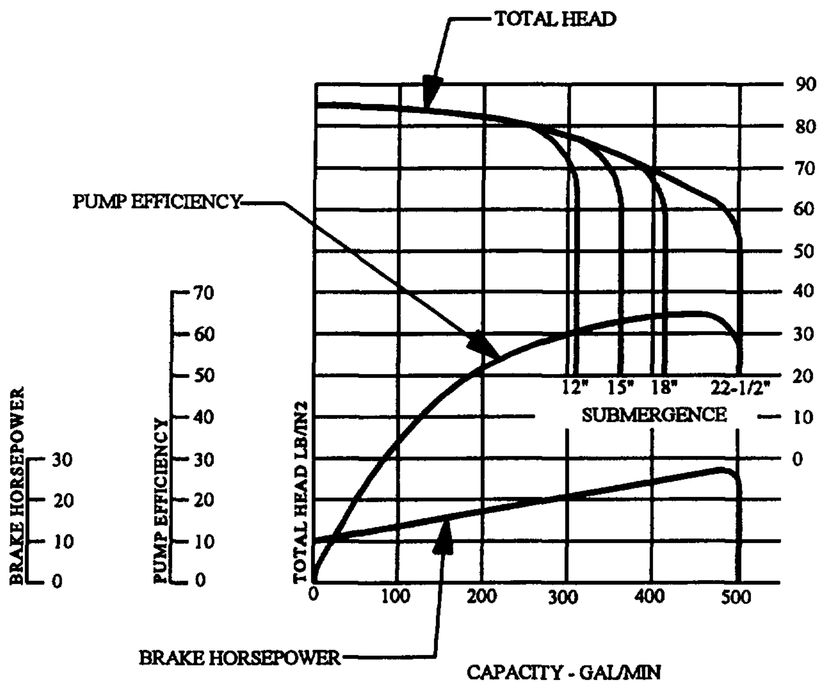



Figure 503-2-6. Characteristic Curve of a Condensate Pump that Operates Almost Continuously in Cavitation.

503-2.6.3.6 Recirculation. A small amount of recirculation occurs at all capacities from the impeller discharge to the suction through the impeller and casing wear ring clearance. Wear will cause the clearance to increase over time and the pump capacity to decrease. Recirculation can also occur in the eye of the impeller when the pump capacity is reduced to about 1/2 to 1/3 of the pump capacity (at the best efficiency point). Recirculation damage is characterized by wear on the inside of the front shroud of the impeller where the vanes begin.

503-2.6.3.7 System Curves. A pump operating in a system must develop a total head which is made up of several parts, each stated in feet:

a. The difference in height between source of supply and the point of delivery.

b. The difference in pressures (if any) on the surface of the supply and on the surface of the delivered liquid.

c. Frictional losses in the piping, valves, and fittings of the system.

d. Entrance and exit losses at the intake and discharge of the system piping.

e. The difference between the velocity head at the pump suction and the velocity head at the pump discharge.

Because the first two parts vary only slightly with flow, or pump capacity, they can be considered together as total static head. The other three parts vary (roughly as the square) with flow, or pump capacity, and can be considered together as friction loss.

If, in a system, the total static head is added to the friction loss and the sum is plotted against capacity, the curve that results is the system-head curve. A system-head curve is shown in Figure 503-2-7.

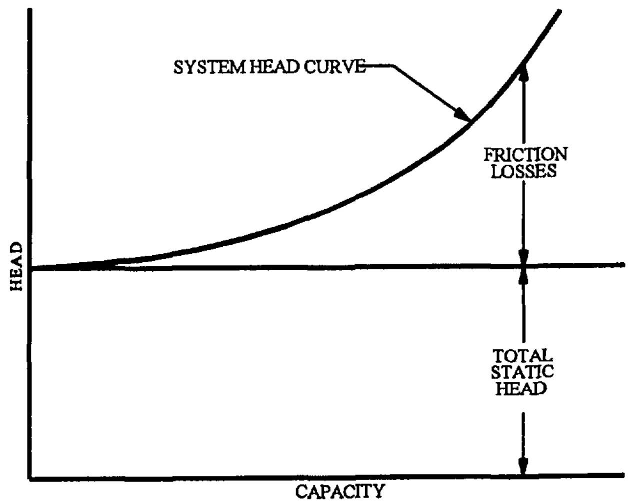



Figure 503-2-7. Friction - Head and Curve System - Head Curve

If the head-capacity curve of a pump, or group of pumps, is plotted together with the system-head curve, the point where the curves cross will be the head and capacity operating point, as shown in Figure 503-2-8.

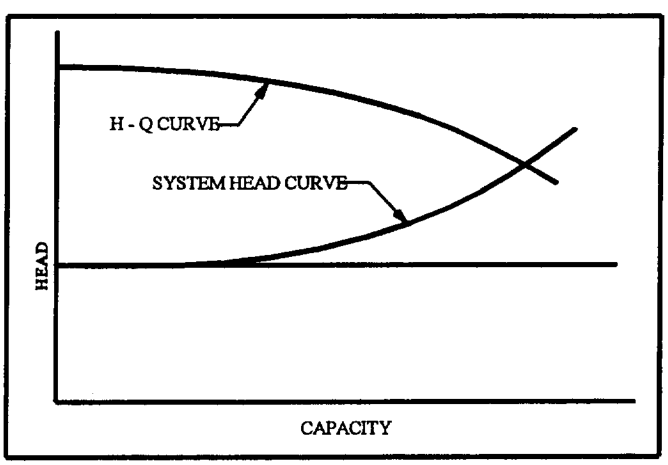



Figure 503-2-8. Pump H-Q Curve Superimposed on System - Head Curve.

System operating conditions can also be set by varying the speed of the pump driver as shown in Figure 503-2-9 or by throttling the discharge of the pump, for example, by adding an orifice in the discharge line. The latter case is shown in Figure 503-2-10.

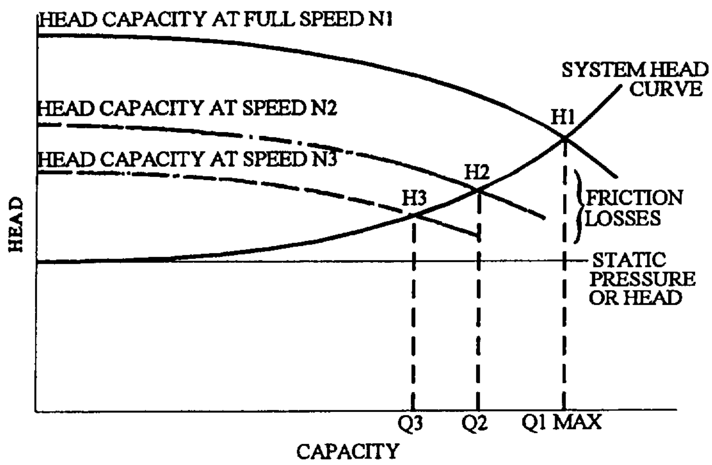



Figure 503-2-9. Varying Pump Capacity by Varying Speed.

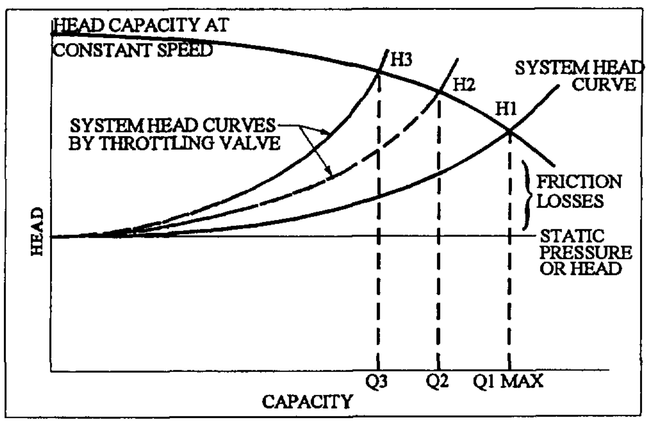



Figure 503-2-10. Varying Pump Capacity by Throttling.

Operating conditions can be varied by operating pumps in series or in parallel. The head-capacity curve or two pumps operated in series is constructed by adding the heads of the individual pumps at any capacity, as shown in Figure 503-2-11. The head-capacity curve of two pumps operated in parallel is constructed by adding the capacities of the individual pumps at any head, as shown in Figure 503-2-12.

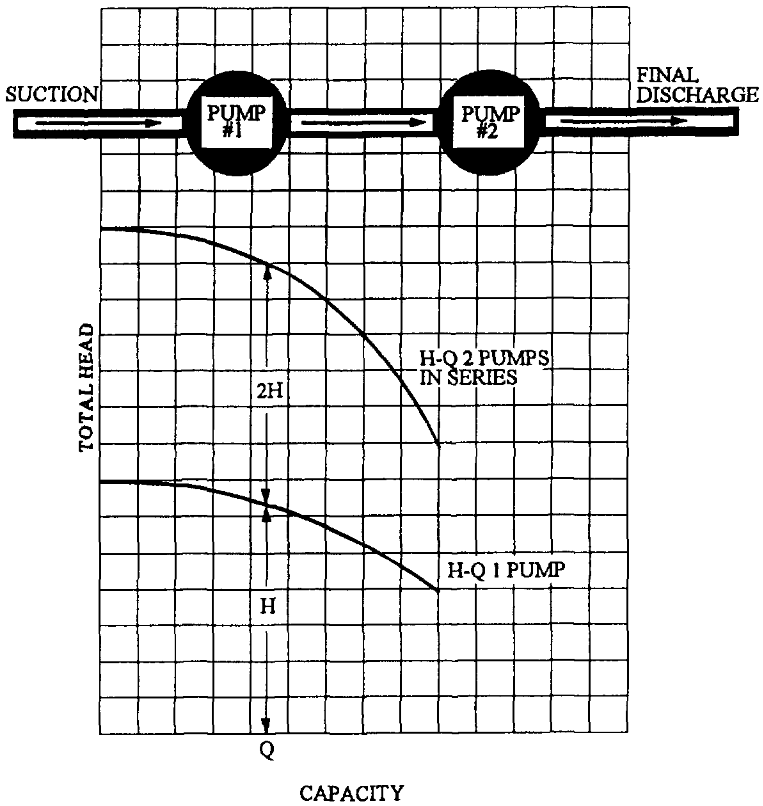



Figure 503-2-11. Operation of Two Pumps in Series

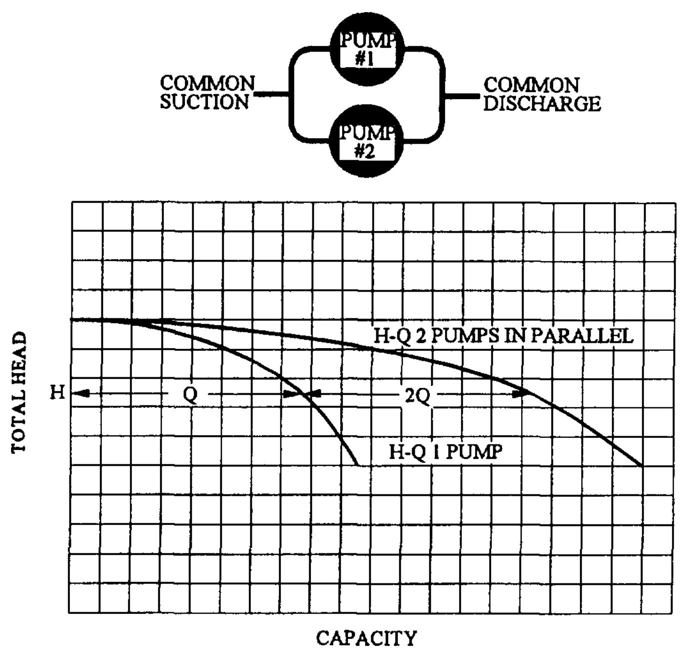



Figure 503-2-12. Operation of Two Pumps in Parallel.

503-2.6.3.8 Series Operation. Centrifugal pumps may be arranged to operate in series when the required pressure exceeds that of one available pump, or where it is necessary to install a booster pump between the supply source and the service pump. Examples of series-operated pumps include potable water booster pumps on larger ships, diesel engine circulating water pumps (in which a booster pump is located below the water line) to supply the engine-attached pump on a higher deck, and feed booster pumps installed between deaerating feed tanks and feed pumps. Important principles for such installations are:

a. The booster pump (acting as the first stage) should have the same or slightly greater capacity rating and a lower required NPSH than the pump it feeds. The booster pump should always be started first and secured last.

b. Operating conditions can be varied by operating pumps in series or in parallel. The head-capacity curve or two pumps operated in series is constructed by adding the heads of the individual pumps at any capacity, as shown in Figure 503-2-11.

503-2.6.3.9 Parallel Operation. Pumps may be installed to operate in parallel to handle varying capacity demands over a wide range or when maximum reliability is required. Examples are main and auxiliary condensate pumps, feed booster pumps, fire pumps, and boiler feed pumps. Operation of several pumps, in parallel, at very low capacities should be avoided to prevent overheating. Pumps should have constantly rising head from rated capacity to shutoff for satisfactory and stable operation in parallel. Where pumps of different capacity ratings are to operate in parallel (such as main and auxiliary condensate pumps), the shutoff head of the smaller pump should be slightly in excess of the shutoff head of the larger pump. This is necessary to ensure that the smaller units are not blocked off the line by the higher pressure developed by one of the larger pumps with which it may be paralleling.

a. The head-capacity curve of two pumps operated in parallel is constructed by adding the capacities of the individual pumps at any head, as shown in Figure 503-2-12.

503-2.6.4 Applications. The following paragraphs describe various centrifugal pump applications.

503-2.6.4.1 Main Feed Pumps. Feed pumps usually are the high speed, multistage type driven by steam turbines. Feed pumps may be horizontal or vertical, and steam turbine or motor-driven.

503-2.6.4.1.1 Different types of feed systems in use are described in Chapter 220 Vol 2, Boiler Water/Feedwater, Test and Treatment. Feed pumps for this system handle water at 107°C to 121°C (225°F to 250°F) with suction pressures of approximately 50 $lb/in^2$ g. This is equivalent to an available NPSH of 142 feet; the pump's required NPSH ranges from 42 to 65 feet. Suction pressure is usually supplied by a feed booster pump because the high speed feed pumps would not have sufficient available NPSH. In some auxiliary ships the de-aerating feed tank (DFT) is located at a height above the feed pump sufficient to ensure adequate suction head without a feed booster pump. Boiler feed pumps require an NPSH that is sufficient to take care of momentary or sudden increases in pump capacity.

503-2.6.4.1.2 Feed pumps require recirculation from discharge to supply source, preferably the feed tanks or de-aerating feed tank, to protect the pump from overheating when operating at low capacity. To withstand the corrosive action of such feedwater, boiler feed pumps are constructed of corrosion-resisting chrome or chrome-nickel alloy steel.

503-2.6.4.2 Feed Booster Pumps. Feed booster pumps are usually vertical, single- or two-stage pumps driven by a steam turbine or electric motor. Their construction is similar to condensate pumps. Feed booster pumps are designed to operate with the same or slightly greater capacity rating and a lower required NPSH than the feed pump. Feed booster pumps take suction from the deaerating feed tank with 6 to 12 feet of static suction head which is about equal to the available NPSH. These pumps are fitted with suction vent connections and a recirculation line back to the feed heater. The recirculation line should be left open when starting or securing and when the pump is discharging to a reciprocating emergency feed or main feed pump. At other times, the recirculation from the centrifugal main feed pump will give the booster pump sufficient protection against overheating at low capacities.

503-2.6.4.3 Condensate Pumps. Main condensate and SSTG (auxiliary) condensate pumps are vertical, single- or two-stage pumps with motor or turbine drives. They are usually installed in parallel. Condensate pump suction piping is arranged with a continuous downward slope from the condenser to the suction nozzle to minimize cavitation and dry operation. Condensate pump elevations are such as to ensure impeller submergence under all conditions of ship pitch or roll. In addition, the condensate pump first-stage impeller is vented back to the condenser. Figure 503-2-13 shows a typical two-stage condensate pump.

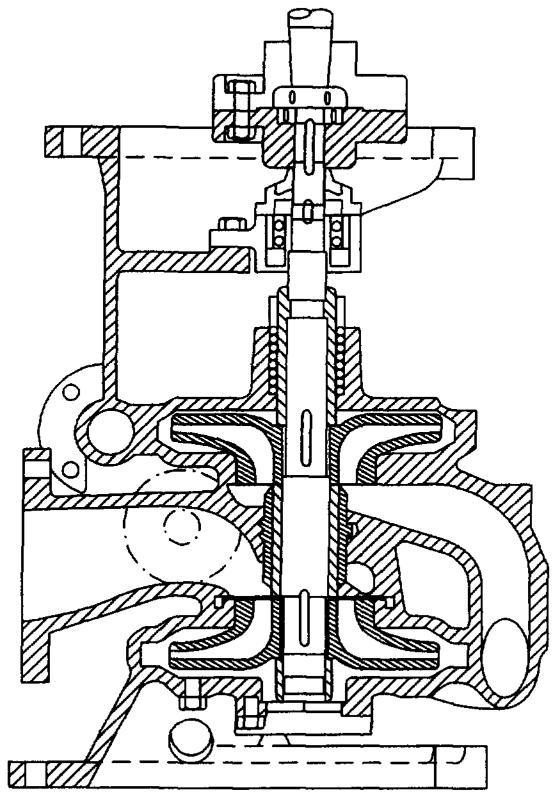


Figure 503-2-13. Typical Vertical Two-Stage Condensate Pump.

503-2.6.4.3.1 Condensate pumps take suction from condenser hot wells under extremely low NPSH conditions. Condensate pumps should be operated with nearly continuous recirculation at low capacities because their low NPSH available, over that required, is insufficient to withstand a temperature rise of more than a degree or two without cavitating. First-stage impellers of older condensate pumps are usually of a top suction design, while newer condensate pumps have a bottom suction design to maximize available NPSH.

503-2.6.4.3.2 Sealing water for condensate pump stuffing boxes is taken from a separate supply source to ensure a continuous clean water supply under positive pressure, preventing air leakage into the stuffing box.

503-2.6.4.4 Main Condenser Circulating Pumps. Pumps for main circulating service are usually the vertical, propeller type driven by direct-connected electric motors or geared steam turbines. Their capacities vary up to 40,000 gal/min at total heads of approximately 30 feet. Main condenser circulating pumps usually operate when maneuvering, backing, or going ahead at low ship speed. At high ship speeds, scoop injection is adequate. For auxiliary ships, main circulating pumps are usually the vertical, centrifugal, double-suction type driven by an electric motor. Water circulation is essential for condensers in service. Insufficient circulating water can result in vacuum loss and, in the extreme, can result in pressure buildup caused by uncondensed steam, probable expansion joint rupture, and condenser deformation. When switching from scoop injection to the main circulating pump, strict adherence to operating procedures must be observed (see Chapter 254, Condensers, Heat Exchangers, and Air Ejectors, for instructions on condensers). Main circulating pumps are provided with bilge suction for emergency dewatering of the bilges. When the main circulating pump is operating on bilge suction, the pump should be started the same as for main condenser circulating service. The main induction valve should then be gradually closed and the bilge suction valve opened. When the pump is operating on a high suction lift, as when pumping bilges, the speed should be reduced to two-thirds of rated speed; slowing the pump will minimize but not eliminate pump noise.

503-2.6.4.5 Miscellaneous Circulating and Supply Pumps. The majority of pumps for applications such as flushing, auxiliary circulating, air conditioning and refrigeration plant circulating, distiller circulating, and potable water have capacities below 2,500 gal/min at total heads under 100 lb/in $^{2}$ . Such pumps are usually of the single-stage centrifugal, volute-type close-coupled to an electric motor. In close-coupled units, the shaft of the motor extends into the pump casing with the impeller secured to the extended motor shaft; the pump casing is bolted to the motor end bell or bearing bracket. The pump casing is fitted with a stuffing box adjacent to the shaft. The stuffing box for these pumps is fitted with a mechanical seal and an auxiliary stuffing box to accept emergency backup packing in case of mechanical seal failure. The shaft is always fitted with a water finger between the auxiliary stuffing box and the motor bearing bracket so that water leakage from the stuffing box will not follow the shaft and enter the motor housing and bearing. In addition to being bolted to the motor, the pump casing is frequently provided with supporting feet, although this construction is not essential where the shaft overhang beyond the motor is short. The Navy has a line of standard close-coupled pumps. This line includes 12 different-sized casings with motors to suit. Size and location of suction and discharge connections and foundation bolting are interchangeable on standard pumps. Motor shaft extensions and bolting to pump casings are interchangeable so that motors of different manufacturers' are interchangeable on pump casings of the same size.

503-2.6.4.6 Fire Pumps. Navy standard fire pump has been developed for 250, 750 through 1000, and 2000 GPM capacities. These pumps are supported by Navy owned technical documentation. General characteristics, drive method and normal usage for each design is presented below. The water end parts are made of titanium which resists the corrosion and erosion of seawater applications. Rated capacities and head for fire pumps are given in Table 503-2-1.

a. 2500 GPM PUMP Horizontal single stage, end suction, low noise diffuser pump close-coupled to a high efficiency, high reliability sealed insulation system electric motor. The application of this pump has been limited to MHC Class.

b. 750-1000 GPM PUMP Horizontal single stage, end suction, double volute pump close-coupled to a high efficiency, high reliability sealed insulation system electric motor. More than 90% of the fleet fire pump applications use the 1000 GPM pump.

c. 2000 GPM PUMP Horizontal split case, single stage, double suction, double volute pump that is flexibly coupled to an electric motor or steam turbine. The application of these pumps has been limited to CV Classes.

**Table 503-2-1** NAVY FIRE PUMPS H VS Q

| Head (psi) | Capacity (gpm) | | | |
|---|---|---|---|---|
| 125 | 750 | 900 | 1000 | --- |
| 150 | 750 | --- | 1000 | --- |
| 175 | --- | --- | 1000 | 2000 |

Prior to the development of the Navy standard design, fire pumps were usually horizontal, single-stage, double-suction type, either electric motor or steam turbine driven. Some of the previous designs are still in use.

503-2.6.4.7 Potable Water Pumps. Potable water pumps are usually of the centrifugal close-coupled type and may be designated as potable water transfer pumps or potable water service pumps. There are usually two pumps installed and operated in accordance with the EOSS, or technical manual. Typically one pump is placed in operation and the other in automatic (standby). Pump operation can be manually or automatically controlled. Automatic control is achieved by use of a low pressure switch located in the discharge piping of each pump and both pumps may be operated in parallel if system demands exceed the rated capacity of one pump. A steep head-capacity curve is required for this application. Additional centrifugal type pumps included in the potable water system are the potable water recirculating pumps, which allow for water treatment to reduce bacterial growth, and hot water circulating pumps which recirculate water in the heaters.

503-2.6.4.8 Distilling Plant Pumps. The submerged tube, vertical basket, and flash type shipboard distilling/desalinating plants use several centrifugal pumps to provide operating services, including the following:

a. Seawater Circulating (Feed) Pump

b. Distillate Pump

c. Saltwater (Seawater) Heater Drain Pump

d. Brine Pump or Eductor

The distillate pump, saltwater heater drain pump, and the brine pump should be vertically mounted, close coupled, or flexible coupled and vented to the upper portion of the distilling plant shell or condenser to prevent the pumps from becoming vapor bound. Other circulating and feed pumps may be vertically or horizontally mounted and close coupled or flexible coupled, depending upon space available.

