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NAVAL SHIPS TECHNICAL MANUAL

CHAPTER 9470

PUMPS

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NAVAL SHIP SYSTEMS COMMAND, WASHINGTON, D. C. 20360

February 1969 Edition

CHAPTER 9470 – PUMPS

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SECTION I. RECIPROCATING PUMPS

Part 1. General

9470.1 BASIC PRINCIPLES

1. The pumping action in any reciprocating pump is dependent upon the positive displacement or the fluid pumped by a piston or plunger. The capacity of the pump is, therefore, determined by the area of the piston and its rate of travel. In order to obtain a practical machine, some method of reversing the direction of the pistons is required. In the direct-acting steam pump, this is accomplished by the steam valves and valve gear; in power pumps, this is accomplished by use of crank and connecting rods.

2. The ability of the pump to produce pressure is dependent upon the ratio of total steam force (steam pressure per unit area x area of steam piston) to total liquid force (pump head x area of liquid piston). In order that pumping may occur, it is necessary that the steam force exceed the liquid force by an amount which slightly exceeds the various mechanical and hydraulic losses encountered. The basic principles for steam pump operation are shown in figure 9470-1.

9470.2 TYPES OF PUMPS

1. Direct-acting reciprocating pumps are classed as follows:

a. Horizontal or vertical.

b. Single or duplex. A single pump has one liquid piston or its equivalent single or double-acting plunger; a duplex pump has two liquid pistons or their equivalent single or double-acting plungers.

c. Single or double-acting. A single-acting unit pumps on one direction of piston travel only whereas double-acting units pump on both strokes. Direct-acting steam pumps are usually double acting.

2. Direct-acting steam pumps are conventionally described by stating the steam cylinder diameter, the liquid cylinder diameter, the length of stroke, horizontal or vertical (H or V), single or duplex (S or D), and single or double-acting (SA or DA). Thus a pump identified as 11 x 8 x 18 VSDA has an 11-inch steam cylinder, 8-inch water cylinder, 18-inch stroke, and is a vertical single double-acting pump.

9470.3 USAGE IN NAVAL SERVICE

1. Direct-acting steam pumps in naval service are installed on combatant ships for the following services:

a. Emergency feed.
b. Fire and bilge.
c. Fuel oil tank stripping and bilge.

2. On auxiliary ships a great number of applications are still served by steam reciprocating pumps, including:

a. Auxiliary feed.
b. Standby fuel oil service.
c. Fuel oil transfer.
d. Auxiliary circulating and condensate.
e. Fire and bilge.
f. Ballast.
g. High pressure evaporator.
h. Lubricating oil transfer.
i. Cargo stripping.
j. General service.

3. Direct-acting steam reciprocating pumps are not obsolete. If the steam conditions are not too severe in pressure, temperature, or superheat, they have many features of simplicity, reliability, and economy of operation and maintenance that still warrant serious consideration for many services.

9470.4 STEAM VALVE GEAR

1. Steam valve gear for direct-acting reciprocating pumps can be classified by the following basic types:

a. The direct-acting flat face slide valve. This type is suitable for low steam pressures, in the order of 150 to 200 psi and below.

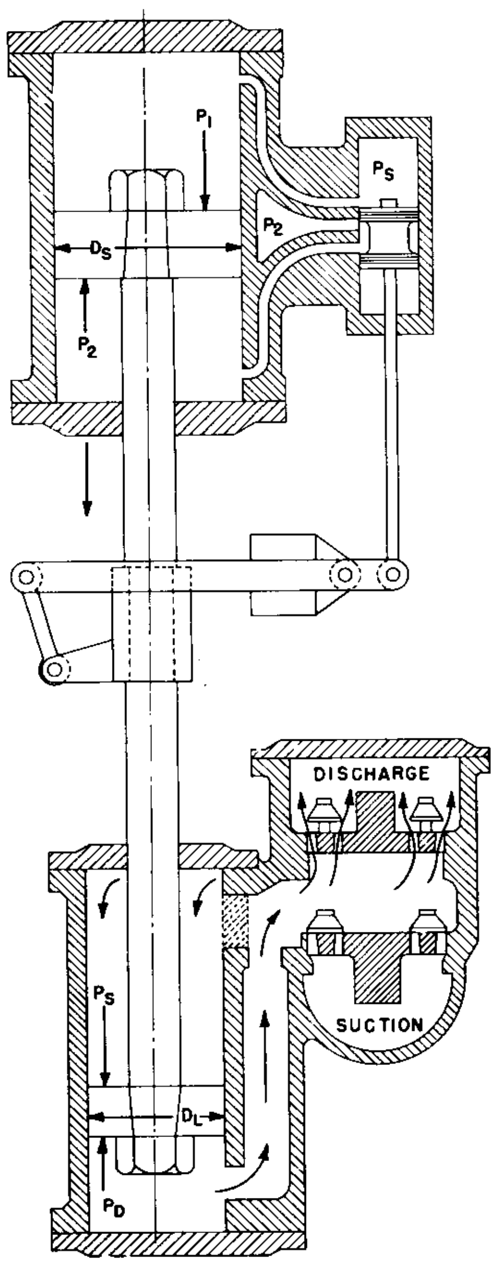
b. A steam thrown piston main valve actuated by a flat faced pilot or auxiliary valve. This type is suitable for moderate steam pressures possibly up to 500 psi and is frequently found on naval auxiliary ships converted from merchant ships.