503-2.6.4.9 Cargo Fuel Pumps. Centrifugal, single stage, double suction, flexibly-coupled cargo fuel pumps driven by an electric motor handle the Cargo Oil (DFM) and JP-5 onload, offload, and transfer requirements. The use of centrifugal pumps to pump viscous fluids can affect pump performance. The effects to be expected are:

a. Head and capacity are reduced below those obtained when pumping water, except that the shutoff head is practically the same.

b. Efficiency is reduced, resulting in a greater brake horsepower required for a given capacity.

Changes in performance are caused by increased impeller disk friction losses and differences in specific gravity. Centrifugal pumps, however, have advantages over rotary pumps in weight and space for large pumping capacities. The effects on cargo oil pump when pumping various viscosities is shown in Figure 503-2-14.

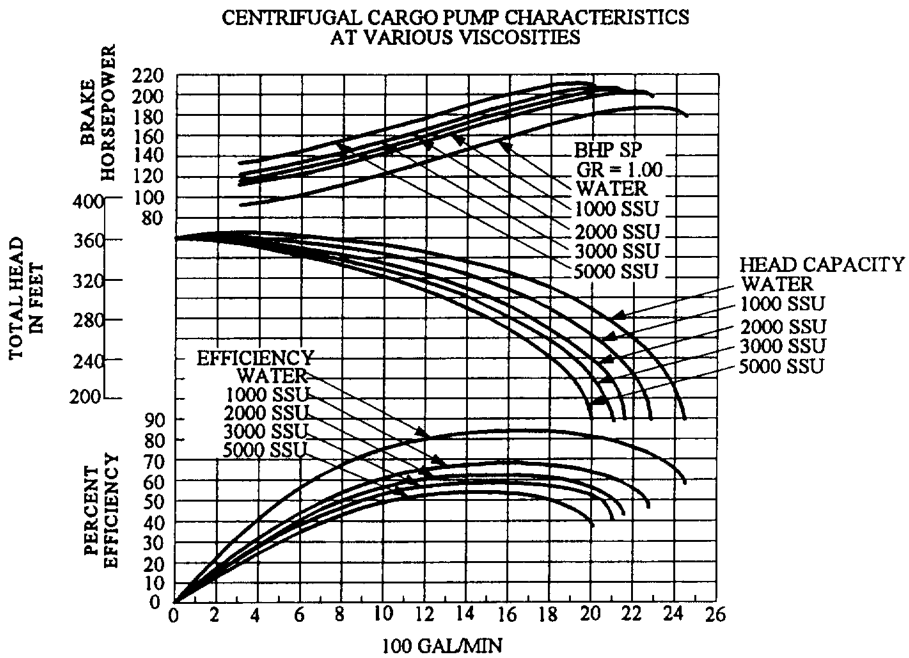



Figure 503-2-14. Effects on Cargo Fuel Oil Centrifugal Pump When Pumping Oils of Different Viscosities.

503-2.6.4.10 Sewage Pumps.

a. Transfer Pumps Each CHT/VCHT and Plumbing Waste Drain tank shall be equipped with two centrifugal sewage transfer pumps in accordance with Mil Spec. MIL-P-24475 or other sewage pumps approved by NAVSEA.. Pump seal leakage shall meet the criteria called out in Sect. 503-5.3.8.3. Each pump shall be capable of developing 10 psig pressure at the deck discharge connection with the discharge valve closed (shutoff head). The pumping system shall operate smoothly without cavitation when pumping to the deck discharge. Where discharge piping to the deck connection is 3 inch or less, a macerator type pump shall be installed.

b. Sewage Aspirator Pumps Aspirator pumps are centrifugal pumps in accordance with MIL-P-24475 and this section, or other pumps as approved by NAVSEA.

c. VCHT Ejectors Two centrifugal ejector pumps shall be installed for each VCHT system for generation of required system vacuum. The ejectors shall be sized so that each individual ejector can maintain normal operating vacuum during peak loads of 5 flushes per minute (Note: On small systems with short piping runs this requirement can be met by modifying the ejector pump activation levels).

d. VCHT Pumps (Ejector/Discharge) For small vessels, two centrifugal ejector/discharge pumps in accordance with this section shall be installed for each VCHT system. Each of the two ejector/discharge pumps shall serve a dual function of generating system collection piping vacuum via the sewage powered ejectors, and discharging the sewage holding tank contents via the pump discharge piping described herein. Each pump shall be sized to ensure proper ejector operation to meet vacuum requirements, and be capable of developing 10 psi gage pressure at the deck discharge connection with the deck discharge valve closed (shut-off head).

503-2.6.4.11 Portable Submersible Pumps. Motor-driven, portable, submersible centrifugal pumps (56225-WE-MMA-010) are intended primarily for damage control. Some pumps of this type are stowed semi-permanently with the suction and discharge hose connected for use as drainage pumps. The pumps are furnished with single or three-phase, water-jacketed ac motors at 120, 240, or 440 volts. Minimum performances for these pumps are given in Table 503-2-2.

**Table 503-2-2.** Pump Minimum Performance

| Capacity (gal/min) | Total Head (feet) |
|---|---|
| 140 | 70 |
| 180 | 50 |

The total heads include 20 feet of suction lift plus hose friction and vertical discharge head. Where the total head required exceeds the head available, two pumps may be coupled for series operation.

**CAUTION**

**Always lower and raise a portable submersible pump by the handling line (rope), never by the electric cable. Secure the handling line to the pump housing through an eye hook installed on the pump for that purpose. Handling the pump by the cable could break the watertight seal where the cable enters the housing. The power cable may be attached to the handling line if considerable slack is left in the cable.**

503-2.6.4.12 Portable Engine-Driven Pumps. The portable, engine-driven pumps previously furnished to the fleet are the P-250, MOD 1, which runs on gasoline; and the P-250, MOD 2, which runs on JP-5. Both of these pumps operate at 250 gal/min at 125 psi. These pumps are dual-service: they can be used as either fire pumps or for dewatering flooded spaces. The operation, care, and maintenance of the P-250 and P-100 pumps are described in Chapter 555 (9930) Vol. 1, Firefighting - Ships.

503-2.6.4.13 Vacuum Priming Pumps. Vacuum pumps are used to prime centrifugal pumps by removing air from the centrifugal pump suction line and pump casing which draws the pumped fluid from the source into the centrifugal pump prior to centrifugal pump startup. Vacuum priming pumps are designed to prime the suction line from the farthest tank in 1 minute, when the tank is less than 5 percent full. Priming pumps are automatically controlled by a pressure switch in the discharge piping of the primed pumps, and by a timing device which stops both the priming and the primed pumps if the discharge pressure is not established within 3 minutes.

503-2.6.4.13.1 Vacuum pumps are also used as a package unit to exhaust condensers prior to startup and during operation by drawing the pressure in the condenser below atmospheric pressure.

## 503-2.7 CENTRIFUGAL PUMP OPERATION.

503-2.7.1 Refer to the Engineering Operation Sequence System (EOSS) or the pump technical manual for specific centrifugal pump preoperation checks, starting procedures, and stopping and securing procedures. Instructions for all pumps cannot be covered here because of the diverse types, designs, and applications in the fleet. The applicable technical manual should be studied before any attempt is made to operate or service the unit. The following general cautions should be observed:

**CAUTION**

**In order to avoid overloading and damaging the driver due to excessive horsepower requirements, radial flow pump should be started with the dis-**

**Caution** - *precedes*

**charge valve closed and both mixed flow and axial flow pumps should be started with the discharge valve open.**

**CAUTION**

**Before starting steam turbine-driven pumps, start electric oil pump or crank hand oil pump; and crack open turbine throttle valve sufficiently to remove water from lines, steam chest, and exhaust casing. Failure to remove water may damage the turbine.**

503-2.7.2 Troubleshooting. If the pump does not operate properly, check the common problems and their causes described in paragraph 503-2.7.2.1 through paragraph 503-2.7.2.10.

503-2.7.2.1 No Liquid Delivered. Causes could be:

a. Air leaks in suction pipe.
b. Air leaks in stuffing boxes.
c. Pump not primed.
d. Pump speed too low.
e. Discharge head too high.
f. Suction lift too high.
g. Impeller passages plugged up.
h. Pump rotation incorrect (for electric motor drive only).
i. Pump impeller installed backward.
j. System valve misalignment.

503-2.7.2.2 Not Enough Liquid Delivered. Causes could be:

a. Air leaks in suction pipe.
b. Air leaks in stuffing boxes.
c. Pump speed too low.
d. Pump rotation incorrect (for electric motor drive only).
e. Pump impeller installed backward.
f. Suction lift too high.
g. Suction sea chest or suction line not sufficiently submerged.
h. Impeller passage partly clogged.
i. Discharge head too high.
j. Insufficient suction NPSH available (indicated by noise and fluctuating pressure).

k. Mechanical defects including worn wearing rings, damaged or eroded impeller, worn stuffing box packing or mechanical seal, or sleeves in need of replacement.

503-2.7.2.3 Not Enough Pressure. Causes could be:

a. Pump speed too low.
b. Air or gas present in the liquid pumped.
c. Mechanical defects.
d. Pump rotation incorrect (for electric motor drive only).
e Pump impeller installed backwards.

503-2.7.2.4 Pump Works for a While, Then Fails to Deliver. Causes could be:

a. Air leakage in the suction line.
b. Air leakage in the stuffing box.
c. Stuffing box water seal plugged.
d. Suction line not sufficiently submerged.
e. Suction lift too high.

503-2.7.2.5 Pump Takes too Much Power. Causes could be:

a. Pump operated at excessive capacity.
b. System head is different than pump rating. For radial flow pumps, the power requirement increases with an increase in flow rate. For axial flow pumps, the power requirement increases toward shutoff. Refer to the manufacturer's power curve to verify if the increased power is due to the pump operating at a point on the head-capacity curve other than rated.
c. Specific gravity or viscosity of liquids other than water may be higher than the pump is designed to handle.
d. Mechanical defects, including rotor binding, shaft bend, stuffing boxes too tight, wearing rings worn, or misalignment.

503-2.7.2.6 Excessive Leakage from Stuffing Box.

a. Misalignment
b. Bent shaft.
c. Interference between rotating and stationary parts.
d. Worn journal bearings.
e. Worn or scored shaft sleeves at packing.
f. Packing installed wrong.
g. Wrong type of packing (if applicable).

h. Rotor out of balance.
i. Sealing liquid contains dirt and grit, causing scoring.
j. Mechanical seal failure (if applicable).

503-2.7.2.7 Short Life of Packing.

a. Speed too high.
b. Misalignment.
c. Bent shaft.
d. Interference between rotating and stationary parts.
e. Worn journal bearings.
f. Worn or scored shaft sleeves at packing.
g. Packing installed wrong.
h. Wrong type of packing.
i. Rotor out of balance.
j. Packing gland too tight, preventing lubrication of packing.
k. Sealing liquid contains dirt and grit, causing scoring.
l. Faulty cyclone separator (if applicable).

503-2.7.2.8 Vibration or Noise.

a. Pump cavitation due to insufficient NPSHA.
b. Quantity of air or gas in liquid is excessive.
c. Speed too high. (Turbine drive)
d. Foreign matter in impeller.
e. Misalignment.
f. Bent shaft.
g. Interference between rotating and stationary parts.
h. Worn journal bearings.
i. Damaged impeller.
j. Rotor out of balance.
k. Packing gland too tight, preventing lubrication of packing.
l. Mechanical failure inside pump causes excessive thrust.
m. Excessive bearing temperature.
n. Bearing incorrectly installed.
o. Dirt in bearings.

p. Bearing rusted from water entering housing.

503-2.7.2.9 Short Life of Bearings.

a. Speed too high.
b. Misalignment.
c. Bent shaft.
d. Interference between rotating and stationary parts.
e. Rotor out of balance.
f. Mechanical failure inside pump causes excessive thrust.
g. Excessive bearing temperature.
h. Bearing incorrectly installed.
i. Bearing rusted from water entering housing.

503-2.7.2.10 Overheating and Seizing.

a. Speed too high. (Turbine drive)
b. Misalignment.
c. Bent shaft.
d. Interference between rotating and stationary parts.
e. Worn journal bearings.
f. Packing gland too tight, preventing lubrication of packing.
g. Wrong type of packing.
h. Rotor out of balance.
i. Packing installed wrong.
j. Mechanical failure inside pump causes excessive thrust.
k. Excessive bearing temperature.
l. Bearings incorrectly installed.
m. Dirt in bearings.
n. Bearing rusted from water entering housing.
o. Loss of suction.

503-2.7.3 Lubrication. Lack of lubrication is one of the primary causes of pump failure. Pump bearings are one of these types:

a. Ring-oiled sleeve or ball bearings.
b. Grease-lubricated ball bearings

c. Oil-lubricated sleeve or ball bearings.

d. Sleeve bearings lubricated by pumped fluid.

e. Roller bearings.

503-2.7.3.1 Check ring-oiled bearings to see that oil rings rotate and carry adequate oil. Check oil level frequently

503-2.7.3.2 Motor-driven pumps and some turbine-driven pumps fitted with ball bearings are usually fitted for grease lubrication. Before initially starting the pump or after long periods of being secured, ensure that bearings are lubricated in accordance with PMS. Grease lubrication is used to:

a. Lubricate the bearing.

b. Exclude water and foreign matter from the bearing housing.

Ensure that water flingers effectively prevent water (from the pump glands) from following along the shaft and entering the bearing housing. If sleeves fitted on pump shafts leak, replace the O-rings. In an emergency, a flinger can be easily made from a sheet of neoprene.

503-2.7.3.3 Turbine-driven pumps are usually fitted with a self-contained force-lubricating oil system supplied by an attached lubricating oil pump (gear or screw-type). Check the oil pressure and flow to or from all bearings including the thrust bearing, ensuring that the attached lubricating oil pump has primed itself. Ensure that cooling-water is flowing through the oil cooler and that all air is vented from the water side of the oil cooler. It may be necessary to bleed air from the lubricating oil system (to maintain steady oil pressure) by opening air cocks on high points of the lubricating oil system and closing the air cocks immediately when oil appears. Ensure that the oil reservoir is free of water and check the bearing housing to ensure that water from pump or turbine glands is not leaking into the lubricating oil system.

503-2.7.4 Vents. Petcocks at the highest point of pump discharge and at the pump casing may need to be opened to release air during start up of pump.

503-2.7.5 Bypass Lines. Friction in centrifugal pumps creates heat which is normally carried away by the liquid being pumped. The heat created frequently results in unacceptable temperature increases at low capacities, particularly for high-pressure pumps. A bypass (also known as recirculation line) from the pump discharge to the source of suction supply will eliminate this temperature rise. The bypass for feed pumps is led from pump discharge to the deaerating feed tank or feed heater. The amount of water bypassed necessary to protect the pump may vary by 5 to 15 percent of rated capacity depending on individual pump characteristics. Fire pumps are generally protected by a bypass of 1 to 5 percent. Bypass lines must be kept open whenever the pump is operating.

503-2.7.6 Priming. Before any centrifugal pump is started, it should be fully primed. The casing and the suction piping must be filled with the liquid to be pumped. The various ways of priming centrifugal pumps are discussed in paragraph 503-2.7.6.1 through paragraph 503-2.7.6.4.

503-2.7.6.1 A pump with a positive head on the suction will prime itself when the vent on top of the pump casing is opened. After all entrapped air has been released, close the vent valve or cocks.

503-2.7.6.2 If the pump operates on a suction lift, prime the pump by the means provided:

a. If the suction line has a foot or check valve and the discharge line contains liquid, fill the pump casing by opening the bypass valve around the discharge valve. Open vent valves or cocks at the top of the pump case. After all entrapped air has been released, close vent valves or cocks. If the discharge line has no liquid, open the valve to the independent supply, if provided, or fill the casing by pouring liquid into a funnel inserted in the vent connection on the casing.

b. If the pump is fitted with an air ejector or priming eductor keep the pump discharge valve closed, open the valve to the ejector from the pump casing and turn on steam or air to the ejector. The pump may be started as soon as the ejector throws liquid continuously. The ejector should be secured after the pump has been started.

c. If the pump is connected to a central priming system, open the valve to the central priming tank, putting the vacuum on the pump case.

d. If the pump is fitted with or connected to a wet-vacuum pump, follow the technical manual for the specific unit. Refer to paragraph 503-2.4.4 for vacuum pump information.

503-2.7.6.3 If a pump does not build up a pressure after priming, it should be stopped and reprimed.

503-2.7.6.4 If the pump is self-priming it is designed to prime itself automatically by an internal hydraulic device or by internal recirculation. Follow the technical manual for the specific unit installed

503-2.7.7 Operation at Best Efficiency Point. Centrifugal pumps should be operated at their rated condition, which is usually the point of best efficiency. Impeller vane angles and liquid passages can be correctly designed for only one point of operation. For any other point of operation the angles and passages are either too large or too small, causing a reduction in pump efficiency at flow points other than design capacity.

503-2.7.8 Excess Capacity. Any centrifugal pump, if operated at excess capacity and low head, may surge and vibrate (causing bearings and shaft problems) and require excess power.

503-2.7.9 Reduced Capacity. When the pump operates at reduced capacity and higher head, recirculation in the pump can result in impeller, casing, and wearing ring erosion. The radial thrust on the rotor also increases causing higher shaft stresses, increased shaft deflection, and accelerated wear on wearing rings and bearings.

## 503-2.8 MAINTENANCE.

503-2.8.1 Using PMS/MRCs. The information provided here is intended to supplement preventive and corrective maintenance procedures of the Planned Maintenance System (PMS). Conduct maintenance procedures as specified on Maintenance Requirement Cards (MRCs). If inconsistencies are noted, submit OPNAV 4700/7, PMS Technical Feedback Report (TFBR).

503-2.8.2 Using Technical Manuals. Instructions in this chapter for pump maintenance and repair are general for all makes and types. Technical manuals (TMs) are furnished for all pump applications except some small miscellaneous service motor-driven pumps. Technical Manuals contain detailed information concerning the specific pump installed. Instructions in this chapter are of a general nature and are intended to supplement equipment TMs.

503-2.8.3 In-Service Observations. Make frequent regular inspections of pumps while they are operating. Investigate immediately any change in the sound of a pump. At least once an hour check:

a. Bearing temperature.

b. Stuffing box temperature and leakage.

c. Pressure gauges.

503-2.8.4 Lantern Rings, Sleeves, and Flingers. If a stuffing box is fitted with a lantern ring, replace the packing beyond the lantern ring at the bottom of the stuffing box (closest to the impeller) and ensure that the sealing water connection to the lantern ring is not blanked off by the packing.

503-2.8.4.1 Usually, pump shafts are fitted with sleeves in the stuffing box area to protect the shaft from wear due to friction between the packing (whether the packing is primary or is a backup to a mechanical seal) and the rotating shaft. Ensure that shaft sleeves are tight on the shafts. These are usually secured by keying to the shafts or by screw threads cut in the direction opposite to the shaft rotation. Care should be taken to see that water does not leak between the shaft and shaft sleeves. If shafts or sleeves are roughened or grooved, they should be turned or ground to give a smooth surface (otherwise, keeping shaft sleeves tight will be difficult.)

503-2.8.4.2 Deflectors (Flingers) are always fitted on shafts outboard of stuffing box glands to prevent stuffing box leakage from following along the shaft and entering the bearing housings. Care should be taken to see that flingers are tight on shafts. If flingers are fitted on the shaft sleeves instead of on shafts, ensure that no water leaks under sleeves.

503-2.8.5 Casing Wearing Rings. The design clearance and the maximum allowable clearance between the impeller and the casing wearing rings are shown on manufacturers' plans and in the individual equipment technical manual. When the maximum allowable clearance is exceeded, the wearing rings should be replaced in accordance with the PMS and the individual equipment technical manual. Failure to replace the wearing rings when the allowable clearance is exceeded will result in a decrease of pump capacity and efficiency.

503-2.8.5.1 Allowable clearances are greater on low pressure pumps (circulating pumps) than on medium and high pressure pumps (condensate and feed pumps). As the differential pressure across the wearing ring increases, closer tolerances must be maintained to prevent increased leakage resulting in reduced pump performance.

503-2.8.6 Pump Lubrication Maintenance. Maintaining high quality oil and grease ensures that the problems related to pump lubrication ( paragraph 503-2.7.3) will be minimized and improves the reliability of the pump unit (pump and driver).

503-2.8.6.1 Turbine driven pumps having a forced lubricating oil system require periodic draining, flushing, and cleaning of the oil system and sump. Detailed requirements for maintaining oil quality are provided in NSTM S9086-H7-STM-010/CH262 and the procedure to accomplish this maintenance is provided in the PMS, the applicable pump technical manual, and/or the technical repair standard. Typically this maintenance consists of draining the system and sump, cleaning the sump, filling the sump with hot flushing oil and operating the system for a period of time. The flushing oil is then drained and the sump is cleaned, followed by refilling the sump with new oil. Sediment and water should be drained from the edge filtration type filter (CUNO or similar type) and filters or strainers cleaned or changed as applicable. Subsequent testing of the system integrity and oil quality is also required.

503-2.8.6.2 Electric motor driven pumps having grease as the lubricating medium should be maintained as required in the applicable technical manual, PMS and, NSTM (Chapter 262). Some bearing housings have drain/ vent plugs on which a grease cup can be attached and screwed down to inject the appropriate amount of grease. Others have a grease fitting for which a grease gun may be used to inject the required grease. Care should be taken to avoid overfilling the bearing housing with grease. When a bearing housing is too full of lubricant the churning grease generates heat which causes grease deterioration which, inturn, can result in bearing failure. Bearing housings should also be checked to ensure that they are free of foreign matter and water, and are secure on the shaft. The source of any water contamination should be identified and corrected as soon as possible.

503-2.8.7 Oil-Lubricated Sleeve Bearings. Clearances should be maintained as shown on the manufacturer's plans or Table 503-2-3. However, detailed information concerning inspection, clearances, and maintenance is available in NSTM Chapter 244, Propulsion Bearings and Seals, if the manufacturer's data is not available.

503-2.8.8 Thrust Bearings. Thrust bearings are designed to absorb the axial forces and may be of the segmented, non-segmented (tapered land), or ball thrust bearing type. All bearing maintenance should be accomplished as detailed in the applicable PMS, technical manual, and NSTM.

503-2.8.8.1 Thrust bearing maintenance associated with the steam turbine driven pumps are covered under the applicable PMS and technical manual, and under NSTM Chapter 244, *Propulsion Bearings and Seals* , or NSTM 231 *Propulsion Turbines* .

503-2.8.8.2 Thrust bearing maintenance for electric motor driven pumps is addressed in the applicable PMS, technical manual, and in NSTM Chapter 244, *Propulsion Bearings and Seals* , or Chapter 300, *Electric Plant, General* .

503-2.8.9 Internal Water-Lubricated Sleeve Bearings. Most types of vertical condensate and main circulating pumps are fitted with a water-lubricated sleeve bearing inside the pump casing. If this type of bearing is fitted in a pump, the bearing must be supplied with enough clean water to ensure adequate lubrication and cooling.