c. A steam thrown piston main valve actuated by a piston type pilot valve. This design is capable of operation at the higher steam pressures and temperatures encountered on naval vessels. The piston main and pilot valves eliminate unbalanced loads and minimize steam leakage and wear.

2. Valve gears of various manufactures differ to the extent that a brief description of several of the designs is desirable. The descripitions following cover a majority of the pumps installed in naval service.

9470.5 DAVIDSON VALVE GEAR

1. The Davidson balanced piston-type valve gear is illustrated in figure 9470-2. Pilot valve "A" is direct-actuated



L = LENGTH OF STROKE IN INCHES.
D_S = STEAM CYLINDER DIAMETER, INCHES.
D_L = LIQUID CYLINDER DIAMETER, INCHES.
P_1 = STEAM PRESSURE P.S.I.G.
P_2 = EXHAUST OR BACK PRESSURE P.S.I.G.
P_D = PUMP DISCHARGE PRESSURE P.S.I.G.
P_S = PUMP SUCTION PRESSURE P.S.I.G. (NEGATIVE IF SUCTION LIFT EXISTS)
N = NUMBER OF DOUBLE STROKES PER MINUTE.
S = PISTON SPEED IN FEET PER MINUTE. $S = \frac{LN}{6}$
E = PUMP EFFICIENCY.

PRESSURE EQUATION

$$(P_1 - P_2)\, D_S^2 \times E = (P_D - P_S)\, D_L^2$$

CAPACITY EQUATIONS

FOR SINGLE CYLINDER, DOUBLE ACTING PUMP:

$$Q\,(\text{G.P.M.}) = .0064\, D^2 LN$$
$$= .0385\, D^2 S$$

THESE FORMULAS INCLUDE AN AVERAGE CORRECTION FOR VOLUMETRIC EFFICIENCY AND PISTON ROD DIAMETER.

Figure 9470-1. Schematic Pressure-Volume Diagram

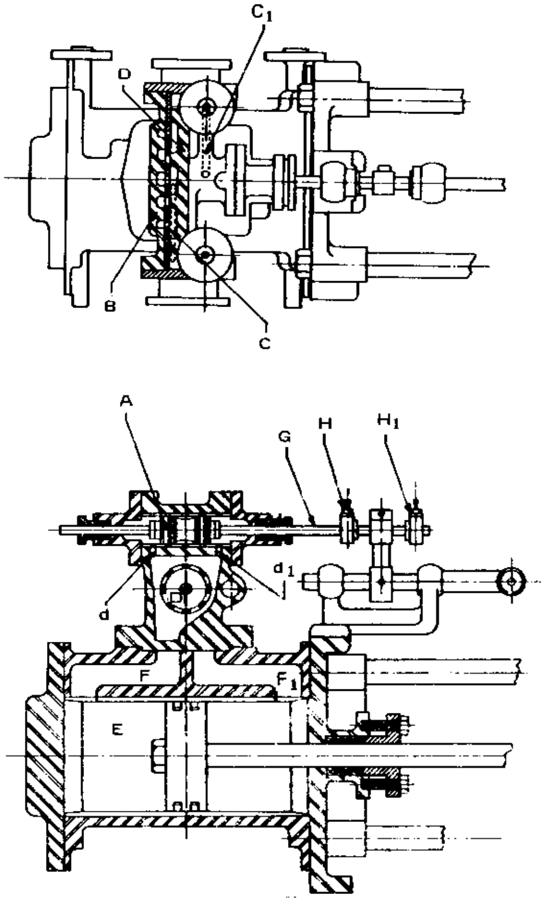


Figure 9470-2

by drive rod and valve rod "G" from the crosshead. The valve rod passes through clearance hole in center of pilot valve. Ends of pilot valve chest are under exhaust pressure only, the pilot valve having drilled balance ports end to end.

2. Steam pressure is maintained in central chamber of pilot valve "A". When valve is actuated by valve rod, inside edge uncovers the port "C" leading steam pressure to one end of main valve chest, at the same time the outside edge uncovers part "C_1" leading exhaust to opposite end of main valve chest, causing valve "D" to throw. Main valve is thrown by inlet steam, not exhaust pressure.

3. Steam is then admitted to steam cylinder "E" through port "F" and port "F_1" is opened to exhaust.

4. When piston has completed the stroke, pilot valve is actuated in opposite direction and reverse cycle takes place; port "C" is opened to exhaust, port "C_1" is opened to steam throwing main valve to opposite end of chest causing steam piston to reverse.

5. The ports "d" and "d_1" of ends of pilot chest bores are fitted with removable liners recessed on outside lead directly into main chest exhaust port and balance the pressure on pilot valve. Both main and pilot chest diameter with drilled ports around the circumference, completely balancing both piston valves. Since the main valve is horizontal, it does not have to work against gravity.

6. The length of stroke may be adjusted by means of tappets "H" and "H_1" on the threaded portion of pilot valve rod "G".

9470.6 WARREN ECLIPSE VALVE

This valve is shown in figure 9470-3 and its operation is as follows:

1. The operating lever is mounted on a fulcrum pin held by a pedestal secured to the centerpiece. At its lower end it is connected to the piston rod crosshead by a link, while near its top is mounted a pin which engages a tappet. On the valve rod link are two adjustable tappet collars. The tappet moves between these collars as the level is moved by the action of the piston rod. These tappet collars transmit the action of the tappet to the valve rod through the valve rod link and knuckle. The valve rod is connected inside the chest to the valve yoke, which sits over the stem of the main slide valve. This valve yoke throughout its entire stroke carriers with it the auxiliary valve which is set in one side. Toward the end of the movement of the valve yoke it is brought in contact with the stem of the main slide valve, slowly moving it until the auxiliary valve causes the main valve driving piston to operate. This last action takes place as follows: As soon as the valve rod begins to move, the auxiliary valve carried by the valve yoke closes the exhaust and opens the steam port of the main valve driving piston, admitting steam to the end from which it had last been exhausted. This balances the valve driving piston, which remains balanced until the main slide valve has been moved to near its central position. At this point the auxiliary valve closes the steam port and opens the exhaust port on the end toward which the valve driving piston is moving, allowing the remainder of the movement of the slide to be completed by the unbalancing of the steam pressure on the valve driving piston. This causes the pump stroke to be reversed.

2. Since it requires more energy to bring a body from a state of rest into motion than it does to accelerate that motion, it will be seen that, by mechanically moving the slide valve from a state of rest and continuing that action through approximately half of the valve's travel, when steam is exhausted from one end of the main valve driving piston, the pressure on the other end instantly and easily accelerates the movement of the slide valve. Positive action is thus obtained. The main valve driving piston is so constructed that danger of sticking is obviated.

9470.7 WARREN PISTON VALVE

The action of this valve is as follows:

1. Referring to figure 9470-4, the operating lever is mounted on a fulcrum pin held by a pedestal secured to the centerpiece. At its lower end it is connected to the piston rod crosshead by a link, and at the upper end it is connected by a pin to the valve rod link. On this link are two adjusting collars. An auxiliary lever with the lower end pivoted on a fixed pin connects to the valve rod. On this lever is mounted a tappet which is moved by the adjusting collars on the valve rod link, one located above the tappet and one below. The motion is thus transmitted from the piston rod to the valve rod, attached to which is the auxiliary piston. The action of the auxiliary piston controls the opening of steam and exhaust ports for the