503-2.8.9.1 Various materials have been used for internal water-lubricated sleeve bearings, including:

a. Laminated phenolic material, such as fabric-base bakelite or micarta (grade FBM).

b. High lead content bronze.

c. Graphited bronze.

503-2.8.9.2 The condition of all internal water-lubricated sleeve bearings should be checked frequently to guard against excessive wear which would cause misalignment and possible shaft failure.

503-2.8.10 Cooling Water. Oil coolers or oil cooling coils of all pumps fitted with oil cooling arrangements should be supplied with clean cooling water, usually seawater. A frequent check of oil temperatures should be made to ensure that the cooling system is functioning satisfactorily. This is particularly important on high-speed boiler feed pumps and on turbine-driven units fitted with worm-gear type reduction gears.

503-2.8.11 Care of Driving Units. Instructions are given in the Naval Ships Technical Manual for maintenance of diesel engines (Chapter 233),auxiliary steam turbines (Chapter 502), and electric motors and controllers (Chapter 302).

503-2.8.12 Couplings. Many pumps are connected to the driver shafts with flexible couplings. These couplings permit slight movement of the pump and driver shafts while transmitting power. Maintenance requirements for satisfactory coupling life are lubrication (if required), alignment, and balance.

503-2.8.12.1 Maintenance. Coupling maintenance should be accomplished in accordance with the PMS. When performed, PMS coverage and periodicity are considered adequate to ensure proper coupling lubrication and alignment, provided there are no obvious operational defects such as undue vibration or noise.

**Table 503-2-3.** Centrifugal Pump Sleeve Bearing Clearances (White Metal)

| **Journal** | | **Bore of Bearing** | | **Wear at Which Bearings Should be Renewed** |
|---|---|---|---|---|
| **Diameter** | **Tolerance** | **Minimum Diametral Clearance, Journal and Bearing** | **Tolerance** | |
| 1 | +0.000<br>-0.001 | 0.0025 | +0.001<br>-0.000 | 0.0075 |
| 2 | +0.000<br>-0.001 | 0.003 | +0.001<br>-0.001 | 0.0075 |
| 3 | +0.000<br>-0.001 | 0.004 | +0.001<br>-0.000 | 0.010 |
| 4 | +0.000<br>-0.001 | 0.005 | +0.001<br>-0.000 | 0.015 |

NOTE

These clearances are to be used only in the absence of other instructions, including approved plans. Allowable wear indicated is approximate only, and is based on average wear ring clearances. All values are in inches.

503-2.8.12.2 Non-Lubricated Couplings on Main Feed Pumps. Machalt 255-30001 (ECP-110) converts main feed pump couplings from a lubricated flexible coupling to a non-lubricated flexible coupling. If installed, check the non-lubricated flexible coupling for wear in accordance with the PMS and pump technical manual. Do not lubricate this coupling.

503-2.8.12.3 Lubrication. Standardized procedures and precautions which should be observed when lubricating the coupling are described in paragraph 503-2.8.12.3.1 through paragraph 503-2.8.12.3.3.

**CAUTION**

**Do not leave the grease fitting in the coupling because imbalance will result.**

503-2.8.12.3.1 For grease-lubricated couplings, remove the two lube plugs (180 degrees apart) and position the coupling for horizontal installations so that one hole is 45 degrees above the horizontal. Install the grease fitting in the top hole and apply grease (DOD-G-24508) until the excess grease flows out of the lower opening. Remove the grease fitting and reinstall both plugs.

503-2.8.12.3.2 The oil should be drained from the oil-lubricated couplings and the couplings should be refilled with the specified oil to the correct oil level as directed by the PMS. This also should be done after any pump overhaul.

503-2.8.12.3.3 During the coupling lubrication, watch for loss of lubricant. Leakage may occur through the coupling flange if the bolts are loose, damaged, or if gasket between the coupling flanges. Usually the gasket thickness is 1/64 inch. Consult the manufacturer″ technical manual for the recommended gasket and flange O-rings.

503-2.8.12.4 Imbalance. Vibration can be caused by imbalance. The couplings are balanced as a set and matchmarked. The hub should be aligned by the match marks before reassembly. The flanges should also be reassembled with attention given to their match marks. The nuts and bolts are also numbered in specific holes which are usually numbered. If the numbers cannot be located, the holes, bolts, and nuts should be numbered during disassembly for identification during reassembly.

503-2.8.12.5 Submission of Failure Couplings. In most cases, close attention to coupling lubrication (if required), alignment, and balance will prevent coupling failure. When a coupling failure occurs, the failed coupling should be sent to Commanding Officer, Naval Surface Warfare Center Ship System Engineering Station (NAVSURFWARCEN), (Code 9232), Philadelphia, PA 19112, for assistance in determining deficiencies.

## 503-2.9 SPEED-REGULATING AND SPEED-LIMITING GOVERNORS.

503-2.9.1 All turbine-driven units are fitted with a speed-regulating or speed-limiting governor.

503-2.9.2 Speed-Regulating Governor. The speed-regulating governor, by its control and regulation of the steam admission to the turbine, automatically maintains the speed of the turbine at a predetermined value, under all conditions of load and exhaust pressure, within the limits of design of the turbine. For additional information refer to NSTM Chapter 502, Auxiliary Steam Turbines.

503-2.9.3 Speed-Limiting Governor. The speed-limiting governors, by control of the steam admission to the turbine, will not permit the turbine to operate at a speed in excess of the governor set point, but will permit the turbine to continue in operation at this speed. The speed-limiting governor is a safety device that protects the turbine from an overspeed condition. For additional information refer to NSTM Chapter 502, Auxiliary Steam Turbines.

**CAUTION**

**These governors shall never be lashed down or otherwise rendered inoperative.**

503-2.9.4 Speed Fluctuation. The governor should be set to give rated speed at rated load conditions, and, with this setting, turbine speed should not exceed the rated speed by more than 5 percent for any condition of load including pump shutoff. If the governor does not function within the set limit, it should be repaired. For satisfactory parallel and series feed booster pump and boiler feed pump operations, governors of all identical pumps must be set for the same speed.

503-2.9.5 Overspeed Trips. Turbines equipped with speed-regulating governors are fitted with an overspeed trip which is a safety device that shuts off steam to the turbine after a predetermined speed has been reached. This predetermined speed is about 110 percent of normal operating speed. For additional information on overspeed trips, refer to NSTM Chapter 502, Auxiliary Steam Turbines.

**CAUTION**

**Overspeed trips shall never be lashed down nor otherwise rendered inoperative.**

## 503-2.10 PUMP PRESSURE-REGULATING GOVERNORS.

503-2.10.1 In addition to speed-regulating and speed-limiting governors, turbine-driven boiler feed pumps and fire pumps are fitted with pump pressure-regulating governors which are automatic throttling valves installed in the steam supply line to the turbine. Variations in pump discharge pressure or in pressure differential actuate the governor causing it to vary the steam flow to the turbine, thereby regulating pump speed. For additional information on pump pressure-regulating governors, refer to NSTM Chapter 502, Auxiliary Steam Turbines.

503-2.10.2 Low Suction Pressure Trips. Turbine-driven boiler feed pumps are also fitted with low suction pressure trips that monitor the pressure in feed pump suction lines. When the pressure decreases to a predetermined level, the steam flow to the turbine is cut off and the pump is shut down. The low suction pressure trip ensures that adequate fluid pressure is available at the inlet of the feed pump. For additional information on low suction pressure trips, refer to NSTM Chapter 502, Auxiliary Steam Turbines.

**CAUTION**

**Low suction pressure trips shall never be gaged or otherwise rendered inoperative.**

## 503-2.11 TESTS AND INSPECTIONS.

503-2.11.1 Centrifugal pumps should be tested or inspected periodically in accordance with PMS requirements. The procedures given in paragraph 503-2.11.2 through paragraph 503-2.11.4 are intended to be used as a general guide and are not inclusive of all tests or inspections required by PMS.

503-2.11.2 Periodically, in accordance with authorized PMS, operate all pumps by steam or power. If power is not available, move pumps by hand. Lubricate the pressure regulating governor (for types requiring lubrication). Check lubricating oil for water and contamination. Periodically, check the thrust position of the pump rotor. As necessary, sound and set up on all foundation bolts, and secure all foundation dowel pins. Periodically, check sleeve bearing clearance by leads or crown thickness measurements as described in NSTM Chapter 244, Propulsion Bearings and Seals.

503-2.11.3 During the overhaul cycle, check internal water-lubricated sleeve bearings and shafts for wear and scoring. Open pump and reduction gear casings for inspection and cleaning. Check clearance of all diaphragms and casing throat bushings, impeller and casing wearing rings, pressure breakdown drums and bushings. Renew as necessary.

503-2.11.4 These periodic tests and inspections are the minimum necessary to ensure safe and reliable equipment operation. Indications of low discharge pressure or other manifestations of improper operation require more frequent or more extensive tests and inspections. Seawater and brine pumps should be opened and inspected as performance dictates. In general, pumps should be turned over and lube oil checked each time a unit is placed in service. Cleaning of lube oil sumps should be dictated by oil condition. The degree of oil contamination will dictate the necessity for further inspection of bearings and journals. For turbine tests and inspections, see NSTM Chapter 502, Auxiliary Steam Turbines.

## 503-2.12 REPAIRS.

**CAUTION**

**Only qualified personnel should attempt repair of these pumps.**

503-2.12.1 General Guidance. The repair and overhaul of a pump are intended to restore it to its original performance profile and not necessarily intended to return the overhauled pump to the original manufacturer's drawing tolerances. In the process of overhauling a pump, the worn or damaged subcomponents are either replaced or repaired. The Supervisors Work Specification shall identify the class of overhaul (i.e., Class B) authorized for the item. Definition of overhaul class is provided in General Specification for Overhaul (GSO) Section 042. Where applicable, the Supervisors Work Specification shall invoke the overhaul of the item to be accomplished in accordance with an approved Technical Repair Standard (TRS). Where an approved TRS does not exist, or is not authorized, the overhaul of the item or system (i.e., valve, pump, filter, pipe) shall be in accordance with applicable drawings or technical manuals as modified by the overhaul criteria paragraphs in GSO Sections 503 and 505.

503-2.12.2 Assembly Drawings. When repairing a pump or making an interior pump examination, have at hand all drawings and available dimensional data pertaining to the pump. Bearing bridge gage readings, clearances between the impeller and casing wearing rings, water seal clearances, or gland adjustments must be corrected if these dimensions have become altered. Altered dimensions result in poor operation which will continue in spite of other major repairs, unless dimensions are correct.

503-2.12.3 Wearing Parts. Centrifugal pump parts most frequently requiring repair or replacement are described in paragraph 503-2.12.3.1 through paragraph 503-2.12.3.4.

503-2.12.3.1 Casing Rings. Casing rings keep the internal bypassing of liquid to a minimum, thereby maintaining the pumps efficiency. Clearances should be checked periodically and whenever the pump casing is opened. If the clearance exceeds the specified clearance stated in the manufacturers' plans or pump technical manual, replace the rings.

503-2.12.3.2 Shaft Sleeves. Operating personnel frequently take up too hard on the packing in an attempt to stop stuffing box leakage; this causes shaft sleeve scoring. Sleeves should be examined whenever a pump is opened, and, if only slightly scored, they should be smoothed; when worn, they should be replaced.

503-2.12.3.3 Sleeve Bearings. Worn sleeve bearings cause the rotor to drop which causes wear of the casing rings. Inspect the condition and contact pattern of the bearing. Bearing babbitt contact is indicated by the polished portion of the bearing surface. Slight circumferential grooving and localized wipes indicated by shiny spots are acceptable, but babbitt build-up requires removal. Unacceptable conditions are wiping, heavy or extensive circumferential grooving, pitting, cracks, embedded dirt, loose babbitt, and discoloration ranging from gray to black, which indicates corrosion and oxidation. Repair or replace the bearing shell if these conditions exist. Bearings of centrifugal pumps should be rebabbitted in accordance with NSTM Chapter 241, Propulsion Main Reduction Gears, Couplings and Associated Equipment, when bridge readings or leads show that maximum allowable wear has occurred. Bearings of centrifugal pumps should be changed when bridge readings or leads show that the maximum allowable wear has occurred. Bearings for shafts of less than four inches in diameter are usually replaced and rebabbitted only in an emergency.″ (See paragraph 503-2.12.4.3) In the absence of such data, tolerances given in Table 503-2-3 should be followed. All bearings which are to be rebabbitted shall be repaired in accordance with NSTM Chapter 244 Propulsion Bearings and Seals.

503-2.12.3.4 Bushings. Bushing clearances should be measured whenever the pump is opened. Bearing wear can cause bushing wear, and bushings should be renewed after bearings are restored to their original dimensions.

503-2.12.4 Inspection and Repair of Pump Parts. The inspection and repair methods required for pump part repairs are described in paragraph 503-2.12.4.1 through paragraph 503-2.12.4.2. Refer to GSO Section 503 for specific inspection and repair information.

503-2.12.4.1 Inspection of Pump Parts. Pump parts may be inspected at the IMA or Depot level by several methods. The following information is provided here as a list of inspection methods and applicable specifications.

503-2.12.4.1.1 Nondestructive inspection is used to determine whether pump parts are defective without destroying the parts. Several methods are used and shall be performed in accordance with MIL-STD-271.

a. Magnetic particle test is used to detect surface or subsurface discontinuities in ferromagnetic materials. Finely divided ferromagnetic particles are applied over the material surface and when the material is magnetized some of the particles will be gathered by the leakage field. This magnetically held collection of particles forms an outline of the discontinuity and generally indicates its location, size, shape and extent. See NAVSEA 0900-LP-003-8000 for acceptance criteria.

b. Liquid penetrant test is used on non-magnetic (non-ferrous and austenitic corrosion-resisting or stainless steel) material parts that contain suspect surface discontinuities. Sometimes called dye penetrant test, a colored dye is poured onto the surface of the casting and the dye will collect in any cracks or surface discontinuities that exist. See NAVSEA 0900-LP-003-8000 for acceptance criteria.

c. Radiography is the process of taking X-rays of the part to determine the integrity and soundness of the casting. Radiography detects subsurface (internal) defects or discontinuities.

d. Brinell hardness is a process used to determine how hard the material is and this number can then be compared to the material specification.

e. Chemical analysis is used to determine whether the material has the correct chemical composition.

503-2.12.4.1.2 Threaded parts are inspected for burnished, galled, crossed, torn, cracked, deformed, or missing threads. Significantly damaged threads can reduce the strength and integrity of a joint.

503-2.12.4.1.3 Wall thickness is inspected to determine the pressure containing ability of a casting. If the wall thickness is reduced beyond a critical dimension due to erosion or corrosion, the casting could rupture or leak.

503-2.12.4.1.4 Sealing surfaces are inspected to determine the pressure containing ability of a joint. If the sealing surface contact area is reduced beyond a critical amount due to erosion or corrosion, the joint could leak.

503-2.12.4.1.5 Keyways and keys are inspected for deformed, cracked or chipped edges, or high spots. These conditions can result in a failed key or an improper fit between parts. If the key fails, the equipment will not rotate.

503-2.12.4.1.6 Gear teeth are inspected for cracks, nicks, or abrasion. These conditions can result in failed gear teeth and inhibit the smooth operation of the equipment.

503-2.12.4.2 Repair of Pump Parts. Pump parts may be repaired at the IMA or Depot level by several methods. The following information is provided here as a list of repair methods and applicable specifications.

503-2.12.4.2.1 Weld repair consists of depositing weld material on a casting to build up an area which is then remachined to the original dimension. Weld repair of pump casings and internals, including pump bore repair, shall be accomplished in accordance with MIL-STD-278. Weld repair is normally used for repairing eroded or damaged casings, building-up wear ring seating areas, and repairing gasket mating faces. Weld repair of new shaft rotating parts (impellers, etc.) shall be in accordance with MIL-STD-278 or NAVSEA S9074-AR-GIB-010. Weld repairs of in-service shaft rotating parts require approval through local Technical Authority.

503-2.12.4.2.2 Impregnation is a process that seals porosities in castings by introducing sealant into the casting pores by pressure or vacuum techniques. Impregnation is permitted, only with NAVSEA approval, on structurally sound nonferrous castings to seal leakage due to porosity or other nonstructural defects. Castings shall be inspected and repaired in accordance with MIL-STD-278 or NAVSEA S9074-AR-GIB-010 and impregnated in accordance with MIL-STD-276. Sodium silicate shall not be used for impregnation.

503-2.12.4.2.3 Thermal spray coating is a process used for repair and corrosion protection of ferrous and nonferrous pump metal parts. This process is used only when the amount of material buildup required is a thin layer. Thermal spray coating shall be accomplished in accordance with MIL-STD-1 687 and is approved by NAVSEA for the following repairs:

a. Repair of static fit areas to restore original dimensions, finish, and alignment.

b. Repair of seal (including packing) areas to restore original dimensions and finish.

c. Repair of fit areas on shafts to restore original dimensions and finish.

503-2.12.4.2.4 Brazing repairs are restricted to reattachment of piping connections on pumps and shall be accomplished in accordance with NAVSHIPS 0900-LP-001-7000.

503-2.12.4.3 Emergency Rebabbitting of Small Bearings Shipboard logistics provides for spare parts allowances which support the replacement of small bearings vice rebabbitting. However, emergency rebabbitting of bearings may be accomplished at Intermediate Maintenance Activities (IMAs) or at the Organizational (Shipboard) Level if materials are available.

503-2.12.5 Pump Repairs Using Epoxy-Based (Polymeric) Compounds.

503-2.12.5.1 Limited Use of Polymeric Repairs. Polymeric-based compounds have been developed for use in repairing mechanical equipment. NAVSEA has approved the use of specific products, in certain applications, for temporary repairs where welding, plating, or flame spraying cannot be accomplished. Currently, polymeric compounds are not authorized for any application in submarine and or in the reactor plant or steam plant of nuclear powered ships.

Systems/pumps where the use of polymeric materials is specifically prohibited are those used for lube oil service/transfer, fuel service/transfer, fresh water drain collection service, (boiler) feed and booster service, main/auxiliary condensate service, and potable water service. Use is restricted to surface ship systems where a failure of the polymeric compound would not contaminate the fluid or cause damage to other components. Polymeric materials are not recommended for use on impellers or other rotating elements or bearing surfaces.

503-2.12.5.2 Polymeric Compound Selection. Polymeric compound selection is critical due to the varying materials, strengths, lubricating properties, surface finishes, thermal expansion and conductivity coefficients which will be required for that specific environment, fluid being pumped, and base metal being coated.

a. Metal filled materials are designed for use where build-up repairs are required. They have higher impact strengths than ceramic materials. They can be drilled, threaded, and machined.

b. Ceramic-based materials are designed to reduce the effects of corrosion and erosion. These materials can be used as a build-up medium and as a surface coating, depending on viscosity. Generally these materials are slightly stronger than metal-based materials, but are less resistant to thermal and mechanical shock and crack easier. Ceramic build-up materials can be machined, but not drilled or tapped.

c. Organic compounds used as fill materials in polymeric compounds shall not be used for pump repairs because they require special handling due to their high solvent content and high carcinogenic potential.

503-2.12.5.3 Pump repair with Polymeric Compounds. All repairs will require not only the correct materials, but proper surface preparation to provide an adequate anchor for the polymeric material application. This is probably the most critical aspect of repair and will require most of the total time. It will require solvent cleaning before any preliminary abrasive blast cleaning, soaking in clean fresh hot water (120-140°F), heating in oven (or with a torch), abrasive blasting, and final solvent cleaning prior to coating.

503-2.12.5.3.1 If the part being repaired has an existing polymeric coating and the coating is not extensively worn it may be possible to restore the coating without extensive preparation. The polymeric compound manufacturer should be contacted to determine the compatibility with the existing coating.

503-2.12.5.3.2 Pumps repaired with polymeric compounds require particular attention to detail because the repairs become brittle with age, or thermal or stress cycling. Therefore the structural integrity must be maintained by the appropriate use of acceptable metal bars, plates, bolts, reinforcing wire mesh, or metal framing that is suitable with the base metal being coated.

503-2.12.5.3.3 Pump casing erosion and joint damage repair is typically either a two-or three-step repair process, depending on the extent of erosion/damage.

**CAUTION**

**Casing joint dimensions are critical to the internal running clearances and must be within design criteria following repair.**

a. Two step process consists of filling and fairing the area with metal or polymeric build-up material. Then sealing the entire repair with a polymeric coating.

b. The three step process requires the use of wire mesh or metal framing attached to the damaged area by screws or bolts as stated in paragraph 503-2.12.5.3.

503-2.12.5.3.4 Close coupled pump casing rings and casing ring seats may be repaired by the use of metal or ceramic build-up materials and then machining the repaired component back to the original dimensions. Concentricity between the shaft centerline and the casing ring seat is critical.

503-2.12.6 Impeller Centering. It is essential to the best centrifugal pump operation that the centers of the impeller exit passages line up accurately with the centers of casing or diffuser waterways; otherwise, excessive turbulence and friction losses may occur, and the thrust load may be increased.

503-2.12.7 Aligning Pump. Pumps should be periodically checked for alignment because piping strains may develop, distorting the unit. Bearing wear and distortion of supporting structure may also contribute to misalignment. With the exception of close coupled pumps, pumps should be aligned in accordance with Section 6 of this chapter.

503-2.12.7.1 Close-Coupled Pump Alignment.

503-2.12.7.1.1 Close-coupled pumps have one common shaft for the pump and motor. When a close-coupled pump has been overhauled, dial indicator readings shall be taken during pump reassembly to ensure that the shaft is straight and that the impeller and stationary parts are perpendicular and concentric.

503-2.12.7.1.2 Normally, the limit of acceptable shaft run out is 0.002-inch TIR unless otherwise specified by the manufacturers' technical manual. When indicating from the shaft to the flange face or the spigot rim or rabbet of the housing or motor bracket, the run out should remain within 0.002 inch. The motor bracket is fastened to the motor, and the pump casing is fastened to the motor bracket.

503-2.12.7.1.3 Because the casing is assembled with one machined surface against another, errors in the end face are cumulative. These errors could result in a varying centerline which deviates from the shaft centerline. In a practical sense, the impeller wearing surfaces and the casing wearing rings may not be concentric. This causes excessive clearance on one side and rubbing on the other. This condition is aggravated if the casing wearing ring is slightly out of round. In the final assembly it is important to ensure that the radial clearances between the wearing rings and impeller surfaces are within tolerance and that the pump impeller is located with acceptable end clearance.

503-2.12.7.1.4 Procedures that shall be used in checking or performing close-coupled pump alignment are given in paragraph 503-2.12.7.1.5 through paragraph 503-2.12.7.1.12.

503-2.12.7.1.5 Check shaft run out. Mount the dial indicator firmly to the motor if practical. Position the sensing arm against the shaft next to the motor bell housing. Zero the indicator. Observe the indicator and rotate the shaft one turn. Turn the shaft to the highest indicated reading. Zero the indicator again. Rotate the shaft one turn. Record the maximum deviation. Run out should be within 0.002-inch TIR or within the manufacturers' recommended tolerances. If outside of these tolerances, the run out must be corrected before proceeding. Excessive shaft run out next to the motor indicates the problem is in the shaft or bearings. Shift the indicator sensing arm to the pump end of the shaft. Check shaft run out. If shaft run out next to the motor is acceptable but pump end run out is excessive, the shaft is bent or damaged. Repair it before proceeding.

503-2.12.7.1.6 Check motor flange end face run out and motor spigot radial run out. Mount the dial indicator firmly to the pump shaft. Position the sensing arm against the face of the motor flange. Zero the indicator. Observe the indicator and rotate the shaft one turn. The pump flange face should be perpendicular to the axis within 0.002 inch. If there is excessive run out, correct the cause before proceeding. Position the indicator sensing arm on the motor spigot. Zero the indicator. Observe the indicator and rotate the shaft one turn. The motor spigot should be concentric within 0.002 inch. If there is excessive run out, the cause may be burrs or a damaged spigot.

503-2.12.7.1.7 Install the motor bracket. Check the end face and radial run out. Mount the dial indicator firmly to the pump shaft. Position the sensing arm against the end face. Zero the indicator. Observe the indicator and rotate the shaft one turn. The spacer face should be perpendicular to the axis within 0.002 inch. If there is excessive run out, the cause may be an incorrectly installed spacer piece or burrs on the mating surfaces. Correct it before proceeding.

503-2.12.7.1.8 Install the close-coupled pump casing. Check the radial and end face run out of the casing and wearing ring (if installed). Mount the dial indicator firmly to the pump shaft. Position the sensing arm against the casing end face. Zero the indicator. Observe the indicator and rotate the shaft one turn. The casing face should be perpendicular to the axis within 0.002 inch. If there is excessive run out, the cause may be an incorrectly installed casing or burrs on the mating surfaces. Shift the sensing arm to the end face of the wearing ring. Zero the indicator.

Check the wearing ring run out. The wearing ring face should be perpendicular to the axis. Acceptable casing face run out with excessive wearing ring face end run out means the wearing ring is seated improperly or is damaged. Shift the sensing arm to the wearing ring inner surface. Zero the indicator. Check the radial run out. The wearing ring should be concentric within 0.002 inch. If the radial run out is excessive, the cause may be incorrectly installed or out-of-round wearing rings. Correct it before proceeding.

503-2.12.7.1.9 Install the impeller. Check the impeller radial and face run out. Mount a dial indicator firmly to the pump casing. Position the sensing arm against the impeller shoulder or hub face. Observe the indicator. Rotate the pump shaft one turn. The impeller should be perpendicular to the axis within 0.002 inch. If there is excessive run out, the cause may be an improperly installed impeller or flaws on the impeller shaft seat and shoulder. Shift the indicator sensing arm to a parallel surface on the impeller (wearing ring surface, if installed). Zero the indicator. Check the radial run out. The impeller should be concentric within 0.002 inch. If there is excessive run out, the impeller may not be centered on the shaft, a burr is on the indicating surface, or the impeller outside diameter (OD) is not machined properly. Correct the excessive run out before proceeding. If the impeller is remachined, reestablish dynamic balance.

503-2.12.7.1.10 Set the end plate in place. Before inserting the hold-down fasteners, mount a dial indicator firmly to the motor, if practical. Position the indicator sensing arm against the top of the end plate. Zero the indicator. Check the clearance between the end plate shoulder and the pump casing by moving (or trying to move) the end plate up and down. Record the reading. If there is any clearance, hold the end plate in the center while tightening the hold-down fasteners. For example, if the indicated clearance is 0.005 inch, move the end plate a maximum of 0.0025 inch to center the end plate on the pump casing.

503-2.12.7.1.11 If the close-coupled pump has supporting pads on the motor and pump end, install and tighten the motor hold-down fasteners. Mount a dial indicator firmly on the motor. Position the sensing arm against the top of the pump casing. Zero the indicator. Measure the clearance between the pump pads and the bedplate by inserting a thickness gage. If there is a gap, shim before inserting bolts. When shimming to correct the gap, add 0.002 to 0.003 inch to the indicated reading. For example, if the indicated gap reading was 0.005 inch, the shim thickness should be 0.007 inch to allow for compression when hold-down fasteners are tightened. While tightening the pump pad holddown fasteners, observe the dial indication to ensure that tightening the bolts does not misalign the pump.

503-2.12.7.1.12 Insert bolts for suction and discharge piping in the respective pump flanges. Before tightening bolts, mount a dial indicator firmly on the motor. Position the sensing arm against the top of the pump casing. Zero the indicator. The piping must be sufficiently aligned to allow assembly of the flange joints by hand without having to use excessive force. Observe the indicator while tightening the bolts to ensure that the piping is not pulling the pumps out of alignment. If there is indication that the suction and discharge piping are exerting excessive forces on the pump flanges, correct by modifying piping alignment.

503-2.12.8 Balancing. Vibration can be caused by imbalance. The couplings are balanced as a set and match-marked. The hub should be aligned by the match marks before reassembly. The coupling flanges should also be reassembled, aligning match marks.

503-2.12.8.1 The nuts, bolts, and holes are usually numbered; however, if numbers cannot be located, the holes, bolts, and nuts should be numbered during disassembly for identification during reassembly.

503-2.12.8.2 All pump and driving unit rotating parts are balanced dynamically for all speeds from 0 to 125 percent of rated speed. The parts are usually balanced on a balancing machine to ensure accurate balance. Balancing machines are usually available only at naval shipyards. A portable balancing outfit is available on tenders for use by individual ships. With a portable unit, the pump and turbine rotors may be balanced in place. Whenever possible the balancing machine or portable balancing outfit should be used in preference to any other method of balancing.

503-2.12.9 Composite Material Centrifugal Pump Components. For years, composite material replacement components have been used in centrifugal pumps when metallic parts were not readily available or obsolete. With the success of this technology offering many benefits to the Navy when properly applied, efforts have begun to procure these composite components through the supply system as certain National Stock Numbers (NSNs) quantities require replenishing. NAVSEA Drawing 803-7226047 provides detailed information on material composition, testing, service applications, stock numbers, etc. for composite material pump parts. Further efforts are underway to use these components for pump repair and overhaul at Intermediate Maintenance Activities and shipyards, and to eventually procure new pump units complete with composite internals. To date, the use of composite material replacement components has been limited to surface ship pumps that are not part of the reactor plant or steam plant of a nuclear propulsion plant. As more experience is gained, the area of application may be expanded.

503-2.12.9.1 Prohibited Systems. Currently, composite components are not authorized for any application in submarines or in the reactor plant or steam plant of nuclear power ships.

503-2.12.9.2 Authorized Systems. Surface ship pumps and systems in which the use of composite components is authorized are:

General Service Circulation Pumps

Auxiliary Circulating Pumps

Auxiliary Machinery Cooling Water Pumps

Distilling Plant Feed Pumps

Distilling Plant Brine Pumps

Potable Water Vacuum Priming Pumps

Diesel Engine Cooling Water Pumps

Refrigeration Condenser Regulating Pumps

Reserve Feed Transfer Pumps

Seawater Service Pumps

Main Circulating Pumps

Fire Pumps

P-250 Fire Pumps

Distillate Pumps

Potable Water Pumps

A/C Chill Water Pumps

Bilge Pumps

Application to other systems would require authorization on a case basis.

503-2.12.9.3 Maintenance Philosophy. Maintenance philosophy for pumps containing composite components differs from that of pumps with metal internals. Although composite materials may have a relatively high strength-to-weight ratio, composites are usually much softer than metals. Composite components should be handled and installed with care to avoid scoring, chipping, and cracking. When installing a composite shaft sleeve, bushing, or other tight fitting component of close tolerances, never strike the composite component with a hard tool or object. Due to the physical differences between composite material replacement components and the metallic components currently in use, consult the composite manufacturer's instructions prior to installation or maintenance.

The use of solvents on composite components shall be avoided. Depending on the composite material, various solvents may or may not effect the component. The intrusion of solvents into a polymer matrix tends to disrupt the associative forces and consequently alter the material's molecular volume. On a larger scale, intrusion of solvents may interfere with the adhesive forces shared between resin and reinforcement. Either of these solvent intrusion conditions may result in undue component swelling and eventual failure.

Special considerations are required when performing maintenance on a pump which had composite components previously installed. Due to moisture absorption, some materials have a tendency to swell. This swelling may impact the tightness of the fit between components with close tolerances such as the impeller or the shaft sleeve. When servicing such a pump, allow ample drying time prior to attempting removal of tight fitting components to avoid damaging the composite components.

503-2.12.9.4 Installation of Composite Components. The following list identifies installation procedures for composite components and is intended primarily for informational purposes only. Each manufacturer shall provide written instructions for installation and maintenance for each composite component. Manufacturer's installation and maintenance instructions shall be complete and address each item below.

Casing Ring Installation with Set Screws

Casing Ring Installation with Press Fits

Split Case Casing Ring Installation

Basic Impeller Installation

Tapered Bore Impeller Installation

Shaft Sleeve Installation

Post-Installation Considerations

Clearances Between Surfaces

Rotating Assembly Balance

Post-Installation Operational Checks

503-2.12.10 Pump Mounting. Pump mounting requirements are driven by the service application, size, and weight. Pumps are typically mounted in either a rigid, vibration, or shock mounted configuration and may encompass the use of Distribution Isolation Material (DIM) mountings or resilient mountings to produce a low noise signature affiliated with that particular component.

503-2.12.10.1 Rigid mounting. Pump units are affixed directly to the hull through a foundation and have no sound isolation properties.

503-2.12.10.2 Vibration mounting. Structureborne and airborne noise is reduced through the use of sound isolated mounting of the pump unit. Sound emissions are reduced or eliminated by use of resilient mounts or compound mountings using an intermediate mass. Compound mountings are typically found on noise critical machinery and equipment. Low frequency Distribution Isolation Material (DIM) mountings and resilient mountings (MIL-M-17185) are used only for specified applications or where approved by NAVSEA.

503-2.12.10.3 Shock mounting. Shock mounting of shipboard pumps is normally required to meet the requirements of MIL-S-901D and insure the survivability and reliability of the pump unit to operate under battle conditions.

# SECTION 3

# ROTARY PUMPS

## 503-3.1 ROTARY PUMP SAFETY PRECAUTIONS.

The following safety measures must be followed exactly to minimize hazards to personnel and equipment while operating rotary pumps:

a. Ensure that relief valves, where fitted, are tested and that they function at the designated pressure.

b. Never attempt to jack a pump by hand when the steam valve to the driving unit is open.

c. Do not tie down or otherwise render inoperative the overspeed trip, speed-limiting, or speed-regulating governors.

d. As specified in PMS, check the setting of speed-limiting where fitted. Ensure that settings limit the speed of the unit to rated speed under rated conditions and that rated speed is not exceeded by more than 5 percent for any loading condition.

e. Never operate a positive displacement rotary pump with a discharge valve closed unless the discharge is protected by a properly-set relief valve of a size sufficient to prevent a dangerous rise in pressure.

f. When inspecting or overhauling vertical rotary pumps, never rely on a flexible coupling to support the total pump rotor weight. Before working on pump rotors, support them with wire rope slings, block and tackle, or chocks.

g. Never close the suction valve while power is being supplied to a rotary pump prime mover. The lubricating fluid film on the rotors, idlers, and pump liners will be lost immediately, and this loss may cause catastrophic pump failure if not corrected immediately.

## 503-3.2 GENERAL DESCRIPTION.

503-3.2.1 Positive displacement rotary pumps are used for pumping all types of viscous fluids on naval ships. Rotary pumps are also used for gasoline or other low viscosity volatile fluids where a high suction lift is required and service is intermittent.

503-3.2.2 Rotary pumps achieve a high efficiency through close clearances and are sensitive to corrosive fluid or fluids containing abrasive solids; this factor is particularly important for rotary pumps handling distillate fuels because any entrapped hard solids can cause premature wear of pump rotors or their bores.

503-3.2.3 Care must be used to ensure proper fuel and lube oil management to prevent a minimum of solids passing through the pumps (see NSTM Chapter 541, Ship Fuels and Fuel Systems) and NSTM Chapter 262, Lubricating Oils, Greases, Specialty Lubricants and Lubrication Systems. In general, rotary pumps are self-priming.

## 503-3.3 ROTARY PUMP TERMINOLOGY.

503-3.3.1 The following terminology applies to rotary pumps:

a. Bearing A bearing supports or positions the shafts on which a rotor is mounted. A bearing may be internal (bearing wetted by the fluid being pumped) or external (bearing isolated from the fluid being pumped). The bearings may be either an antifriction (roller or ball bearing) or fluid film type (sleeve and journal).

b. Body The body is an external part which surrounds the periphery of the pumping chamber and which also may form one end plate. It is sometimes called a casing or housing.

c. Direction of rotation Drive shaft rotation is designated as ″clockwise″ (CW) or ″counterclockwise″ (CCW) as determined when viewing the pump from the driver end.

d. End plate An end plate is a part which encloses an end of the body to form the pumping chamber. One or more are used depending the construction of the pump. It is sometimes called a head or cover.

e. Fluids and liquids In this NSTM, the term ″fluid″ covers liquids, gases, vapors, and mixtures thereof. The word ″liquid″ is used only to describe true liquids that are free of vapors and solids. The word ″fluid″ is more general and is used to describe liquids that may contain, or be mixed with, matter in other than the liquid phase.

f. Gland A gland is a part which may be adjusted to compress packing in a stuffing box. It is sometimes called a gland follower.

g. Inlet or suction port One or more openings in the pump through which the pumped fluid may enter the pumping chamber.

h. Lantern ring A lantern ring is an annular ring located in a stuffing box to provide space between or adjacent to packing rings for the introduction of a lubricant or a barrier fluid, the circulation of a cooling medium, or the relief of pressure against the packing. It is sometimes called a seal cage.

i. Mechanical seal A mechanical seal is a device located in a seal chamber and consists of rotating and stationary elements with opposed faces. A rotating element is fastened and sealed to the shaft. A stationary element is mounted and sealed to the end plate or body. At least one element is loaded in an axial direction so that the seal faces of the elements are maintained in close proximity to each other at all times. Usually, the seal faces are highly lapped surfaces on materials selected for low friction and for resistance to corrosion by the fluids to be pumped.

j. Outlet or discharge port One or more openings in the pump through which the pumped fluid may leave the pumping chamber.

k. Pumping chamber The pumping chamber is the space formed by the body and end plate(s), into which fluid is drawn and from which fluid is discharged by the action of the rotor(s).

l. Radial seal A radial seal is a device in a seal chamber which seals on its outside diameter through an interference fit with its mating bore, and on the rotating shaft with a flexible, radially loaded surface. Radial seals include lip type seals, O-rings, V cups, U cups, etc., and may or may not be spring loaded.

m. Relief valve A relief valve is a mechanism designed to control or to limit pressure by the opening of an auxiliary passage at a predetermined pressure. A relief valve may either be integral with the body or end plate, or attachable. It may be adjustable through a predetermined range of pressures, or have a fixed setting. It may be designed to bypass the fluid internally from the pump outlet to the pump inlet, or externally through an auxiliary port. Terms commonly used in specifying relief valve performance are:

Cracking pressure - Sometimes called the set pressure, start-to-discharge pressure, or popping pressure the pressure at which the valve just starts to open. This pressure cannot be determined readily in a valve which bypasses the liquid within the pump.

Full-flow bypass pressure - The pressure at which the full output of the pump flows through the valve and the auxiliary passage.

Reseating pressure - The pressure at which the valve closes completely. This pressure is usually below the cracking pressure and is difficult to measure accurately when the liquid is bypassed within the pump.

Percent overpressure - Sometimes called the percent accumulation or percent regulation - The difference between the bypass pressure and the cracking pressure, expressed as percent of the cracking pressure.

n. Rotating assembly The rotating assembly generally consists of all rotating parts essential to the pumping action but also may include other parts specified by the manufacturer.

o. Rotor A rotor is a part which rotates in the pumping chamber. One or more are used per pump. It is sometimes referred to by a specific name such as gear, screw, or impeller, etc.

p. Seal chamber A seal chamber is a cavity through which a shaft extends and in which leakage at the shaft is controlled by means of a mechanical seal or a radial seal.

q. Stuffing box A stuffing box is a cylindrical cavity through which a shaft extends and in which leakage at the shaft is controlled by means of packing and a gland.

r. Timing gear A timing gear is a part used to transmit torque from one rotor shaft to another, and to maintain the proper angular relationship of the rotors. It may be outside the pumping chamber and is sometimes called a pilot gear.

s. Unloading valve A valve located downstream of the pumps discharge used to regulate and maintain the system pressure at a predetermined level. These valves are not attached to the pump and usually recirculate the fluid to the fluid source rather than the suction side of the pump. In many systems these valves enable the pump to operate in a standby mode.

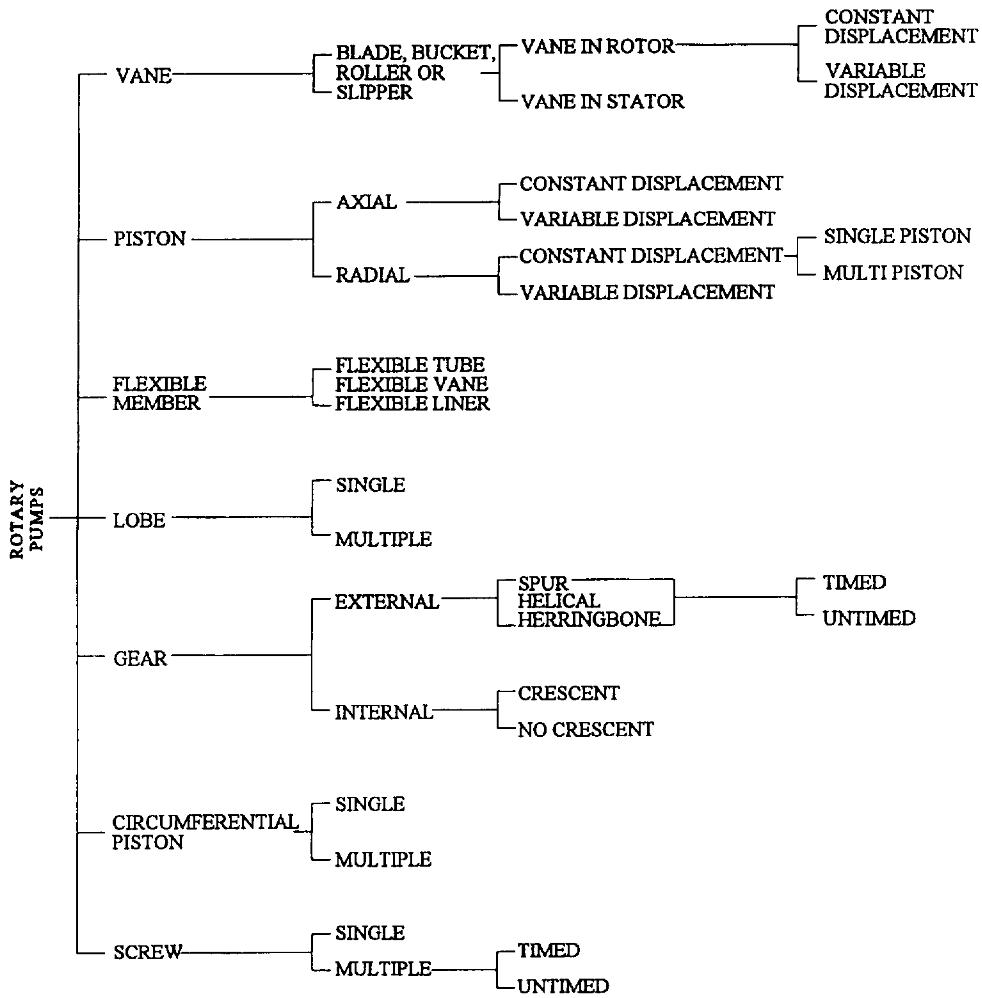



Figure 503-3-1. Types of Rotary Pumps.

## 503-3.4 DESCRIPTION

503-3.4.1 Rotary pumps are positive displacement pumps. They consist of a casing containing a rotating assembly running in close clearances and in such a manner that fluid is trapped in a moving space or chamber formed by the rotor and casing. As the rotor turns, the fluid is carried to the discharge side of the pump where it is pushed or squeezed from the pumping space to the discharge. The rotor assembly may consist of pairs of intermeshing gears, screws, lobes, cams or sliding pistons, vanes, blocks, or other forms. Rotor assembly position is maintained through the use or journal and thrust bearings. Rotor assemblies more common to the naval service are described in this section.

503-3.4.2 Classification and Types. Rotary pumps are classified by rotor types, shaft position or orientation, drive, or power types.

503-3.4.2.1 Rotary Pump Classification By Rotor Type. Rotary Pumps as classified by the Hydraulic Institute, ANSI/HI 3.1-3.5, are divided into seven types as in Figure 503-3-1. Classification of rotary pumps commonlyfall into five different types;

a. The gear type pump, which is the most common, consists of the external gear pump, including either spur, helical, and herringbone teeth; the lobular or impeller type; and the internal gear. Depending on the design and service, gear type pumps may be furnished either with or without timing gears. Gear type pumps carry the fluid being pumped between the gear teeth and the wall of the pump chamber, and is displaced when they mesh. All gear pumps have constant displacement characteristics.

b. The vane type pumps consist essentially of the sliding vane and the swinging vane or bucket type. Although the majority are designed for constant displacement, there are designs available in the sliding vane type where the displacement is controlled by a device which governs the amount of the vane movement. In vane type pumps the fluid being pumped fills a chamber formed by the vanes, rotor, and the casing wall and is discharged as the relative size of the chamber is reduced at the discharge side of the pump.

c. Screw pumps may have one, two, or three rotors with threads of various contours, depending on the design. The single screw type pump is more generally known as a progressive cavity type pump. Its two major components consist of a rotor and a stator. The rotor is a single external helix with a round cross-section, which is precision machined. The stator consists of a double internal helix, generally molded from a tough abrasion-resistant polymer within a steel tube. As the rotor turns within the stator, cavities are formed which progress from the suction to the discharge end of the pump, conveying the pumped material. The continuous seal between the rotor and the stator helices keeps the fluid moving steadily at a fixed flow rate proportional to the rotational speed of the pump. Depending on the design and service requirements of the pump, some two rotor pumps require timing gears whereas others do not. Three rotor screw type pumps do not use timing gears. All screw type pumps are of the constant displacement type. The fluid being pumped by a screw type pump is carried in the spaces between the screw threads and is displaced axially as the threads mesh.

d. The pumping action of the sliding shoe pump is derived from the rotation of three or more eccentric discs on a single rotor, each of which is closely fitted to a displacement chamber or shoe. The eccentric movement of each disc is comprised of horizontal and vertical components. The horizontal component of the disc's movement provides for the displacement action of the pump, and the vertical component of the disc's movement controls the entry or discharge paths or ″valving″ for the flow of the fluid. The movement of the shoe in the vertical direction alternately covers and uncovers the appropriate suction and discharge ports.

e. Multiple piston pumps may be of the radial or axial type. In either of these configurations, a series of pistons are actuated by a rotor mounted off center in the casing. Pumps of this type have very high volumetric efficiencies and are used a great deal in hydraulics. They are particularly well suited where variable displacement is required.