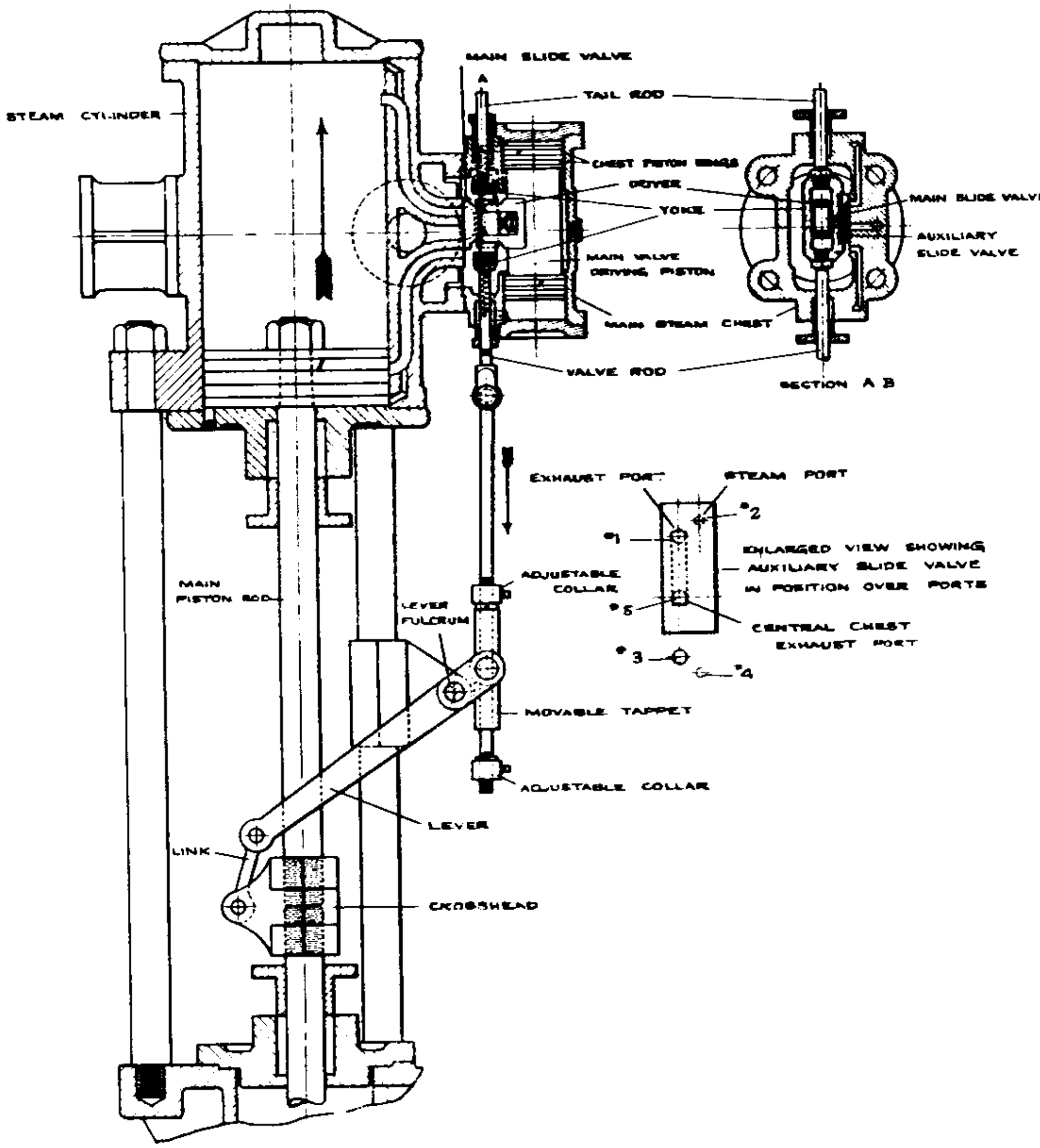


Figure 9470-3

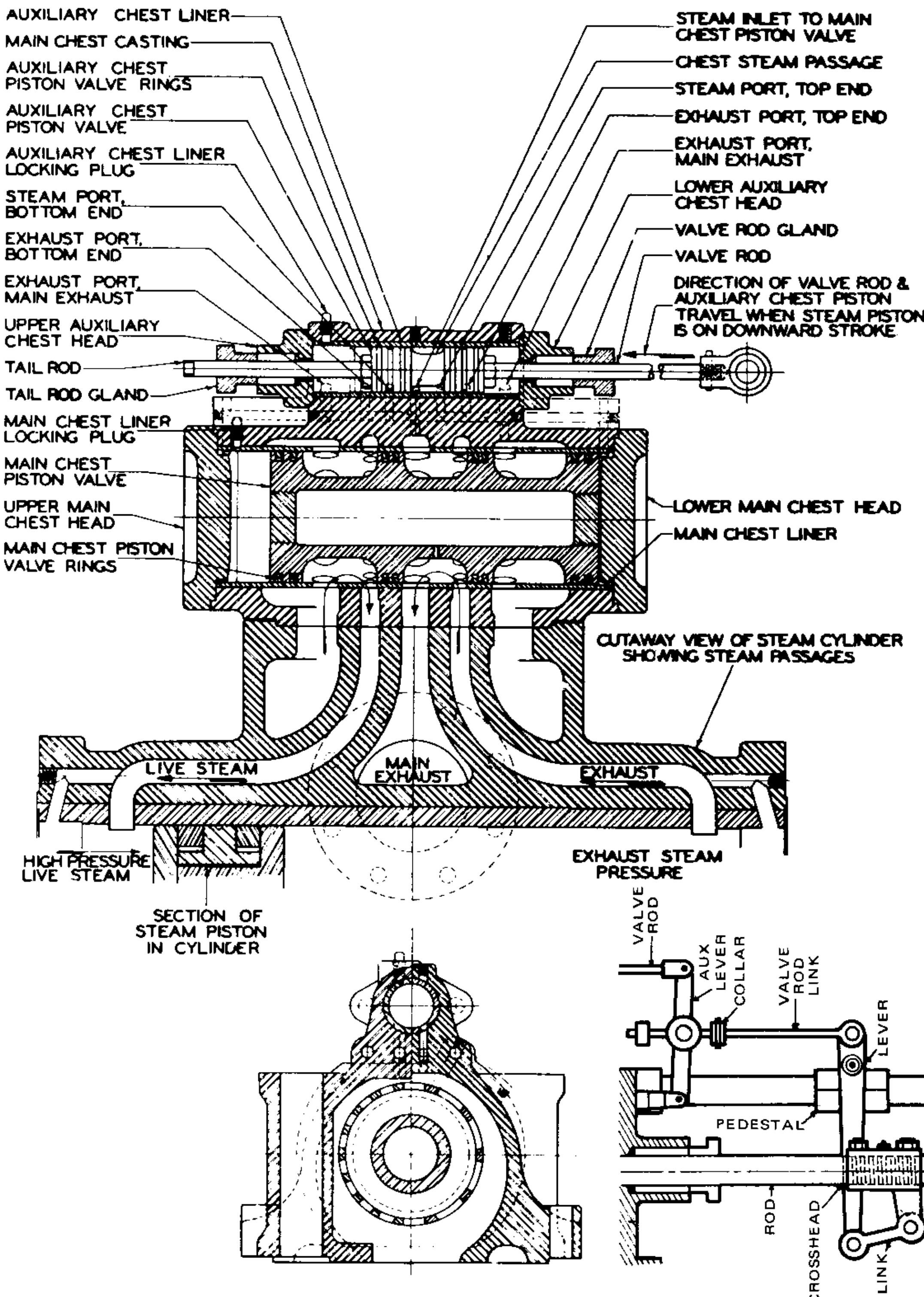


Figure 9470-4

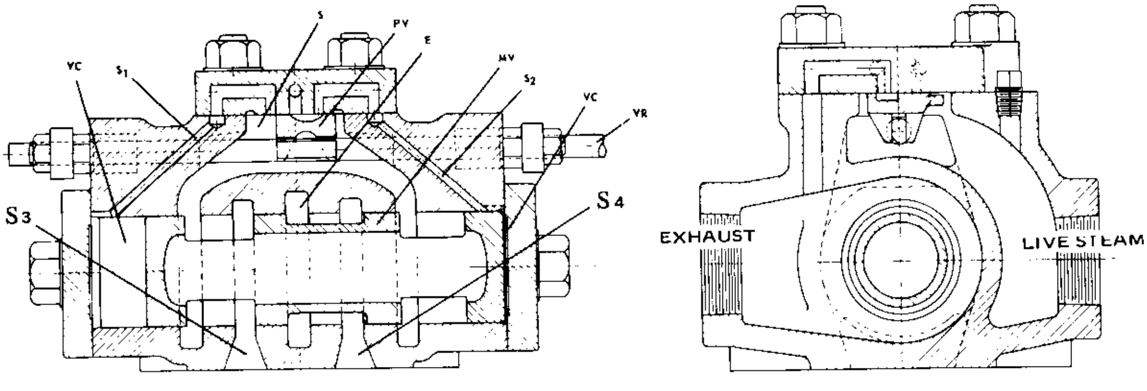


Figure 9470-5

main piston valve causing it to move and open the steam and exhaust ports to the main steam piston.

2. Referring again to figure 9470-4, the steam piston having started its downward stroke the valve rod link is moved upward and the lower adjusting collar, toward the end of the stroke, comes in contact with the tappet, thereby transmitting the upward movement to the auxiliary lever, the valve rod, and the auxiliary chest piston valve. As the auxiliary chest piston valve begins to move upward, it closes the steam port, top end, and with just a little lag closes the exhaust port, bottom end. Traveling further it opens the top exhaust port and with just a little lag opens the steam port, bottom end, thus allowing steam to reach the bottom end of the main piston valve pushing it upward to the top end of the chest, reversing the flow to the main steam piston in the cylinder.