503-3.4.2.1.1 Timing Gears. As indicated in the above paragraph, several types of rotary pumps may be furnished with or without timing gears depending on their service requirements and design. The decision to use or not to use timing gears in a rotary pump must involve consideration of several advantages or disadvantages of each. The power transmission methods of these pumps is what differs. In pumps with timing gears, the rotors never come in direct contact with each other as the power is transmitted by their timing gears. In pumps without the timing gears, the power is transmitted by the power rotor to the idler rotor by way of the lubricating fluid film between the two rotors.

a. Pumps with timing gears have the advantages of having excellent suction capability. They can pump any fluid (including abrasives) and they can be run dry for short periods of time. The disadvantages of having timing gears are that they are more expensive, they may not be repairable, they are generally pressure limited, and the design becomes far more complex because of the requirement of additional seals.

b. Pumps which do not use timing gears have the advantages of higher speed, good efficiency, lower noise, and generally longer life. The disadvantages of a pump without timing gears are that they are moderately expensive because of the close tolerances in the rotor mesh, and the rotor material requirements may be limited.

503-3.4.2.2 Other Rotary Pump Classifications. Rotary pumps are also classified by their shaft position or orientation (horizontal, vertical, or inclined) and by power types (motor or turbine driven).

503-3.4.3 Service Application. Rotary pumps are used in the naval service for these principal applications (which are discussed in paragraph 503-3.6).

503-3.4.3.1 Fuel Services - Clean.

a. Main Fuel Service - Steam Turbine Driven Ships

b. Fuel Booster Service - Gas Turbine Driven Ships

c. Port Fuel Service - Steam Turbine Driven Ships

d. Diesel Oil Service

e. JP-5 Service

503-3.4.3.2 Fuel Services - Contaminated.

a. Fuel Stripping

b. Fuel Transfer

c. Cargo Stripping

d. JP-5 Transfer

e. JP-5 Stripping

503-3.4.3.3 Lubricating Oil Pumps.

503-3.4.3.4 Heavily Contaminated Seawater.

a. Oily Wastewater Transfer

b. Bilge Stripping

## 503-3.5 ROTARY PUMP CHARACTERISTICS.

503-3.5.1 General Characteristics. Rotary pumps are a positive displacement type whose capacities are directly proportional to pump speed subject to capacity losses as discussed in paragraph 503-3.5.2 through paragraph 503-3.5.9. Rotary pump pressure developed is equivalent to that imposed by resistance of the system to which it discharges. The pressure capability of the pump is generally only limited by the pump casing bursting strength or the pumping power available. It is for this reason that rotary pumps are fitted with relief valves.

Rotary pumps operate best at moderate capacities and heads. Normally, heads are limited to 500 PSI or less, but they have been used for applications up to 2500 PSI. Efficiencies are normally in the 70-80% range. These pumps find their principle applications in the handling of viscous fluids such as fuel oil, lube oil, and oily waste. There are several advantages and disadvantages encountered in the use of rotary pumps: Advantages:

a. Good for handling high viscosity fluids

b. Self priming

c. Can produce suction lifts from 20-25″ Hg

d. Can be directly driven by electric motors (but must be gear driven from a steam turbine)

e. Require low maintenance

f. Are relatively compact Disadvantages:

a. Operate at a constant capacity for a given RPM

b. Require an auxiliary pressure regulating device or relief valve to prevent excessive pressures

c. Require that close tolerances be maintained between pumping elements and the casing

d. Cannot operate at shutoff

e. Require a constant supply of lubrication from the fluid being pumped

f. Cannot handle gritty fluids

503-3.5.2 Displacement. Rotary pump displacement is the volume that the rotating elements displace during each rotor shaft revolution. Displacement is independent of all operating conditions and is a theoretical capacity (assuming complete filling of the pumping spaces and no internal recirculation losses).

503-3.5.3 Capacity. Rotary pump capacity is the quantity of fluid actually delivered under specified conditions. Capacity is equal to the displacement times the revolutions per minute minus losses due to slip or inlet conditions. The ratio of the actual capacity to the displacement expressed in percent, is known as the pump's volumetric efficiency.

503-3.5.3.1 Slip. Slip is the quantity of fluid that bypasses from discharge to suction through the internal working pump clearance. Slip varies directly with pressure and inversely with viscosity.

503-3.5.3.2 Inlet Conditions. Fluid conditions at the pump suction are important to the pump performance, and include suction pressure or lift, viscosity, operating temperature, vapor pressure, and the amount of entrained or dissolved air or gas. Capacity losses resulting from inlet conditions which do not permit complete pumping element filling are not considered to be slip.

503-3.5.4 High Suction Lift Effect. Rotary pumps in like new condition will lift oil, as well as reciprocating pumps. The volumetric efficiency will drop off rapidly if the pump is required to operate on a vacuum or suction lift exceeding that for which it is designed. This is due to the effects of cavitation. Figure 503-3-2 is a representative curve showing a typical rotary pump's drop in volumetric efficiency. Different designs will display unique curves showing these same general trends. The upper curve A shows that with the pump handling viscous oil (700 SSF), the capacity is maintained well above the rated capacity of 250 gpm at 25 in. hg. When the pump is handling the same oil at a lower viscosity (higher temperature - 70 SSF), the lower capacity curve B drops off abruptly at more than 19 to 20 in. hg suction lift. This capacity reduction is caused in part by increased slip but to a greater extent by oil deaeration and vaporization.

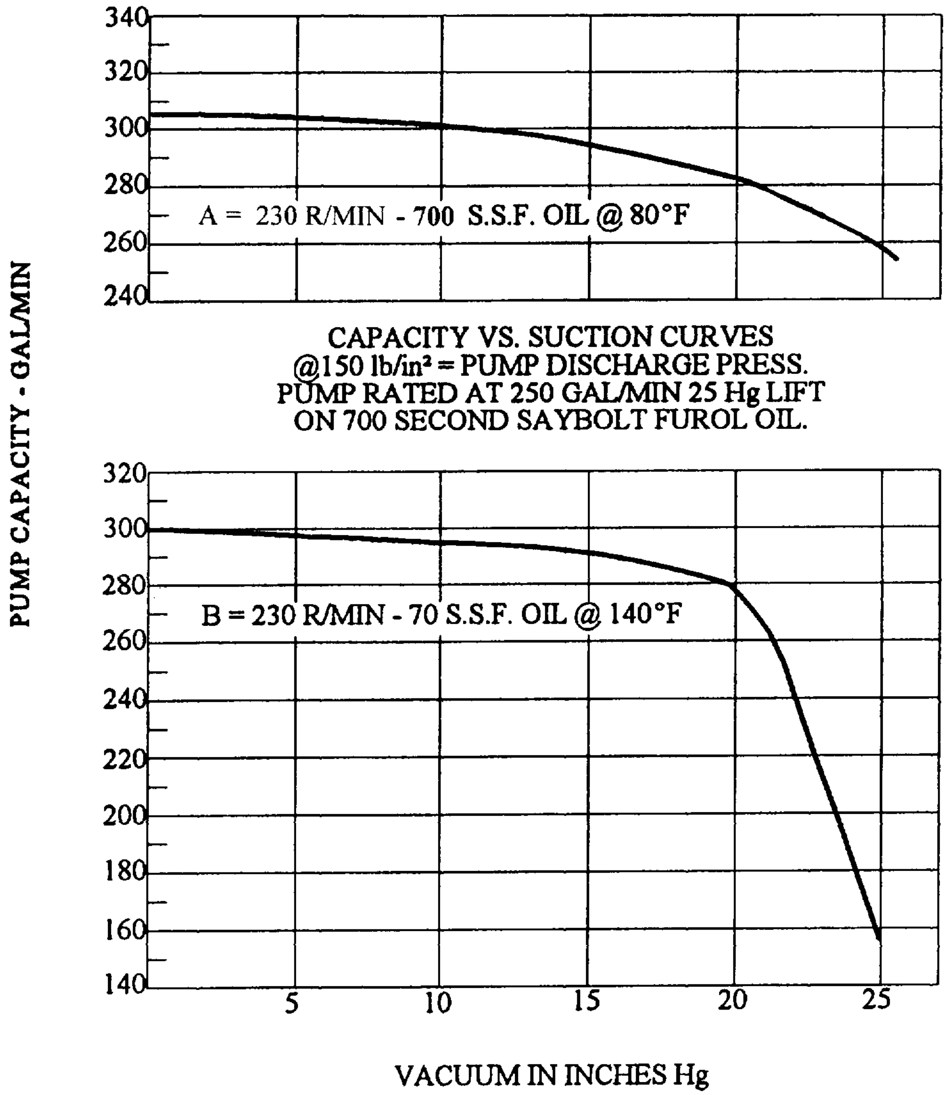



Figure 503-3-2. Contrasting Capacity Curves.

503-3.5.5 Entrained Gas Effect. Most liquids are susceptible to air or gas entrainment (the trapping of bubbles in the liquid). Air entrainment is common in systems where recirculation occurs and the liquid is exposed to air through either mechanical agitation, leaks, or improperly located drain lines. Entrained air or gas in liquids handled by rotary pumps has an effect on pump performance, both mechanically and hydraulically, especially where negative inlet pressure (suction lift condition) exists. When the inlet pressure is below atmospheric pressure, entrained gas in the fluid will expand and take up a larger part of the pump displacement, thereby reducing its liquid capacity. The effect of entrained gas on liquid displacement is shown in Figure 503-3-4. If the entrained air or gases is a large percentage of the volume handled, there may be noise and vibration, loss of liquid capacity, and pressure pulsations. The pulsation frequency depends on the speed and number of closures in the rotor element during each revolution. Main reduction gear lube oil pumps are particularly susceptible to air entrainment.

503-3.5.6 Dissolved Gas Effect. Many liquids contain dissolved air or gas in the solution. The solubility of air or gas in liquids varies with the type of liquid and the pressure to which it is subjected. For example, lube oil at atmospheric pressure and temperature may contain up to 10 percent of dissolved air by volume. Dissolved air or gas in liquids handled by rotary pumps has a similar effect on pump performance, but not as significant, as entrained air or gas as described in paragraph 503-3.5.5. The effect of dissolved gas on liquid displacement is illustrated in Figure 503-3-3.

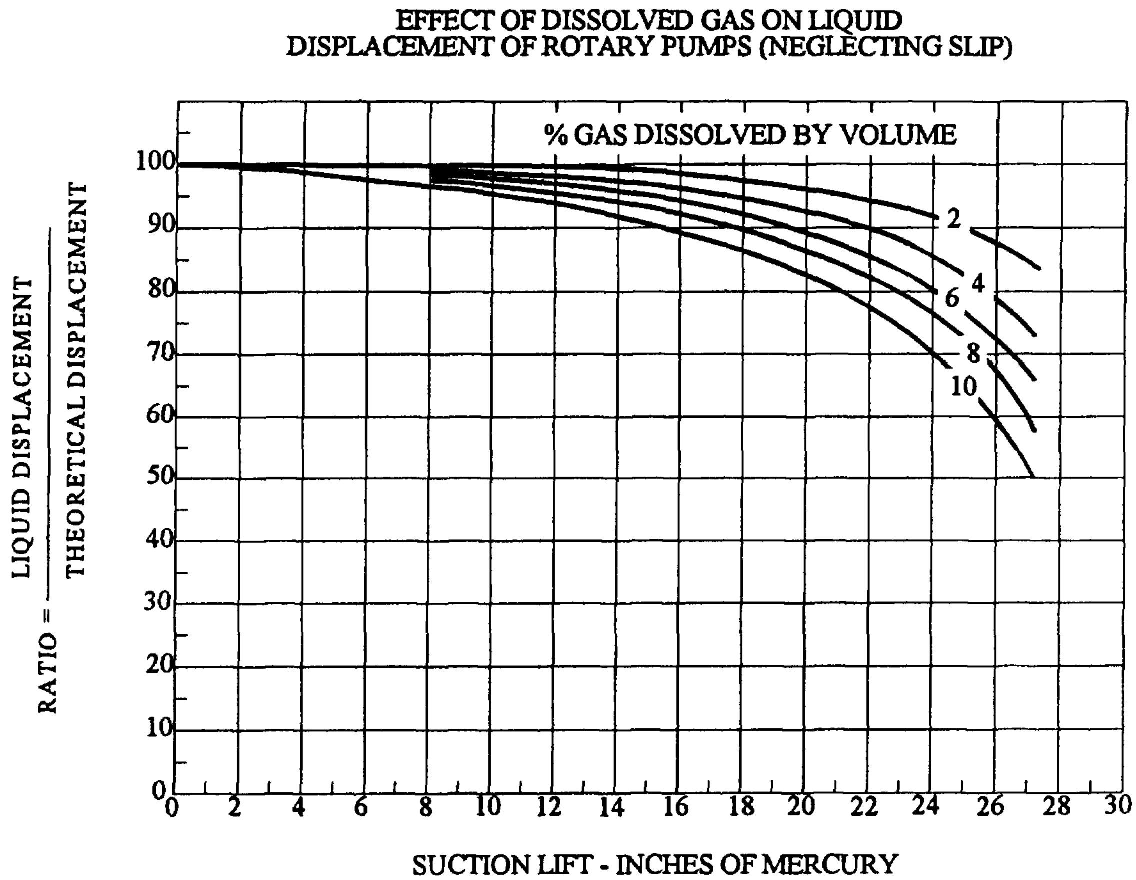



Figure 503-3-3. Effect of Dissolved Gas on Liquid Displacement.

503-3.5.7 Viscosity Effect. Viscosity is the internal resistance of a fluid to a shearing stress, or more simply, it can be thought of as the fluid's resistance to flow. It can be measured in a variety of units. In most liquids handled by shipboard pumps, the viscosity decreases with an increase in temperature. The effect of an increase in the viscosity of a fluid on a pump is that it will increase the mechanical pump losses, reduce the pump efficiency, and result in an increase in the power required to pump the fluid at the same flow rate.

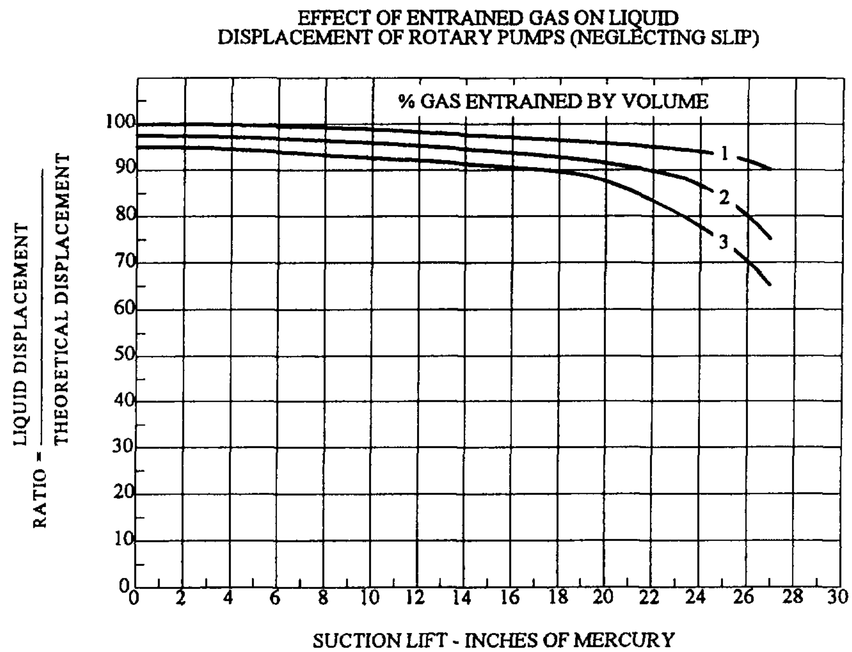



Figure 503-3-4 Effect of Entrained Gas on Liquid Displacement.

Viscosity is measured either as a dynamic (absolute) or kinematic parameter. The difference between these two parameters is a factor of the mass density of the fluid. The units for the measurement of dynamic viscosity are Pound Seconds per Square Foot (English) or Centi-Poise (Metric). The units for the measurement of kinematic viscosity are Square Feet per Second (English) or Centi-Stoke (cSt, Metric). Other arbitrary units for the measurement of kinematic viscosity exist. These are: Seconds Saybolt Universal (SSU) and Seconds Saybolt Furol (SSF).

The following relationships/conversions exist:

cSt = 2.240 SSF - 184/SSF (SSF = 25 to 40)

cSt = 2.160 SSF - 60/SSF (SSF> 40)

cSt = 0.226 SSU - 195/SSU (SSU< or = 100)

cSt = 0.220 SSU - 130/SSU (SSU> 100)

The major effects of a change in viscosity are:

a. Slip will increase with a decrease in viscosity. In some cases the manufacturer will down rate a pump for pumping oils at less than 100 SSU.

b. As the viscosity of the pumped fluid increases, the mechanical losses increase, the pump efficiency decreases, and the pumping power requirement increases.

c. It may be necessary to slow a pump when pumping highly viscous fluids to prevent overloading the driver and to permit sufficient time for the fluid to flow into the pumping elements of the pump (prevent cavitation).

d. The NPSHR will be increased when pumping more viscous fluids.

503-3.5.8 Bearings. Rotary pump bearings are classified as either external or internal. External bearings are separated from the fluid pumped by shaft packing or mechanical seals. Since fuel (diesel) has low viscosity, seals in pumps in fuel service must be carefully installed to prevent leakage (see Section 5). Internal bearings are located between the rotor elements and the packing and depend on the pumped fluid for lubrication. Internal bearings are not suitable for pumps handling fluids containing abrasives or having low viscosity (lacking lubricity).

503-3.5.9 Gland Sealing. If pumps are required to take suction at lower than atmospheric pressure and have a packed stuffing box, it is important that the shaft stuffing boxes be fitted with a lantern ring and sealed by the pumped fluid (see Section 5). These precautions prevent entrance of air into the pump. Stuffing box gland sealing fluid is taken from the pump discharge.

**503-3.6 APPLICATIONS.**

503-3.6.1 Fuel Services-Clean. Clean Navy Distillate Fuel per MIL-F-16884 (NATO, F-76) or JP-5, MIL-T-5624 (NATO, F-44) is to be provided to the ships boilers, diesel engines, diesel generators, gas turbine propulsion engines, gas turbine generators and aircraft refueling stations. This is accomplished through the use of rotary positive displacement pumps. Most of those in use are MIL-P-19131 Type II, III, X, or XI. The fuel viscosity range is from 32 Saybolt Second Universal (SSU) to 1500 SSU. Specific services are detailed in the following sub-paragraphs:

503-3.6.1.1 Boiler Fuel Service Pumps - Steam Turbine Driven Ship Classes. Two steam, turbine driven or motor driven two-speed screw type pumps are typically used in the boiler fuel service system to provide the boiler furnace with clean Diesel Fuel Marine (DFM). These pumps may have to operate at a suction pressure of up to 10 inches and a discharge pressure of up to approximately 350 - 450 psig depending upon the particular plant operating requirements.

503.3.6.1.2 Fuel Service Pumps - Gas Turbine Ship Classes. There are typically two motor-driven, dual (high/low) speed, rotary vane, or screw-type Fuel Service Pumps per propulsion plant designed to provide clean fuel to the gas turbine module through a common header to the fuel service heater. These pumps also supply fuel to gravity tanks. Built-in relief valves and a system unloading valve are designed to prevent the pump from over pressurizing the system.

503-3.6.1.3 In-Port Fuel Service Pumps - Steam Turbine Driven Ships. Two-speed screw-type (Port/Cruising) fuel pumps are found on some ships, and serve as lighting off and low load in-port service pumps. These pumps have bypass or unloader valves installed in the system to return fuel from the burner line to the pump suction.

503-3.6.1.4 Diesel Oil Service Pumps. Diesel oil service may be provided by most of the rotary, positive displacement type pumps; however, screw type, gear type, and vane type make up the bulk of those in service. Space and weight factors play a prime part in selecting the correct pump. On most small or medium sized engines these pumps are usually attached directly to and driven by the engine.

503-3.6.1.5 JP-5 Service Pumps. Typically a rotary screw or sliding vane type pump is used to take suction from the JP-5 service tanks and discharge to the aircraft fueling station

503-3.6.2 Fuel Services - Contaminated. Pumps used to handle contaminated fuel are subject to abrasive fluids and solid contaminants. Pump materials are extremely important in these situations. Some pumps used on ships with compensated fuel systems have been provided with stellite liners to improve reliability. Typically the viscosity range will be from 32 Saybolt Seconds Universal (SSU) to 4000 SSU and the following pumps will be included:

503-3.6.2.1 Fuel Stripping Pump. Sliding vane (type X) pumps or rotary screw (type II) pumps with timed rotors which prevent metal-to-metal contact are most often used to pump the fuel, sludge, seawater and sediment that are found in the bottom of the fuel tanks.

503-3.6.2.2 JP-5 Stripping Pump. Electric motor driven sliding vane (type X) pumps or sliding shoe (type XI) pumps are most often used to pump contaminated JP-5 to the JP-5 Stripping Tank or to the JP-5 transfer main through closed locked valves.

503.3.6.2.3 Cargo (F-76 or JP-5) Stripping Pumps. Two electric, motor-driven sliding vane (type X) pumps or sliding shoe (type XI) pumps operating in a suction lift condition are connected in parallel to a common suction header and discharge header. The stripping pump takes suction from the tank stripping manifolds and each JP-5 Pump casing and suction line. They also act as continuous priming pumps for the centrifugal main cargo pumps.

503-3.6.2.4 Fuel Transfer Pump. Rotary screw-type pumps with or without timing gears and sliding vane type pumps are most often chosen for fuel transfer pumps. These pumps take suction of fuel storage tanks and discharge to fuel service tanks, other fuel storage tanks, or to the deck riser for inter ship transfer. On gas turbine ships with compensated fuel systems, the fuel transfer pump discharges to a fuel oil purifier.

503-3.6.3 Lubricating Oil Pumps.

503-3.6.3.1 Main Reduction Gear Pumps. The main reduction gear lubricating oil pumps are typically vertical screw type pumps with or without timing gears. These pumps may be steam turbine driven, motor driven (two speed), or attached to and driven from the main reduction gear. The standby and emergency lube oil pumps are also of the same design and driven by a two speed motor. The steam turbine driven pumps and the electric motor driven pumps can be operated in both the automatic and manual modes. Motor driven pumps can normally be started and stopped remotely.

On some ships there are air driven coast down pumps to provide continuous oil supply to the gears and bearings when there is a loss of electrical power.

Lube oil pumps operate at maximum capacity regardless of system operating conditions or demands, and frequently are required to pump air-entrained oil. Pumping this air/oil mixture may result in a pulsating oil pressure, causing critical casualties.

503-3.6.3.2 Lube Oil Pumps for Auxiliary Turbine Driven Equipment. Lube Oil pumps on auxiliary turbine driven equipment (Feed Pumps, Condensate Pumps, Fuel Oil Service Pumps, and Main Circulating Pumps; e.g.) typically are gear type pumps which are driven directly from the turbine reduction gear and provide bearing oil, gear oil, and governor operating oil. Typically the viscosity range will be from 32 Saybolt Seconds Universal (SSU) to 4000 SSU.

503-3.6.4 Contaminated Waste Fuel/Oil/Water Pumps.

503-3.6.4.1 Oily Waste Transfer Pump. Oily Waste Transfer Pump takes suction from the oily waste suction main. The suction main has connections for bilge wells, oily waste bilge collection tanks, waste oil tanks, hose connections and oily waste holding tanks. The oily waste transfer pump discharges to the oily waste holding tanks or to overboard discharge connections. Typically electric motor-driven sliding shoe pumps are used.

503-3.6.4.2 Bilge Stripping Pumps. Bilge stripping pumps take a suction from the bilges and discharge to overboard connections or oily waste holding tanks. Typically steam reciprocating bilge stripping pumps are used for this service. However new or replacement bilge stripping pumps shall not be steam driven.

503-3.6.4.3 Portable Oily Waste Transfer Pumps. Portable oily waste transfer pumps are typically pneumatic powered double diaphragm pumps which take suction directly from bilge wells and discharge to the oily waste holding tanks or overboard connections.

503-3.6.4.4 Oil/Water Separator Units. Oil/Water separator unit pumps take suction from the oily waste holding tank and either draw through or discharge to the oil/water separator units. The units discharge to a oily waste holding tank, waste oil tank, or to an overboard discharge connection. Typically Progressing Cavity (type I) or flexible vane pumps are used for this service.

503-3.6.5 AFFF Service. AFFF pumps take suction from AFFF service tanks and discharge to the AFFF service system. Typically, sliding vane pumps are used for this service

## 503-3.7 ROTARY PUMP OPERATION.

Refer to the Engineering Operation Sequence System (EOSS) or the pump technical manual for specific rotary pump preoperation checks, starting procedures, and stopping and securing procedures. Instructions for all pumps cannot be covered here because of the diverse types, designs, and applications in the fleet. The applicable technical manual should be studied before any attempt is made to operate or service the unit. The following general caution should be observed.

**CAUTION**

**A rotary pump is a positive displacement pump and must not be started against a closed or restricted suction or discharge valve.**

## 503-3.8 TROUBLESHOOTING ROTARY PUMPS.

Trouble shooting of the pumping units (Pump and Motor) and the system operation is inter-related. Those general and more common re-occurring problems requiring troubleshooting are addressed herein. Engineering Operation Casualty Control (EOCC) procedures and Engineering Operating Sequencing System (EOSS) procedures which are ship specific take precedence in the event of any casualty.

In the event of any casualty or unusual operating characteristic, the primary course of action should be to report the problem to the immediate supervisor, followed by an investigation into the cause of the problem and then the correction of the problem. In most cases it is better to place a stand-by or alternate unit on the line and secure the affected unit until a thorough investigation can be made to determine the cause of the problem.

503-3.8.1 No Liquid Delivered. If the pump fails to discharge fluid when first started, stop the pump and investigate the probable causes as follows:

a. System valves improperly aligned. Verify valve alignment in accordance with EOSS.

b. Pump suction strainer or filter element (if installed). Clean as necessary.

c. Pump casing, suction lines and mechanical seal or packing for air leakage into pump casing.

d. Check suction tank level and ensure pump is properly primed.

e. Ensure bypass relief valve is properly adjusted and not being held off its seat by foreign matter/debris. Adjust the relief valve cracking pressure or clean the debris out as necessary.

f. Correct pump rotation. Reverse any two motor leads, if required, to correct rotation.

g. Suction lift is too high. Check the suction with a vacuum gage. Fluid may be vaporizing.

503-3.8.2 Low Capacity. If the pump discharges but does not deliver sufficient capacity, the cause may be due to the following:

a. Improper valve alignment. Verify that the system is aligned in accordance with EOSS.

b. Pump speed too slow. If multiple speed motor is used, ensure that the proper speed has been selected. If a turbine driven unit is used and normal operating steam pressure is available, reset the pressure regulating or speed limiting governor. Determine if the pump driver is requiring excessive power. See paragraph 503-3.8.4.

c. Partially plugged suction line or strainer/filter. Check pump suction strainer or filter element (if installed) for blockage. Clean if required.

d. Air leaking into pump casing. Check pump casing, suction lines, and mechanical seal or packing gland for air leakage. Repair as necessary.

e. Low suction tank level. Check suction tank level to ensure pump is properly primed. Change tank suction or fill tank to levels above the suction pipe

f. Bypass, relief, or unloader valve open. Check the bypass to ensure it is closed and check the relief or unloader valve to ensure that it is properly adjusted and not leaking due to foreign matter or debris between the seat and disc.

g. Pump rotation is incorrect. Verify the rotational direction, and, if required, change any two pump motor leads to correct pump rotation.

h. Suction lift is too high. Check the suction with a vacuum gage. Fluid may be vaporizing.

503-3.8.3 Loss of Suction. If the pump delivers fluid for a while and then loses suction or fails to deliver, the cause may be due to the following:

a. System misalignment or pump suction valves not fully opened. Verify valve alignment in accordance with EOSS.

b. Pump suction line, strainer, or filter plugged (if installed). Clean as necessary.

c. Pump casing, suction lines and mechanical seal or packing gland leaking air into the system. Inspect for air leakage. Repack or repair as required.

d. Low suction tank level. Change tank suction or fill tank to a level above the suction pipe.

e. Suction lift is too high. Check the suction with a vacuum gage. Fluid may be vaporizing.

503-3.8.4 Excessive Power Required. If the pump sounds as if it is struggling to meet its demand, or its discharge rate remains low, it may be indicative of an excessive power requirement being placed on the driver. If possible, the power requirements of the pump should be determined by the steam inlet pressure or the motor amperage, as applicable. If an excessive power requirement is determined to exist, then this condition may be due to:

a. Mechanical defects such as a bent shaft, rotor binding, mechanical seal failure or stuffing box packing too tight, casing distortions caused by piping connections, foundation misalignment, or internal pump part failure (bearings, timing gears, etc.).
b. Suction or discharge lines are obstructed. Check pump suction and discharge pressures. Verify system valve positions. If time allows record all operating parameters for post operation analysis (to include pump speed, fluid viscosity, fluid temperature, and motor amperage or steam inlet pressure, as appropriate). If unable to determine the cause, place a standby or alternate pump unit on line and secure the affected unit.
c. Pump speed is too high. If possible, reduce motor speed by pressing low speed switch or adjusting pressure regulating or the speed limiting governor on steam powered units.
d. Fluid being pumped is (cold) heavier or more viscous than specified. Reduce pump speed.

503-3.8.5 Vibration or Noise. Excessive pump noise or vibration may be due to the following:

a. Pump foundation or resilient mount bolts are loose. Correct if necessary.
b. Mechanical defects such as misalignment, bearing failure, bent shafting, rotor interference. Correct as necessary.
c. Rotor out of balance. Repair as necessary.
d. Air leaking into suction. Check pump mechanical seal or packing and suction piping.
e. Air or gas entrained in fluid. Vent pump discharge or suction as necessary.
f. Fluid vaporizing in the suction. Reduce suction lift.
g. Relief valve chattering. Reset cracking or reseating pressure.

503-3.8.6 Pump Wears Rapidly. If any pump wears rapidly, this could be due to:

a. Misalignment. Investigate and repair as needed.
b. Pump running dry or with insufficient fluid. When the pump repairs have been completed, ensure that the pump has been properly primed and vented and is operated with suction from a tank which has sufficient level and NPSHA to prevent a loss of fluid at the pump suction.
c. Contaminated fluid entering pump through dirty strainer or filters (if installed). Clean strainer, or replace filters, and purify fluid being pumped.
d. Pump internal components made of materials which are incompatible with the fluid being pumped. Repair as necessary.
e. Pump is operating at greater-than-rated pressure. Check pump discharge valve and piping for alignment or obstruction and correct as necessary.
f. Pump operating with fluid of lower than specified viscosity. Check operating requirements and correct as necessary.

503-3.8.7 Short Life of Mechanical Seals and Packing. Most pumps being designed and produced today use mechanical seals as the preferred means of sealing the pump shafts against air leakage into the casing. Packing is still used on existing units in the fleet, but considered to be the secondary means of sealing the shaft opening in the casing. Pumps with packing are being replaced mechanical seal units at overhaul during the life cycle schedules. Packing is also used in an emergency in the event of mechanical seal failure. Short life of mechanical seals or packing may be due to:

a. Misalignment. Check alignment and correct as necessary.
b. Mechanical seal or packing improperly installed. Replace mechanical seal or packing in accordance with the technical repair standard, Preventive Maintenance System (PMS) Maintenance Requirement Card (MRC) or the technical manual.
c. Wrong packing (packed units only). Replace packing with approved packing.
d. Packing too tight preventing lubrication of the packing and shaft. Repack packing gland and adjust leakoff.
e. Worn bearings. Repair or replace as required.
f. Worn or scored shaft sleeve at packing area. Replace shaft sleeve and repack pump.
g. Bent shaft. Repair or replace as required.
h. Interference between stationary and rotating parts. Repair or replace as required.
i. Rotor out of balance. Balance rotor as required.
k. Contaminated sealing water to packing gland or seal. Identify source of contamination and correct as required.
l. Pump speed too high. Reduce pump speed if possible.

503-3.8.8 Short Life of Bearings. If bearing failures are recurring on short interval check for the following:

a. Misalignment between pump and motor, as well as piping connections and foundation/resilient mount bolts. Check alignments and correct as required.
b. Bearings improperly lubricated. Lubricate in accordance with the PMS or technical manual.
c. Bearing incorrectly installed. Correct as necessary.
d. Bent shaft. Correct as necessary.
e. Interference between rotating and stationary parts. Correct as necessary.
f. Internal part failure causing excessive thrust. Correct as necessary.
g. Rotor out of balance. Correct as necessary.

503-3.8.9 Overheating and Seizing.

a. Misalignment.
b. Packing installed improperly or wrong packing installed. Repack pump in accordance with PMS or technical manual.
c. Bearings improperly lubricated or dirt in bearings. Replace and lubricate bearings.
d. Packing gland too tight preventing lubrication of packing and shaft. Repack pump gland.

e. Mechanical failure inside pump causes excessive thrust. Repair as necessary.

f. Bent shaft. Correct as necessary.

g. Rotor out of balance. Correct as necessary.

h. Interference between stationary and rotating parts. Correct as necessary.

503-3.8.10 Suction Line Vapor Lock. When a pump is in good condition and builds a good vacuum on suction but does not build discharge pressure after all other possible problems are corrected, vapor lock in suction piping should be investigated. Investigation should be initiated particularly in long lengths of fuel suction piping and especially after new oil cargo has been received or the oil has been heated. Shift suction to another tank or group of tanks nearer the pump; usually the pump will build pressure immediately.

503-3.8.10.1 The lighter hydrocarbons such as gasoline have much higher vapor pressure than heavier fuels at the same temperature. If heavy pounding of the pump occurs when pumping fuel, the vacuum on the suction side of the pump usually is excessively high or the temperature of the fuel is high for the type of fuel being pumped. These factors cause partial vaporization in the suction line or pump casing.

503-3.8.10.2 In measuring suction lift, first measure absolute suction pressure available. Ensure that the resulting absolute suction pressure is greater than atmospheric pressure minus the vapor pressure at the pumping temperature plus 2 lb/in $^2$ (approximately 4.1 in. hg). For example: a pump which is pumping gasoline with a vapor pressure of 11 lb/in $^2$ absolute should have an absolute suction pressure available greater than $14.7 - (11 + 2) = 1.7\ lb/in^2$ or a suction lift less than 3.5 in. hg (conversion from lb/in $^2$ to in. hg):

$$1.7 \text{ lb./in}^2 \times 2.03 \text{ in. Hg/lb/in}^2 = 3.46 \text{ in. Hg}$$

503-3.8.10.3 Vapor lock is most apt to occur in pockets where the suction line overpasses other piping or ship structure, particularly in the pump proximity. It will be difficult for the pump operator to determine whether malfunction resulting in noise and pounding is caused by deaeration or vaporization; however, decreasing the suction lift by reducing pump speed, for example, will alleviate vapor lock. If the pump suction line has a vent line installed, the vapor from the suction can be removed by opening the vent valve and then closing it once the vapor is released.

## 503-3.9 MAINTENANCE.

503-3.9.1 General Guidance. Information provided here supplements preventive and corrective PMS maintenance procedures. Maintenance procedures shall be as specified on MRCs. If inconsistencies are noted, submit OPNAV 4700/7, PMS Technical Feedback Report (TFBR).

503-3.9.2 Using Technical Manuals. Instructions contained in pump technical manuals should be followed in rotary pump maintenance. Instructions in this chapter are of a general nature and are intended to supplement equipment technical manuals.

503-3.9.3 Wearing Plates and Liners. Both the design clearance and the maximum allowable clearance between pump rotors and casing wearing plates and cylinder liners are shown on manufacturers' drawings and in the equipment technical manual. When actual clearance exceeds allowable limits, parts should be renewed in accordance with PMS and the technical manual requirements. Failure to renew parts and restore design clearance when allowable clearances are exceeded will result in pump capacity and efficiency decrease.

The allowable clearance for low pressure, low suction lift pumps (for example, lubricating oil, fuel booster, and tank drain pumps) is larger than the allowable clearance for medium and high pressure pumps (for example, boiler fuel service pumps). As the pressure drop across the clearance space increases, closer tolerances must be maintained to prevent degraded pump performance. Similarly, as suction lift increases, the allowable clearances must be tighter so that the pump will be able to pull a vacuum and lift the fluid into the pump. Increased pressures and internal clearances result in a greater loss of pump capacity due to increased pump internal fluid recirculation or slip. Materials for wearing plates and liners shall be compatible with the system usage and may include stellite or ceramic materials with NAVSEA approval.

503-3.9.4 Thrust Bearings. Thrust bearings should be inspected, and clearances checked in accordance with applicable PMS and technical manual. Bearings shall be replaced, not rebabbitted.

503-3.9.5 Timing Gears. Some pump types may have timing (or synchronizing) gears which are fitted to rotor shafts to maintain correct clearances between the two pumping rotors during operation. To maintain correct clearances, gears must be locked to the rotor shaft in correct position to maintain the clearance between rotor elements as shafts make a complete revolution; therefore, no lost motion is permissible adjacent to keys or pins holding rotors and gears in position on the shafts.

503-3.9.6 Motors and Controllers. Instructions in NSTM Chapter 302, Electric Motors and Controllers, should be followed for motor-driven unit maintenance.

503-3.9.7 Turbines. Instructions in NSTM Chapter 502, Auxiliary Steam Turbines, should be followed for turbine-driven unit maintenance.

503-3.9.8 Pump Pressure-Regulating Governors. Boiler fuel service, and lubricating oil pumps, are fitted with adjustable constant-pressure-type pump governors in addition to turbine governors described in paragraph 503-2.9 and NSTM Chapter 502, Auxiliary Steam Turbines.

503-3.9.9 Tests and Inspections. Rotary pumps should be tested or inspected periodically in accordance with PMS requirements.

503-3.9.9.1 Typically maintenance requirements include both the driver and the pump. Drivers (motors and turbine) are discussed in their applicable NSTMs and PMS. The following list is representative of the pump end maintenance requirements and do not include special Start-up or lay-up requirements Periodicity requirements are stated in the technical manual and PMS.

a. Rotating the pump by power or hand to insure freedom of rotation
b. Inspecting the mechanical seal for leakage
c. Lubricating pump bearings
d. Inspect resilient mounts and grounding straps
e. Lubricating flexible coupling
f. Renewing oil in speed increaser/reducer (as applicable)
g. Testing operating relief valve
h. Inspecting shaft alignment and mounting bolts

i. Conducting operational performance testing

j. Checking internal clearances (bearings, plates, liners, rotors, timing gears, casing throat bushings, diffusers, and gears, etc.).

## 503-3.10 OVERHAUL OR REPAIR.

Rotary pumps are to be overhauled and/or repaired in accordance with the requirements of General Specifications for Overhaul of Surface Ships (GSO), the applicable Technical Repair Standard (TRS), and the equipment technical manual.

503-3.10.1 Assembly Drawings. When repairing or making an interior pump examination, it is essential that all PMS performance test data, drawings and available dimensional data be at hand. Frequently, alterations occur in such important dimensions as clearance between rotors and casing wearing plates and liners, and shaft seal or gland adjustments, resulting in poor operation. This problem will persist in spite of other major repairs until the real cause is corrected. For these reasons, unauthorized field conversion or parts substitution should be avoided.

503-3.10.2 Renewing Wearing Plates and Liners. Whenever the pump casing is opened, the clearance between casing liners and plates and various parts of the rotors and shaft should be measured to determine if excessive wear has taken place and if renewal is necessary. If bearings are worn excessively, it is reasonable to expect that liner replacement is necessary. Liners should not be renewed unless worn pump bearings are restored to their original readings. The required bearing oil clearance is usually given on the manufacturers' plans. In the absence of this data, the table of tolerances and clearances provided in Table 503-2-3 should be followed.

503-3.10.3 Repairs using polymeric compounds are prohibited for most if not all PD pump applications.

# SECTION 4

# RECIPROCATING PUMPS

## 503-4.1 RECIPROCATING PUMP SAFETY PRECAUTIONS.

Adherence to the following prescribed precautionary practices will help ensure safe operation of reciprocating pumps.

a. Never use a jacking bar to start a pump while the steam valve to the pump is open.

b. Except in an emergency, boiler feed pumps shall not be used for purposes other than those connected with the service of the boilers or use of feedwater.

c. Before opening a steam cylinder or steam valve gear ensure that drains are open and that steam and exhaust root valves are wired closed and tagged DO NOT OPEN.

d. Before opening the water cylinder or valve chest of a pump handling water at a temperature in excess of 48.9°C (120°F), ensure that suction and discharge valves are wired closed and cylinder and valve chests are drained.

e. Always open steam cylinder drain valves when the pump is shut down and leave them open until the pump is again started and cleared of condensation.

f. Never attempt to operate reciprocating pumps with the discharge closed

## 503-4.2 TYPES OF RECIPROCATING PUMPS.

503-4.2.1 Reciprocating pumps are positive displacement pumps that are divided into three general types: steam, power, and controlled volume.

503-4.2.2 A further breakdown of the different types is shown in Figure 503-4-1. The construction of a pump is usually divided into two parts: liquid end and drive end. The liquid end is the end of the pump that actually pumps the liquid. The drive end of a power pump contains the necessary mechanism to convert rotary input energy into reciprocating linear pumping energy. The drive end of a steam pump contains the steam piston that provides the force to drive the liquid end piston. Steam pumps are still utilized as emergency boiler feed pumps on a number of steam driven naval vessels.

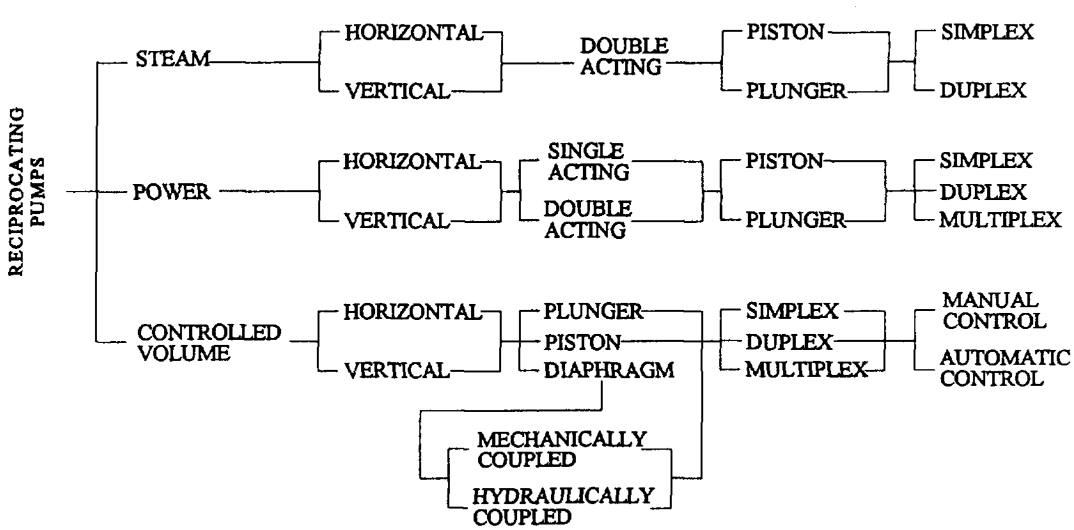



Figure 503-4-1. Types of Reciprocating Pumps.

## 503-4.3 RECIPROCATING PUMP CHARACTERISTICS.

503-4.3.1 Reciprocating pumps are positive displacement units that discharge a definite quantity of liquid during piston or plunger movement through the stroke distance. However, not all the liquid may reach the discharge pipe; leaks or bypass arrangements may prevent this. Without these leaks or bypass arrangements, the volume of liquid displaced during one stroke of the piston or plunger equals the product of the piston or plunger area and the stroke length.

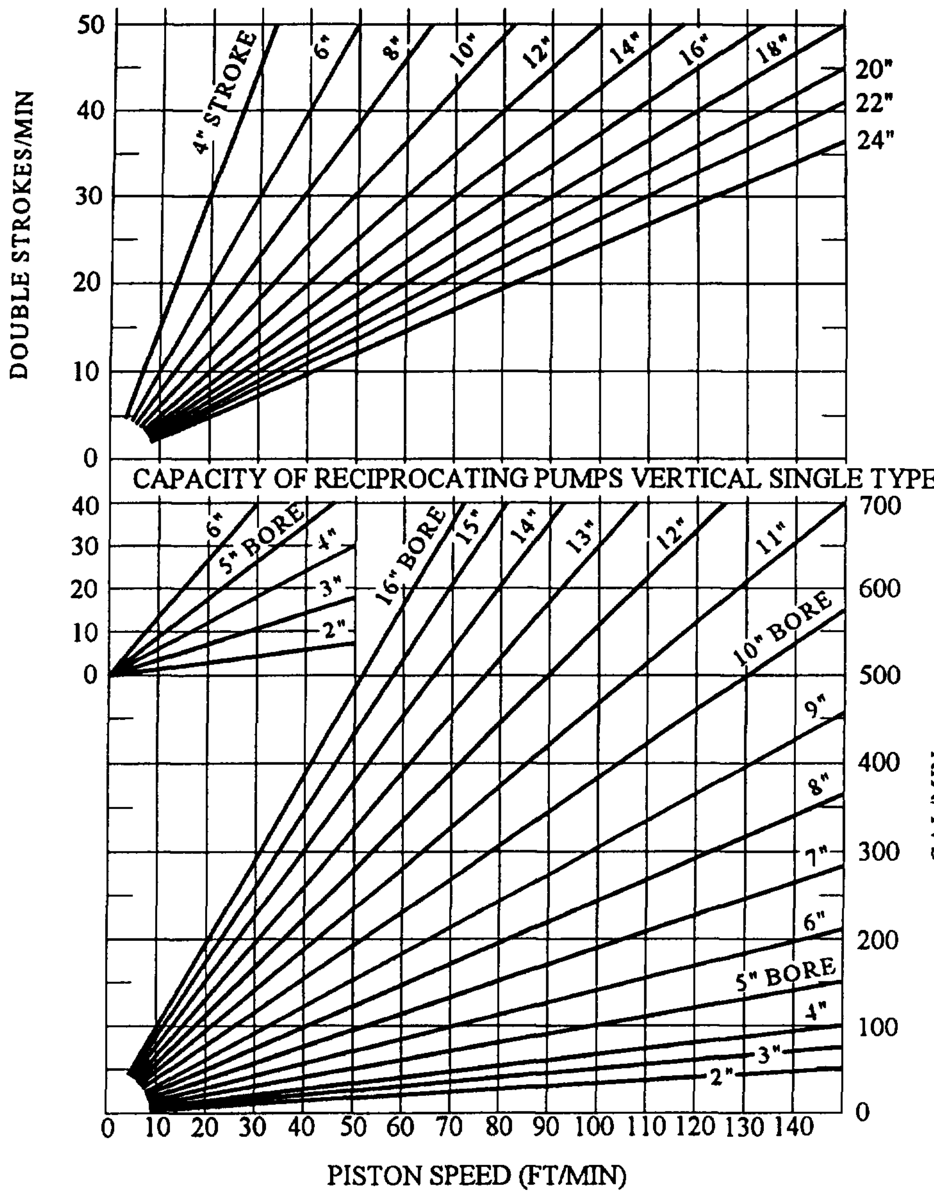



Figure 503-4-2. Reciprocating Pump Capacity Chart.