3. The valve rod and tail rod stuffing boxes are subject to exhaust pressure only. Live steam enters only between the outside spool on each end of the auxiliary chest piston valve and the next inner spool. The exhaust is between the central spools and there is a balancing port running parallel with the auxiliary chest piston valve, connecting the exhaust passage with each end of the valve. The main chest piston valve is fitted with four snap rings. The auxiliary chest piston valve also is fitted with snap rings. Both operate in renewable liners of alloy iron which have been bored and honed. The tension of the snap rings prevents the main valve from moving when steam is turned off.

9470.8 WORTHINGTON SIMPLEX VALVE

This valve is illustrated in figures 9470-5 and 9470-6. Its operation is as follows:

1. The pilot valve PV is moved back and forth by the valve rod VR which, through links and levers, is actuated from the pump piston rod R. The pilot valve controls the admission and exhaust of steam to the ends of the valve chest VC. This moves the main piston valve MV back and forth.

2. The main piston valve controls the admission and exhaust of steam to the ends of the steam cylinder C of the pump. With the pilot valve in the position as shown in figure 9470-5, steam is admitted from the live steam space S through the left hand steam port S_1 to the end of the main piston valve forcing it to the position as shown.

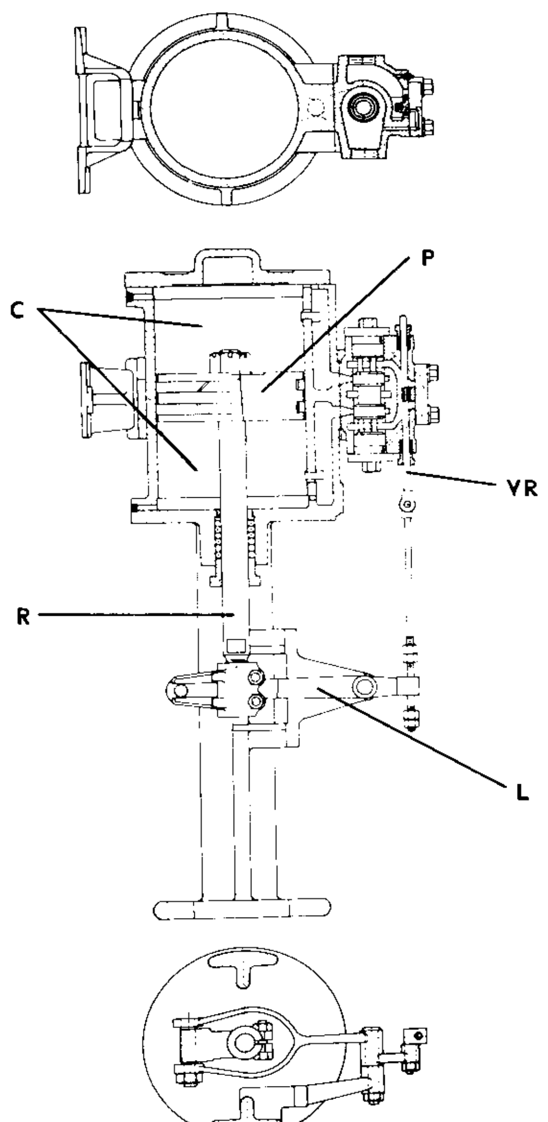


Figure 9470-6

3. With the main piston valve MV as shown, steam is admitted from the live steam space through the left hand port S_3 into the main steam cylinder C forcing the steam piston P towards the opposite end. At the same time the steam from the opposite end of the steam cylinder is exhausted through the right hand port S_4 to the steam exhaust E.

4. When the steam piston has reached the opposite end of the cylinder, the valve rod linkage L will have moved the pilot valve to the left so that live steam is admitted from the live steam space through right hand steam port S_2 to the other end of the main piston valve MV forcing it to the opposite end from that shown. In this position live steam is admitted to the main steam cylinder through the right hand port S_4 forcing the piston toward the opposite end. At the same time the steam from the opposite end of the steam cylinder is exhausted through the left hand port S_3 to the steam exhaust.

9470.9 WORTHINGTON PISTON VALVE

1. Figure 9470-7 shows the Worthington Navy-type piston valve. The reciprocation of the main steam piston P is controlled by the functioning of main piston valve V and auxiliary piston valve AV. The auxiliary valve AV is mechanically operated through lever and cranks from the piston rod. When steam is admitted through the steam nozzle, it fills the chambers marked S. There is one steam port and one exhaust port leading to each end of the auxiliary cylinder AC. The admission and exhaust of steam to and from the auxiliary cylinder is controlled by movement of the auxiliary valve AV. This causes the main steam valve V to reciprocate and admit and exhaust steam to and from the main steam cylinder MC and thus move the main steam piston P forward and back.