503-4.3.2 When the piston or plunger has reached it maximum travel in one direction, a method of reversing the piston or plunger direction is required. In the direct-acting steam pump, the piston or plunger direction is reversed by the steam valves and valve gear (actuating mechanism); in power pumps, piston or plunger direction is reversed by the crank and connecting rods. In controlled volume pumps, various mechanisms such as eccentric cams or cranks are used to translate rotary motion into reciprocating motion. Pumping capacity of a steam pump handling liquids is often provided by charts such as the one shown in figure 503-4-2.

## 503-4.4 RECIPROCATING PUMP OPERATION.

503-4.4.1 General Guidance. Refer to the Engineering Operation Sequence System (EOSS) or the equipment technical manual for reciprocating pump preoperation checks, starting, stopping, and securing procedures.

503-4.4.2 Troubleshooting. Troubleshooting reciprocating pumps will require a working knowledge of the pump and the system it serves. Detailed instructions are available in the applicable equipment technical manual and Ships Information Books (SIBs). Also available at the deckplate level are the Engineering Operating Casualty Control (EOCC) procedures and Engineering Operating Sequencing System (EOSS).

## 503-4.5 MAINTENANCE.

503-4.5.1 Maintenance of reciprocating pumps will require a working knowledge of the pump and its system. For detailed instructions, consult the appropriate technical manuals.

## 503-4.6 REPAIRS.

503-4.6.1 Drawings and Data. When a pump is to be repaired or examined, assembly and detail drawings along with available dimensional data must be at hand. After overhaul, important dimensions frequently become altered (for example, the width of and distance between steam ports, the length of rods and steam valves, and the diameter of pistons). These changed dimensions may cause poor operation that will continue, despite other major repairs, until the real cause of the problems is recognized and corrected.

503-4.6.2 Measurements. Whenever reciprocating pumps are opened for repairs, measurements of the cylinders and valve chest on the fore, aft, and athwartship diameters at the top, middle, and bottom shall be taken with a micrometer. The results of these measurements shall be recorded on OPNAV Form 4790-2K and OPNAV Supplement Form 4790-2L, with an accompanying diagrammatic sketch showing measurements obtained and relevant data.

503-4.6.3 Specific Repairs. Specific problems and remedial action that can be taken by ship’s force are described in pump technical manuals.

# SECTION 5

# MECHANICAL SEALS AND PACKED STUFFING BOXES

## 503-5.1 GENERAL DESCRIPTION.

503-5.1.1 In order to prevent excessive leakage at the point where the shaft penetrates the pump, casing seals are required. This leakage may take the form of the loss of some of the fluid being moved by the pump by its exiting the casing, or in the case of a pump operating in a suction lift condition where the casing pressure is less than atmospheric, of air entering the casing. There are generally two types of seals which are used to prevent this leakage: packing or mechanical seals. Older design pumps may use either packing or mechanical seals depending on the specific pump design; however, more recently designed pumps, with the exception of reciprocating pumps, require the use of mechanical seals as the primary sealing method with the use of packing as an emergency back-up.

## 503-5.2 TERMINOLOGY.

503-5.2.1 The following terminology applies to mechanical seals and packed stuffing boxes:

a. Backup Stuffing Box A recessed portion of the gland and cover of a mechanical seal subassembly designed to accommodate two or more rings of packing.

b. Bellows That portion of a mechanical seal which is a flexible secondary sealing assembly which joins the primary sealing face to the shaft.

c. Gland A follower which compresses packing in a stuffing box.

d. Seal Gland Retains the stationary element of a mechanical seal.

e. Gland Plate A pressure-containing housing that is attached to the pump casing and holds the stationary part of the seal.

f. Lantern Ring An annular piece used to establish a liquid seal around the shaft and to lubricate the stuffing box packing. Lantern rings are used in pumps which operate with a suction lift condition to prevent air leakage into the stuffing box.

g. Mating Ring A precision lapped seal face normally mounted in a gland plate.

h. Mechanical Seal A mechanical device located in the pump stuffing box, consisting of a stationary element and a rotating element, each with a smooth, flat sealing face that prevents the flow of liquid or gas into or out of the pump casing.

i. Non - Pusher Seal That portion of a mechanical seal's secondary seal which maintains the seal by accounting for any axial motion of the primary seal by the expansion or contraction of a bellows type arrangement. These seals are generally elastomeric bellows, corrugated metal bellows, or welded metal bellows.

j. Packing A pliable lubricated material used to seal around that portion of the shaft located in the stuffing box.

k. Primary Seal That seal interface of a mechanical seal which is oriented perpendicular to the shaft, consisting of the primary seal ring and the mating ring.

l. Primary Seal Ring A precision lapped seal face which is held in the seal assembly. On some designs it is rotating while on others it is stationary.

m. Pusher Seal  That portion of a mechanical seal's secondary seal which is dynamic in nature and moves axially on the shaft to account for any axial movement of the primary seal due to sealing face irregularities. These seals are generally O-rings, Anti-X-rings, chevrons rings, etc.

n. Semi - Split Mechanical Seal  A mechanical seal where only the mating ring, primary ring and elastomers are split in half.

o. Full- Split Mechanical Seal  A mechanical seal where all components of the seal are split in half.

p. Auxiliary Packing  Usually consisting of two rings of preformed packing for emergency use. If mechanical seal leakage becomes uncontrollable and excessive, the packing is installed until the seal can be repaired or replaced.

q. Auxiliary Packing Gland  Used in conjunction with the auxiliary packing, and is defined as a follower which compresses the auxiliary packing.

r. Seal Chamber  See Stuffing Box.

s. Secondary Seal  That portion of the mechanical seal which prevents the escape of the pumped fluid or ingress of air at the points where the mechanical seal is connected to the shaft and the casing.

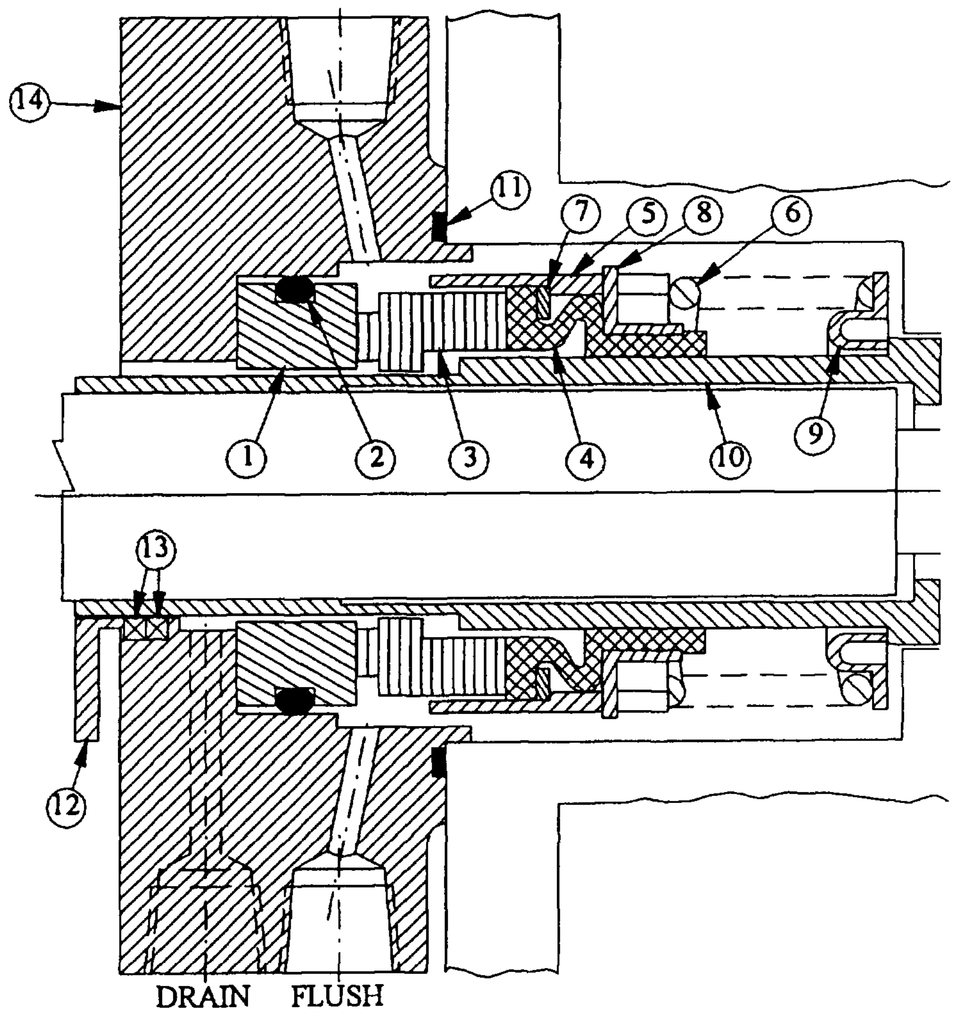
t. Shaft Sleeve  A cylindrical piece fitted over the shaft to protect the shaft through the stuffing box and which may also serve to locate the impeller on the shaft.

u. Stuffing Box  A portion of the casing through which the shaft extends and in which packing and a gland or a mechanical seal is placed to prevent leakage.

v. Throat Bushing  A bushing which is mounted at the interior of the stuffing box which restricts or limits the flow into or out of the seal cavity.

## 503-5.3 MECHANICAL SEALS.

503-5.3.1 Basic Construction.  A mechanical seal consists of a stationary element fixed within the pump casing, cover, or gland and a rotating element connected to the pump shaft or shaft sleeve. Each element includes a seal ring whose face has been highly lapped. The materials of the rotating and stationary seal faces are selected for a combination of low friction and resistance to corrosion by the liquid being sealed. The mechanical seal is designed to operate with the seal faces in close contact with a very thin film of the liquid between them to reduce friction and wear. One of the elements, either the rotating or the stationary element, of the mechanical seal is provided with axial flexibility, either in the form of a spring assembly and sliding elastomer seal (pusher seal) or by use of a bellows (non-pusher seal). This allows the mechanical seal to function properly, maintaining the very thin film of fluid between the seal faces, while accommodating the normal axial movement between the pump shaft and the pump casing. The spring assembly is preloaded to ensure that the mechanical seal remains closed while the pump is depressurized. When the pump is operating the seal faces are kept in contact by forces and spring pressure. Figure 503-5-1 illustrates a typical mechanical seal. The design of mechanical seals currently in service represent a large cross section of configurations and materials. The majority of pumps in service use a single mechanical seal to prevent to loss of pumped fluid exiting the casing or to prevent the intrusion of air from outside of the casing when the pump is under a suction lift condition. However, there are some applications of mechanical seals which require double or tandem seals or special sealing arrangements. These are necessary in those pumps handling hazardous fluid systems (e.g. fuel oil, acids, chemicals, and sewage) due to the higher safety and reliability requirements. Since the application of mechanical seals in Navy pumps, there has been an effort to standardize the seals. All mechanical seals used in Navy new-design pumps are required to meet the criteria of ASTM F1511-96, unless otherwise approved by NAVSEA. Special requirements for sewage pumps are contained in ASTM F1511-96 and paragraph 503-5.3.8.3. Backfit of mechanical seals and conversion of mechanical seals from non-split to split requires approval by NAVSEA for pumps in propulsion plants of nuclear-powered ships and by NSWCCD-SSES 9232 for all other pumps.




| NO. | PART | NO. | PART | NO. | PART |
|---|---|---|---|---|---|
| 1 | MATING RING | 7 | DISC | 13 | PACKING |
| 2 | 0-RING | 8 | DRIVE BAND | 14 | GLAND |
| 3 | PRIMARY RING | 9 | SPRING HOLDER | | |
| 4 | BELLOWS | 10 | SLEEVE | | |
| 5 | RETAINER | 11 | 0-RING | | |
| 6 | SPRING | 12 | AUX. GLAND | | |

Figure 503-5-1. Typical Non-pusher Mechanical Seal.

503-5.3.2 Seal Operation. Successful operation of a mechanical seal depends on proper installation, an adequate supply of clean flushing liquid, and proper venting of the stuffing box. Mechanical seals should never be operated dry, without sealing liquid in the pump stuffing box. The seal depends on the cooling provided by the liquid in the stuffing box and the lubricating provided by the thin film of liquid between the seal faces for proper operation. Unlike packing, mechanical seals do not require periodic adjustment during operation and should operate with negligible leakage without attention.

503-5.3.3 Mechanical Seals vs. Packing. Mechanical seals are preferred over packing in most cases because they minimize stuffing box leakage. They are not used for centrifugal pumps which may run dry or cannot provide an adequate supply of flushing liquid. Mechanical seals for Navy applications are being standardized through the development of a Navy appendix to ASTM Specification F1511-96. When replacing a mechanical seal, only these mechanical seal identified on the pump APL should be used. Ceramic sealing faces are not acceptable in pump mechanical seals. All replacement of ceramic seal face materials should also be done in accordance with the pump APL. If the APL has not been updated to include approved materials, contact NAVSEA, or NSWCCD-SSES 9232. Typically, a backup stuffing box which accommodates two or more rings of packing for use in the event of a seal failure is provided on surface ships.

503-5.3.4 Pressure Breakdown Devices. Seawater pumps for submarines are provided with mechanical seals and pressure breakdown devices. The pressure breakdown device limits the leakage to an acceptable level if the mechanical seal fails under the maximum suction pressure which the pump is designed. Seawater pumps have a backup stuffing box designed with a minimum of two packing rings for use if the mechanical seal fails. The packing rings may be inserted without removing the mechanical seal.

503-5.3.5 Cyclone Separators. Fire pumps and all other centrifugal seawater pumps with a total head of 30 psi or more are usually provided with cyclone separators. Cyclone separator fittings and the pump casing are the straight-thread type with an O-ring seal. Cyclone separator tubing and fitting material for seawater pumps except Navy Standard Titanium fire pumps is copper-nickel (70-30) in accordance with MIL-T-16420. Cyclone separator, tubing, and fitting material for Navy Standard Titanium fire pumps is titanium.

503-5.3.6 Design and Materials. Refer to ASTM 1511-96 for all mechanical seal design specifications and materials.

503-5.3.7 Axial Positioning.

503-5.3.7.1 The axial positioning of the shaft seal during installation is a critical factor in achieving reliable seal operation. Mechanical seals are axially positioned on the pump shaft by positive means, such as a step or shoulder on the shaft, or by a stub or step sleeve that is positively located on the pump shaft. Seals are not axially positioned by set screws or by sleeves held in place by set screws: set screws do not provide a secure connection to the shaft, and upset metal on the pump shaft can damage O-rings installed over the pump shaft. With the seal installed in the correct position, the spring assembly provides the proper axial preload to the seal faces and sufficient axial travel so that the normal end-play, seal-face wear, and thermal growth of the pump shaft can be accommodated without affecting the thin film of liquid between the seal faces. Incorrect axial positioning of a mechanical seal can force the seal faces to separate, causing excessive leakage, or force the seal faces into hard contact, causing seal-face deterioration and overheating.

503-5.3.7.2 The procedure for setting the axial position of a mechanical seal is specific to the application, and the procedure given in the technical manual should be used. In most cases the axial position is set by a step or shoulder on the shaft sleeve; however, in some special applications, such as submarine seawater pumps, the seal axial location may be set using shims to accommodate variations in the pump dimension.

503-5.3.7.3 When replacing seals or other pump parts affecting axial seal location, such as shaft sleeves, do not reuse shims without verifying that they produce the proper seal location.

503-5.3.8 Leakage and Replacement Criteria. Leakage and replacement criteria for newly installed mechanical seals and in-service mechanical seals are as follows:

503-5.3.8.1 Non-Flammable Liquids. When non-flammable liquids are being pumped, mechanical seals shall be replaced in accordance with the following guidelines:

503-5.3.8.1.1 The following Installation criteria applies:

a. New Seal Installation:
   1) After initial 30 minute run-in period no visible seal leakage is expected for new seal installations. If the seal leaks more than 5 drops per minute a decision should be made locally on whether a second replacement is warranted.
b. In-Service Seals:
   1) When seal is removed for any reason.
   2) When seal leakage rate approaches a steady steam (equivalent to 60 drops/minute).
   3) When seal leakage results in liquid being sprayed on surrounding equipment and spaces.
   4) When seal leakage causes a safety hazard or a maintenance burden.

503-5.3.8.1.2 The following seal replacement criteria applies to non-flammable, combustible liquids such as lube oil:

a. New Seal Installation:
   1. After initial 30 minute run-in period no visible seal leakage is expected for new seal installations. The seal shall be replaced when leakage exceeds 5 drops per minute.
b. In-Service Seals:
   1) When seal is removed for any reason.
   2) When seal leakage rate results in constant puddling of liquid on the pump casing or foundation.
   3) When seal leakage results in liquid being sprayed on surrounding equipment and spaces.
   4) When seal leakage causes a safety hazard or a maintenance burden.

503-5.3.8.2 Flammable Liquids. The following applies to flammable liquid pump services (that is, fuel oil, JP-5, gasoline, and so forth):

a. Leakage Test:
   1. Wipe pump tell-tale hole or seal housing dry of any dampness.
   2. Observe and collect leakage for a 30-minute period.
b. Replacement Criterion:
   1. Zero measurable (dripping) leakage is permitted over the 30-minute period for satisfactory seal performance (slight dampness at the tell-tale hole or seal housing is acceptable).

**NOTE**

**If emergency backup packing is installed, the leakage and replacement criteria in paragraph 503-5.4.3 for packed stuffing boxes apply.**

503-5.3.8.3 Sewage Pump Leakage. Double seals lubricated with oil from a dead-ended reservoir located within the pump body act as a buffer fluid system to contain the effluent

a. Leakage Test:

   1. Visually inspect mechanical seal for oil/sewage leakage as required by EOSS procedures, technical manual requirements, or routine operating procedures.

b. Replacement Criteria:

   1. Zero leakage is acceptable. If oil leakage is detected from the seal the pump must be secured, isolated and the seal replaced.
   2. Contamination of the oil in the reservoir with sewage does not constitute a seal failure. The oil in the reservoir should be replaced and periodically checked as required by applicable PMS. Often, oil may take on a milky appearance form entrained air. Also, leakage of a few drops of oil outside the reservoir does not constitute a failure.

503-5.3.9 Installation of Mechanical Seals. Procedures for installation of mechanical seals are specific to the seal manufacturer, pump design, and the application. The application-specific installation procedures should be used. The instructions provided here provide general guidance.

503-5.3.9.1 Cleanliness. The reliability of a mechanical seal depends on it being free of debris that could interfere with the proper movement of internal parts or the proper mating of tightly-toleranced parts.

a. Remove any loose matter from the pump shaft, stuffing box, gland plate, and other stationary parts. Wipe these parts clean using clean, lint-free cloths.

b. Use extreme care when handling the seal faces. The lapped surfaces must be kept clean and free from scratches. Keep the seal faces in their protective wrapping or covered with a clean, dry, lint-free cloth until installed.

c. If the seal faces become soiled, they may be cleaned with water (or another solvent, if approved by the seal manufacturer in the instructions specific to the seal) and a clean, lint-free rag. Flush away any debris that might scratch the face before wiping the seal face clean. Scratched seal faces should generally be replaced, not relapped. However, if the scratch is minor and a replacement mechanical seal assembly is not available, the mechanical seal face may be relapped. Lapping instructions with acceptance criteria specific to the seal, the necessary lapping equipment, and qualified personnel are required if a seal face is to be relapped. Otherwise, replace the part. If lapping cannot be done on site, it may be possible to return the part to the seal manufacturer for relapping and later use.

d. Ensure that the spring assembly is clean and free of debris that could impede its free axial travel.

503-5.3.9.2 Burrs and Sharp Edges. Carefully examine the pump shaft, shaft sleeve, and other parts that have tight clearances or that come in contact with elastomer parts during installation or operation. Remove any burrs or sharp edges that could interfere with the tight clearances or damage the elastomer parts.

503-5.3.9.3 O-Ring Fits. Check O-rings installed in grooves for a proper, snug fit. For O-rings which must be stretched over another part during installation, such as the pump shaft, ensure that sufficient time is provided for the 0-ring to contract to a snug fit before installing the mating part.

503-5.3.9.4 Lubrication of Secondary Seal. On pusher type seals, the elastomer secondary seal (e.g. O-ring or U-cup), which slides on the pump shaft or shaft sleeve in the assembly providing axial flexibility to the mechanical seal, should be lubricated. The surface against which the secondary seal slides should also be lightly lubricated in the narrow band where the secondary seal will slide. The lubricant used should only be that specified by the manufacturer for the specific application. Some elastomer materials are adversely affected by inappropriate lubricants. For example, EPR (ethylene-propylene rubber) material swells when lubricated with petroleum-based substances.

**CAUTION**

**Do not use petroleum jelly, TFE (tetrafluroethylene), or silicon grease on an elastomer seal. EPR must not be lubricated with any petroleum-based substance.**

503-5.3.9.5 Lubrication of Seal Faces. In general, the lapped seal faces do not require lubrication at assembly; they are lubricated by the liquid being pumped. Liquid lubricant should not be applied to the seal faces unless it is specifically called for in the manufacturer's instructions. Greases and other solids bearing lubricants should never be used. They will ruin the seal.

503-5.3.9.6 Seating of Seal Faces. The seal face, in the element of the mechanical seal not containing the spring assembly, must be squarely seated without cocking to ensure proper seal performance and prevent secondary seal damage. Otherwise, the sliding secondary seal in the other element will be subject to a once-per-revolution sliding cycle. This can cause fretting of the elastomer seal and the seal sleeve/pump shaft. When installing the seal face, ensure that the seating surface is free of debris, burrs, and other upsets that could prevent the seal face from seating properly. If the seal face must be pressed into a recess with an O-ring sealing the outside diameter, ensure that the seal face is fully pressed against the seat during installation.

503-5.3.9.7 Alignment. Proper alignment is important to reliable operation of a mechanical seal. When installing mechanical seals, ensure that the pump shaft is straight and centered within the stuffing box within prescribed tolerances. Ensure that the close-tolerance fit, which locates the gland plate in the stuffing box, is free of debris and burrs and that the gland plate is properly seated when installed. Before coupling the pump to the drive motor, verify that the axial end-play in the motor is within tolerances. Ensure that the angular and radial alignment of the pump and its coupling assembly to the drive motor are within prescribed limits when the pump is reassembled. In particular, ensure that tapered couplings are properly assembled and that coupling faces are free of burrs and other debris.

503-5.3.9.8 Gland Plate Torquing. Ensure that the gland-plate bolting is properly torqued and not overtightened. Excessive tightening of the gland-plate bolting can lead to distortion of the gland plant and stationary seal face.

503-5.3.9.9 Filling and Venting. The pump and stuffing box should be filled with the pumped fluid and fully vented to remove all of the air from the pump. If some air remains in the pump, the seal faces may not be properly lubricated, producing overheating of the seal face, wear, and failure.

## 503-5.4 PACKED STUFFING BOXES.

503-5.4.1 In a packed stuffing box, the seal between the moving shaft and the stationary cavity of the stuffing box is accomplished by forcing rings of packing between the two surfaces. A gland holds the packing in the stuffing box and is also used to control the amount of leakage along the shaft by tightening or loosening the gland nuts (see Figure 503-5-2). When a pump is designed to work on a suction lift, stuffing boxes are fitted with lantern rings or seal cages (usually located near the axial center of the stuffing box) to prevent air leakage into the stuffing box and to lubricate the packing. Sealing liquid is injected into the seal cage.

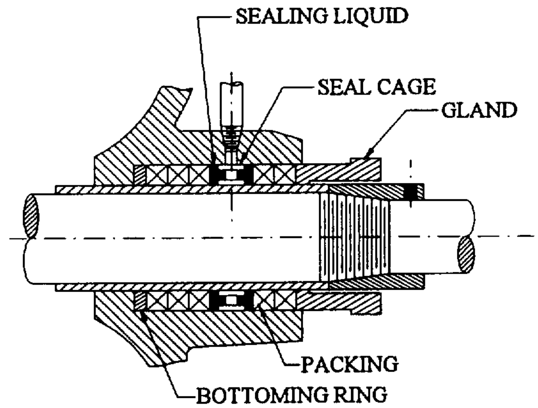



Figure 503-5-2. Packed Stuffing Box.

503-5.4.2 When clean, cold water is pumped, the seal water line is usually connected to the pump casing near the discharge. To avoid excessive pressure in multi-stage pumps, the connection is to the first stage. An independent external liquid sealing supply is desirable under any of these conditions:

a. When suction lift is high.

b. When discharge pressure is low.

c. When the pump momentarily runs dry such as with a condensate pump.

d. When pumping hot water.

e. When pumping brine.

f. When pumping water with high chemical content.

503-5.4.3 Leakage and Replacement Criteria. Leakage and replacement criteria for packed stuffing boxes and emergency backup stuffing boxes follow.

**WARNING**

**Do not air-test a seal using the candle-flame method on a pump handling oil, gasoline, or other flammable liquid.**

503-5.4.3.1 Non-Flammable Liquids. Take particular care to ensure that seal water piping is open and clean, allowing adequate flow of seal water to lubricate the packing. Otherwise, seal or stuffing box packing can wear rapidly from increased friction, causing shaft scoring. This in turn increases the clearance between the shaft and the packing, allowing air to be drawn in or fluid to leak out. On pumps which operate with a suction lift, test seal efficiency with a candle flame to see if air is being drawing into the pump. Even with the pump under a positive suction head, allow water to drip from the stuffing box to lubricate and cool the packing. Replace packing when the packing gland has been tightened to the point that it cannot control leakage. Partially replace packing only as a temporary or emergency action.

**CAUTION**

**Do not draw packing glands too tight. Packing glands should be adjusted to allow a small amount of leakage to provide adequate packing lubrication and cooling.**

503-5.4.3.2 Flammable Liquids. Leakage from flammable liquid pumps (for example fuel oil, JP-5, gasoline) should be maintained at 1 to 5 drops per minute. Monitor stuffing boxes carefully when starting the pump. At the first sign of overheating, stop the pump. The stuffing boxes should cool before restarting. When stuffing box leakage must be severely limited, as when pumping a flammable liquid, packing should be changed as frequently as required to maintain 1 to 5 drops per minute leakage and prevent shaft scoring.

503-5.4.4 Installation Procedure for Packing. The packing used in reactor plant pumps shall be in accordance with the pump technical manuals. All other water service pumps shall be packed with non-asbestos packing in accordance with MIL-P-24790 and paragraph 503-5.4.4.1 through paragraph 503-5.4.4.8, or as otherwise approved by NAVSEA except when the pump technical manual proprietary non-asbestos packing. In the latter case, the packing and procedure in the pump technical manual shall be followed. Any problems with installing, using, removing, and replacing non-asbestos packing shall be directed to NAVSEA.

503-5.4.4.1 Remove all old packing, and lantern ring (seal cage); if installed, from the stuffing box. Be careful not to damage the shaft sleeve. Clean the stuffing box and examine the shaft sleeve for scoring. A deeply scored or rough shaft sleeve will result in short packing life and should be replaced.

503-5.4.4.2 If spool packing is used, cut rings making the ends square. Trim the ends so the butt clearances are 1/8 inch to 3/16 inch when the ring is wrapped around the shaft sleeve. Make sure the ends remain parallel. If time permits, soak the packing rings in water for several hours before installing.

503-5.4.4.3 Install the first ring of packing by inserting the butt joint into the stuffing box. Gradually press the rest of the ring with the fingers, starting at the butt joint and working around the circumference of the ring until the ring is evenly positioned in the stuffing box. Press this ring of packing into the stuffing box as far as it will go by installing gland halves and applying force until the gland contacts the stuffing box. Remove gland halves. The ring should fit snugly in the stuffing box without overlapping the ends. Use this technique to install all the packing rings.

503-5.4.4.4 Install the remaining packing rings, (stagger the joints 90 degrees). Where applicable, refer to the technical manual for lantern ring installation. If the last ring does not fit in the box but partially enters the box, do not force the ring into place; remove it. Refer to the individual equipment technical manual for the number of rings to install.

503-5.4.4.5 Install gland halves, bolt together, and install the gland assembly washers and nuts. With the gland true to the shaft, finger-tighten the gland nuts until the gland contacts the top ring of the packing. If enough head is available, apply suction pressure to the pump and allow water to flow freely from the stuffing box for at least 30 minutes. This will provide time for the packing to absorb water, which will provide lubrication during start-up and protection from burnout.

503-5.4.4.6 Start the unit. Run the pump for at least 1/2 hour with the gland leaking freely. Over time, gland leakage may decrease when the pump is rotating without tightening the packing. The procedures in paragraph 503-5.4.4.7 through paragraph 503-5.4.4.13 are for inspection and adjustments after the shaft has been packed.

503-5.4.4.7 Tighten gland nuts in small increments, gradually reducing gland leakage to the amount specified in the technical manual. If none is specified, reduce to a maximum of 32 oz (1 quart)/min. Do not try to make this adjustment all at one time. Wait about 30 minutes between gland nut adjustments. To ensure that the gland does not cock and compresses the packing evenly, tighten nuts equally by counting the turns made. Use this procedure whenever nuts are adjusted. When gland leakage has been significantly reduced, it may be necessary to adjust gland nuts in small fractions of turns (or flats).

503-5.4.4.8 If one ring of packing was left out as allowed in paragraph 503-5.4.4.4, secure the pump and suction pressure. When packing has been compressed sufficiently to permit installation of the last ring, install the additional ring at this time.

503-5.4.4.9 Continue pump operation, gradually reducing the leakage further. Enough gland leakage must be maintained to cool the packing. One symptom of inadequate cooling is steam coming from the stuffing box. When the leakage is 20 to 30 oz/min, gland nut adjustment should not be more than 1 flat of the nut at a time. Below 20 oz/min, adjustment should be less than 1/2 flat of the nut. Allow about 30 minutes between adjustments to stabilize the packing. Sometimes, when small adjustments are made, the gland nuts can be turned without a noticeable change in leakage rate. Do not reduce gland leakage too much, since loosening the gland nuts does not always result in increased gland leakage.

503-5.4.4.10 If leakage decreases too rapidly, slack off gland nuts to increase leakage. If the stuffing box heats up, indicating that the packing is overheating, secure the pump. Allow the stuffing box to cool, loosen the gland nuts about 1/2 turn, and restart the unit. This may restore gland leakage to the minimum required.

503-5.4.4.11 The minimum gland leakage rate obtained will depend on four factors: the speed of the motor, the diameter of the shaft sleeve, the pump internal pressure the packing seals against, and the temperature of the water. As each of these factors increases, the minimum amount of gland leakage that should be obtained to prevent burning the packing increases. For example, a pump with a speed of 3,600 r/min, shaft sleeve diameter of 3-3/4 inches, a packing sealing pressure of 100 lb/in $^2$ g, and a water temperature of 65.6°C (150°F) requires a minimum gland leakage of 8 oz/min (this is a stream about the size of a pencil). On the other hand, another pump pumping water of about the same temperature but with a speed of 1,800 r/min, a shaft sleeve diameter of 1-1/8 inch and a packing sealing pressure of 25 lb/in $^2$ g can obtain a gland leakage rate of 0.1 oz/min. For pumps with variable speeds, suction temperatures, or packing sealing pressures, the minimum gland leakage rate should be adjusted to a rate acceptable for all operating conditions.

503-5.4.4.12 Once the packing has been broken-in, limit the amount of leakage to an amount that will not undesirably affect the equipment or surrounding spaces. Gland leakage, however, should always be less than 32 oz/min.

503-5.4.4.13 Periodically monitor the packing leakage rate. Adjust the gland nut only when gland leakage exceeds acceptable rates. When adjusting the gland leakage rate, do not make large or rapid adjustments of gland nuts. When making adjustments to the stuffing box leak off rate, you may see smoke coming from the stuffing box area. If so, then secure the pump, isolate the suction, and discharge and repack the pump with new packing as required by the technical manual or MRC.

# SECTION 6

# ALIGNMENT

## 503-6.1 WHAT IS ALIGNMENT?

Alignment is the process of adjusting two or more machines that are coupled together so their shaft centerlines form one continuous straight line. The term shaft-to-shaft alignment should be used instead of coupling alignment. Even though the couplings are aligned properly, the shaft centerlines may not form one continuous straight line. This depends on whether the couplings were bored straight and true and were machined perfectly about their rim and face. The goal is to align the shaft centerlines, not the couplings. An alignment check is the process of measuring the offset and angularity of one shaft centerline in relation to the other shaft centerline.

## 503-6.2 WHY ALIGN EQUIPMENT?

The pump and driver must be properly aligned at normal operating temperatures to eliminate undue mechanical stresses on the shafts, bearings, coupling, and mechanical seals. It is important that the centerlines of the two shafts form one continuous straight line that does not exceed the maximum tolerances recommended in the ″Indicator Reverse Method of Pump Shaft Alignment Instruction Manual″, NAVSEA S6226-JX-MMA-010. Alignment should be checked to determine whether the shafts are within the specified tolerance of offset and angularity. Alignment checks are required under the following conditions: upon installation of a pump or driver, when the baseplate or foundation is repaired, as part of a Class B overhaul of the unit, if a maintenance action disturbs or removes a rotor, replaces a bearing or seal, modifies or affects any attached piping, and by PMS when specified. Additionally, if the equipment experiences increased vibration or failure of bearings, couplings, or mechanical seals, an alignment check should be performed lAW NAVSEA S6226-JX-MMA-010. Alignment should also be performed when new or overhauled equipment is installed or when the equipment alignment maximum tolerances as specified in NAVSEA S6226-JX-MMA-010, have been exceeded.

## 503-6.3 IMPORTANCE OF PROPER PIPING ALIGNMENT.

Proper suction and discharge piping alignment is also critical to the alignment of the pump and driver. Misalignment of suction and discharge piping connections can cause pump and driver misalignment. Piping flanges and fastener holes must be aligned so that no external force is required to attach the flanges. Chain falls, pry bars, jacks, or wedges shall not be used to position piping. Proper piping alignment is achieved when correct sized flange fasteners fit through the fastener holes without applying external force to the piping. Ideally, flanges should be brought together freely with minimal gap between flange faces. This gap should be equal over the entire periphery of the flange bolt (fastener) circle. Prior to connecting suction and discharge piping, set up dial indicators on the coupling hubs. When connecting the pipe flanges, observe the dial indicators for movement or strain imposed on the equipment by the piping.

Eliminate any strain imposed. Minor piping adjustments can be accomplished by modifying and adjusting the pipe hangers. Major piping adjustments will require cutting and refitting the piping or relocating the hanger. Navy Resilient Mount Handbook NAVSEA 0900-LP-089-5010 should also be reviewed if pumps are equipped with sound and vibration dampening devices. Flexible piping connections will also impact alignment of the pump and motor to each other and to the piping connections.

## 503-6.4 DOWEL PIN PLACEMENT.

It is important to install new tapered dowel pins of the approved material (304 stainless steel) that pass through the equipment mounting feet and into the baseplate. Dowel pins have a snug fit as opposed to the anchor fasteners which are usually in slotted holes. Dowel pins will positively locate the equipment and prevent shifting and misalignment. Dowel pins should be installed as specified by the original equipment manufacturer in the equipment technical manual. If dowel pin locations are not specified in the technical manual the dowel pins should be installed in the front feet (feet closest to the coupling) of each unit. Use the following procedure to install dowel pins:

a. Use tapered dowel pins with 1/4 inch per foot taper and of sufficient length to extend completely through the equipment foot and baseplate.

b. Ream tapered holes (1/4 inch per foot taper) through the equipment foot and the baseplate.

c. Insert dowel pins in holes and secure pins.

**503-6.5 TERMINOLOGY.**

The following terminology applies to alignment:

a. Machine To Be Shimmed (MTBS) When aligning a pump-driver unit, it is usually necessary to adjust only one of the machines. The machine chosen to be moved or adjusted should be the one that has the fewest physical restriction. For example, when performing a pump-motor alignment the motor is usually the machine chosen to be moved due to the restrictions of the piping of the pump. The machine that is chosen to be moved or shimmed shall be referred to as the MTBS.

b. Stationary Machine (SM) The machine that is not moved shall be referred to as the stationary machine. The SM shaft centerline is used as a reference point for all movements as well as the reference to describe any misalignment present between the two shafts.

c. Alignment The process of adjusting a piece of machinery, the MTBS, so the centerline of its shaft is accurately positioned relative to the SM shaft centerline. In most cases acceptable alignment is achieved by adjusting the MTBS in both the vertical and horizontal planes.

**NOTE**

**Alignment of close-coupled pumps is covered in Section 2. CENTRIFUGAL PUMPS**

d. Vertical Misalignment When the MTBS must be shimmed to adjust the machinery position in the vertical place in order to have acceptable alignment.

e. Horizontal Misalignment When the MTBS must be moved to adjust machinery position in the horizontal plane in order to have acceptable alignment.

f. Angularity The angle formed between the MTBS shaft centerline and the SM shaft centerline. This angle is expressed as a slope of the angle in thousandths of an inch per inch. Angularity is determined in both the vertical and horizontal planes and is expressed as vertical angularity and horizontal angularity.

g. Offset The distance between two shafts when measured at the centerline (midpoint) of the coupling. The distance is expressed in thousandths of an inch. Offset is determined in both the vertical and horizontal planes and is expressed as vertical offset and horizontal offset.

h. Front Feet (Foot) The inboard feet, the feet closest to the coupling, of the MTBS.

i. Back Feet (Foot) The outboard feet, the feet farthest from the coupling, of the MTBS

j. Zero (0) The process of setting the dial indicator to 0.

k. Indicator Sag The bending of the dial indicator mounting hardware. This typically occurs on horizontal mounted machinery when the dial indicator is rotated from the 12 o'clock position to the 6 o'clock position. This bending, referred to as sag, occurs due to the force of gravity acting on the weight of the indicator and mounting hardware.

l. Soft Foot  A condition that exists when the bottom of all four feet of the MTBS are not sitting square on the machine base. This can be compared to a chair that has one short leg.

m. Hold-Down Bolts and Nuts  The bolts and nuts that are used to secure each foot of the movable machine to the machine base.

n. Jacking Bolts  Horizontally positioned bolts on the machine base which are located at each foot of the machine and are used to adjust the horizontal position of the machine.

o. Thermal Growth  The amount of change in relative shaft position which occurs when horizontally mounted machinery changes from room temperature to operating temperatures. This change occurs due to the expansion or contraction of the material supporting the shafts.

p. Hot Alignment  The alignment condition of a unit at operating temperature. When hot, the desired relative position of the MTBS will be perfectly in line with the SM shaft; this will result in zero angularity and zero offset.

**NOTE**

**Motor driven units must be run a minimum of four (4) hours to achieve operating temperature. Turbine driven units must be run in a minimum of two (2) hours to achieve operating temperature. Hot alignment readings must be taken within 30 minutes of shutting sown machinery. Actual physical adjustments should not be attempted (in neither the vertical not horizontal plane) until the unit has cooled down to room temperature as thermal contraction will continuously occur during the cooling process. Final results of any adjustments would be unpredictable, particularly in the vertical plane (even if adjustments are only attempted in the horizontal plane).**

q. Cold Alignment  The alignment condition of the unit at room temperature. This is frequently an intentional misalignment condition which will result in a satisfactory hot alignment. The amount of intentional misalignment, equal to the amount of thermal growth, is called the cold alignment setting.

r. Cold Alignment Setting  The desired position of the MTBS shaft, relative to the SM shaft centerline, at room temperature which will result in a satisfactory hot alignment. The cold alignment setting is a specific value which varies for each piece of machinery. The mechanic determines the cold alignment setting for a piece of machinery by obtaining a satisfactory hot alignment, and then measuring the cold alignment. This cold alignment condition will be the cold alignment setting for that unit.

s. Indicator Reverse Method  Radial dial indicator readings (Rim Readings) taken on both the driver and driven shafts for the purpose of calculating the relative positions of the shafts. Sometimes referred to as the Double Dial Method.

t. Double Dial Method  The Indicator Reverse Method is sometimes called the ″Double Dial Method″ since two dial indicators are used; however, because other alignment procedures also use two dial indicators, ″Indicator Reverse″ is the only correct title for the procedure described in this Manual.

## 503-6.6 APPROVED ALIGNMENT METHODS

This section briefly describes shaft alignment methods currently endorsed by the NAVSEA technical authority. The preferred method of pump shaft alignment is the Indicator Reverse Method. This method is described in detail in the ″Instruction Manual for the Indicator Reverse Method of Pump Shaft Alignment″ (S6226-JX-MMA-010). The indicator reverse method can be performed using the machinery alignment kit with dial indicators, a laser alignment system, mathematical calculations using dial indicator readings or a graphical method using dial indicator readings. When interference prevents installation of brackets used to perform the indicator reverse method of shaft alignment, a rim and face or straight edge/feeler gauge method can be substituted. It should be noted that the indicator reverse method must be used if at all possible, when performing pump shaft alignments.

503-6.6.1 INDICATOR REVERSE METHOD An alignment check using the indicator reverse method of pump shaft alignment as detailed in NAVSEA S6226-JX-MMA-010, is the preferred method when determining the alignment condition of coupled machinery. Horizontally mounted, flexibly coupled pumps must have final shaft alignment checks accomplished with the unit at normal operating temperatures to compensate for the affects of thermal growth. This is known as a final hot alignment check. Final hot alignment checks must be performed using procedures detailed in NAVSEA S6226-JX-MMA-010 and comply with its recommended tolerance range. Vertically mounted, flexibly coupled pumps do not require a final hot alignment check, since variables associated with thermal growth in a vertical unit have no affect on unit alignment. Therefore, vertically mounted pump final alignment checks should be performed in the cold condition using procedures detailed in NAVSEA S6226-JX-MMA-010 and comply with its recommended tolerance range. This method utilized mathematical calculations to determine the relative position of the pump shaft with respect to the driver shaft. The procedure can also calculate the exact moves required in both the vertical and horizontal planes to bring the shafts into proper alignment. Maintaining proper alignment using the indicator reverse method will extend the operational life of coupled pumps by reducing or eliminating the excessive forces on the bearings, seals, shafts and coupling. Performing shaft alignment checks in accordance with NAVSEA S6226-JX-MMA-010 and correcting those units found to have alignment conditions outside the recommended tolerance range will extend the operational reliability of the pump. Additionally, the overall life cycle costs to maintain the units will be significantly reduced.

503-6.6.1.1 HOT ALIGNMENT CHECK. When a machine goes from non-operating to operating condition, a temperature rise occurs, causing the metal of the support structure to expand. The expansion of metal components causes the shaft to rise. This can significantly influence the equipment alignment. Therefore, when aligning a pump, extreme care must be taken to account for thermal growth. When equipment technical manuals do not specify a thermal growth value, align equipment in the cold condition, operate it for a sufficient period of time to allow the bearing and support structure temperatures to stabilize (about 4 hours for motor driven pumps and 2 hours for steam turbine drives), then perform a hot alignment check. The accuracy of a hot alignment check depends on how soon after shutdown the dial indicator readings are taken. The use of shaft mounted brackets is recommended. They should be assembled to the fullest extent possible prior to equipment shutdown so that readings can be taken before the equipment cools. Readings should be taken within 30 minutes of shutdown. The indicator reverse method should be used for the hot alignment check because the flexible coupling does not have to be disassembled to obtain the required readings. Refer to the alignment instruction manual, NAVSEA S6225-JX-MMA-010, for detailed procedures on performing hot alignment checks and making appropriate adjustments. If the alignment readings are entered into the alignment computer, it will give the corrective moves required to align the equipment for normal operating temperature. If the hand held alignment computer is not available, calculations may be performed either with a pocket calculator or manually as detailed in the alignment technical manual.

**NOTE**

**The indicator reverse method for pump alignment as described in S6226-JX-MMA-010 is the preferred procedure for Navy Pumps. In some cases, the**

**pump arrangement may not allow for proper placement of the dial indicators and shaft rotation. Therefore, the following alternative methods are presented for use in those instances.**

503-6.6.2 RIM AND FACE METHOD. The rim and face method can be done either of the following ways for horizontal pump units:

a. One rim reading and one face reading on the stationary machine with the dial indicator mounting brackets attached to the MTBS. Both shafts must be rotated simultaneously to check proper shaft alignment.

b. One rim reading and one face reading taken on the MTBS with the dial indicator mounting brackets attached to the stationary machine. Both shafts must be rotated simultaneously to check proper shaft alignment.

For vertical pump unit alignment, it is necessary to use both rim and face readings, though not simultaneously. First, angular misalignment must be corrected by shimming the driver. Second, the offset misalignment must be corrected by moving the driver flange on the baseplate.

503-6.6.3 STRAIGHT EDGE AND THICKNESS GAUGE METHOD. The straight edge and thickness gauge method is an approximate method of checking alignment. It should be used to get the equipment close on an initial setup or as a last resort for alignment. This method is not an accurate alignment. It should be used only when precision instruments are not available. The following procedure should be used for the straight edge and thickness gauge method to rough align equipment being installed:

a. Ensure that machine base and bottom of machine feet are clean and free of rust and burrs. Clean if necessary.

b. Use only clean shims of the approved material (304 stainless steel) and remove any burrs. Do not use laminated shims or brass shims.

c. Check for proper coupling end gap in accordance with the applicable technical manual.

d. Check angular alignment by inserting gauge between the coupling faces at 90-degree intervals. If any of the four readings differ by 0.002 inch or more, adjust the angular alignment by shimming or moving the motor. Recheck angular alignment after each movement.

e. Check offset alignment by placing a straight edge across both coupling rims at 90-degree intervals. Adjust or shim the motor until the straight edge rests evenly on the coupling rim at all positions.

f. After all adjustments have been completed, recheck both angular and offset alignment.

g. Connect the suction and discharge piping. Recheck angular and offset alignment.

# APPENDIX A

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