2. The exhaust from the main and auxiliary cylinder is also controlled by the main and auxiliary valves respectively and lead into the chamber E and out through the exhaust nozzle.

3. The strokes of the pump may be lengthened or shortened by moving the collars on the valve rod link farther apart or closer together.

9470.10 CAPACITY CHARTS

For determining the capacity of a reciprocating pump handling liquids (not air pumps), figure 9470-8 will be found convenient. The capacities indicated apply to single-cylinder pumps, in good repair, operating at full stroke.

9470.11 POWER PUMPS

Low pressure, motor-driven reciprocating pumps (power pumps) are occasionally used in naval service for fresh water sanitary, bilge, ballast, and fuel oil transfer services. For small capacities they have a better efficiency than other types and are self priming. Such pumps are generally horizontal. Both plunger and piston types are used. Their crankshafts are driven from the motor by gearing or V-belt drive. The liquid end of these pumps is similar to that of direct action steam pumps and the instructions in parts 2, 3, 4, and 5 of this section should be followed.

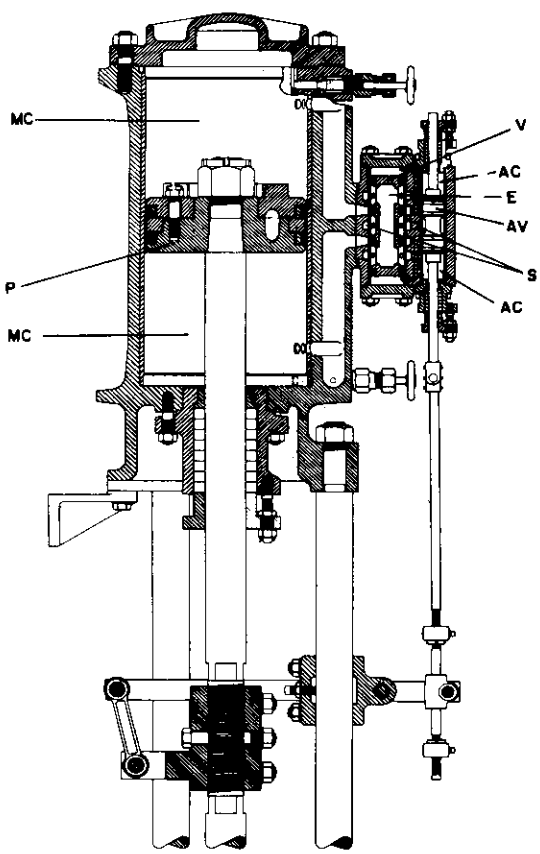


Figure 9470-7

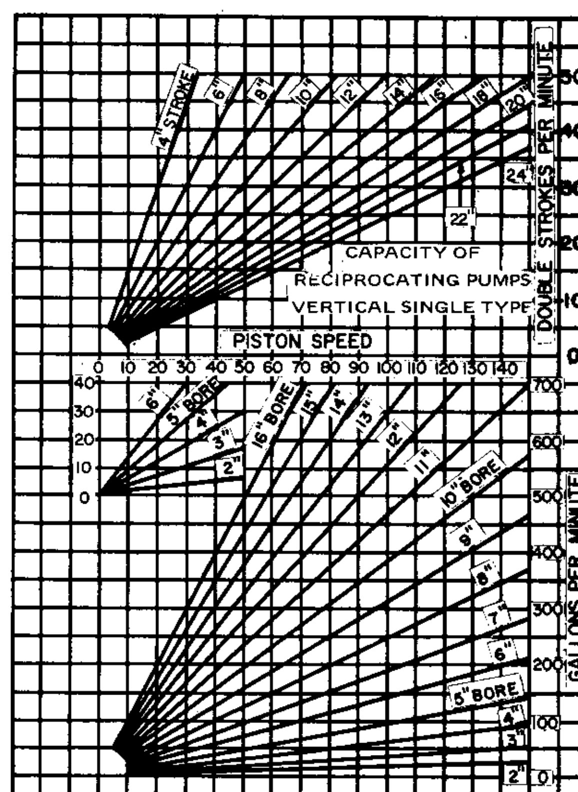


Figure 9470-8

9470.12 VARIABLE CAPACITY BOILER FEED PUMPS

Special types of plunger pumps are used for boiler feed service on a number of auxiliary ships. These pumps are variable capacity triplex units and are governed by controlling the plunger stroke by automatic devices. This permits a stepless variation in capacity from zero to maximum using a constant speed motor or turbine. The inherent high efficiency of the plunger pump for low capacity at high pressure is thus maintained over a wide range of pump output. Figure 9470-9 shows the Aldrich Groff variable capacity feed pump. Figure 9470-10 shows the general arrangement of the Worthington pump; figure 9470-11 shows the detail of the crank and eccentrics and the stroke-shifting mechanism.

Part 2. Operation

9470.21 PREPARING PUMP FOR OPERATION

The following steps should be taken before putting a pump into operation for the first time, after an overhaul, or after the ship has been drydocked:

1. Check alinement and correct if necessary. If pump is operated out of line, scoring of rods and liners will result.
2. Steam and liquid lines should be free from scale and foreign matter.
3. Check all packing and repack if necessary.
4. Move steam pilot valve rods by hand to be sure pilot valve moves easily.
5. Check all connections and fittings to ensure they are tightly in place.

9470.22 STARTING PROCEDURE

To start a reciprocating pump proceed as follows:

1. Oil the pins of the steam valve operating gear and set up on all grease cups.
2. Open the liquid end valves:
   a. Suction.
   b. Discharge.
3. Open the cutout (or root) valves in the:
   a. Exhaust line.
   b. Steam line.
4. Open steam cylinder drains:
   a. Top.
   b. Bottom.
   c. Valve chest.
5. Open exhaust valve at pump.
6. Crack the throttle valve and open slowly so as to admit steam and warm up gradually.
7. Close the steam cylinder drains after the pump makes a few strokes and the steam cylinder is clear of water.
8. Bring the pump up to the proper speed by sufficiently opening the throttle valve. If pump is controlled by a pressure governor, open throttle gradually until governor takes control of pump and then open the throttle valve fully.
9. Adjust the cushioning valves, if fitted, until an adjustment is obtained that permits silent and smooth working of the pump, i.e., sufficient pump speed at the end of the stroke without knocking. After best point of operation is obtained, cushion valve should be set and not changed.

9470.23 STOPPING AND SECURING

To stop and to secure, proceed as follows:

1. Close the throttle valve.
2. Close the exhaust.
3. Open cylinder drains.
   a. Top.
   b. Bottom.
   c. Valve chest.
4. Close the water end suction valve.
5. Close the water end discharge valve.
6. Close the steam and exhaust cutout valves (root valves).
7. After steam cylinder is drained, close the valve chest drains leaving the steam cylinder drain valves open to prevent hydraulic action. (See article 9470.25.10.)

9470.24 FAILURE TO START

If a pump fails to start proceed as follows:

1. Secure the pump. Do not attempt to adjust the tappet collars.
2. Examine the discharge and the exhaust line for closed valves or for a valve disk that has become detached from its stem.
3. If none is found closed, the steam piston may be frozen, particularly if the pump has not been in service for some time. Jack the pump with a bar to determine if there is excessive friction. If so, the trouble probably lies here. (See article 9470.81.)
4. Disconnect the auxiliary valve stem from the operating gear without disturbing the adjustment of the tappet collars. Open the exhaust, suction, and discharge valves and then crack the throttle. Work the auxiliary valve by hand. (The auxiliary valve should work freely by hand.)
5. Should the pump still refuse to start, secure the pump. Remove the steam valve chest cover and examine the main valve to see if it has over-ridden or stuck.
6. If the pump cannot now be started, a complete overhaul of the working parts of the steam end is necessary to stop steam leakage (either in steam piston or valves), the most probable cause of the pump's failure to start.

9470.25 TROUBLES

The main features in the operation of a reciprocating pump, with the principal causes of trouble and the remedies therefor, are enumerated below:

1. **Jerky operation.** Jerky operation on starting is usually caused by a failure to take suction. To correct this proceed as follows:
   a. See that all stop or check valves in the suction line are open and that the line is clear of obstructions.
   b. Should a reciprocating feed pump become vaporbound when taking suction from a variable speed feed booster pump, immediately speed up the latter unit. Unless feed pump regains suction immediately, shift suction to reserve feed tanks until pump is cooled off; turn a hose on the pump cylinder. In the event a standby pump is available, shift to it until the vapor-bound pump has cooled down. Vent valves, if provided, on pump valve chest covers or discharge line will assist the pump in freeing itself of vapor.
   c. Pumps having a suction lift, such as bilge pumps, may require priming before they will take a suction. Salt water pumps can usually be primed from the sea by opening the sea suction valve for a short interval.
   d. Tight piston packing may cause jerky operation.
2. **Loss of discharge pressure.**
   a. Low steam pressure.

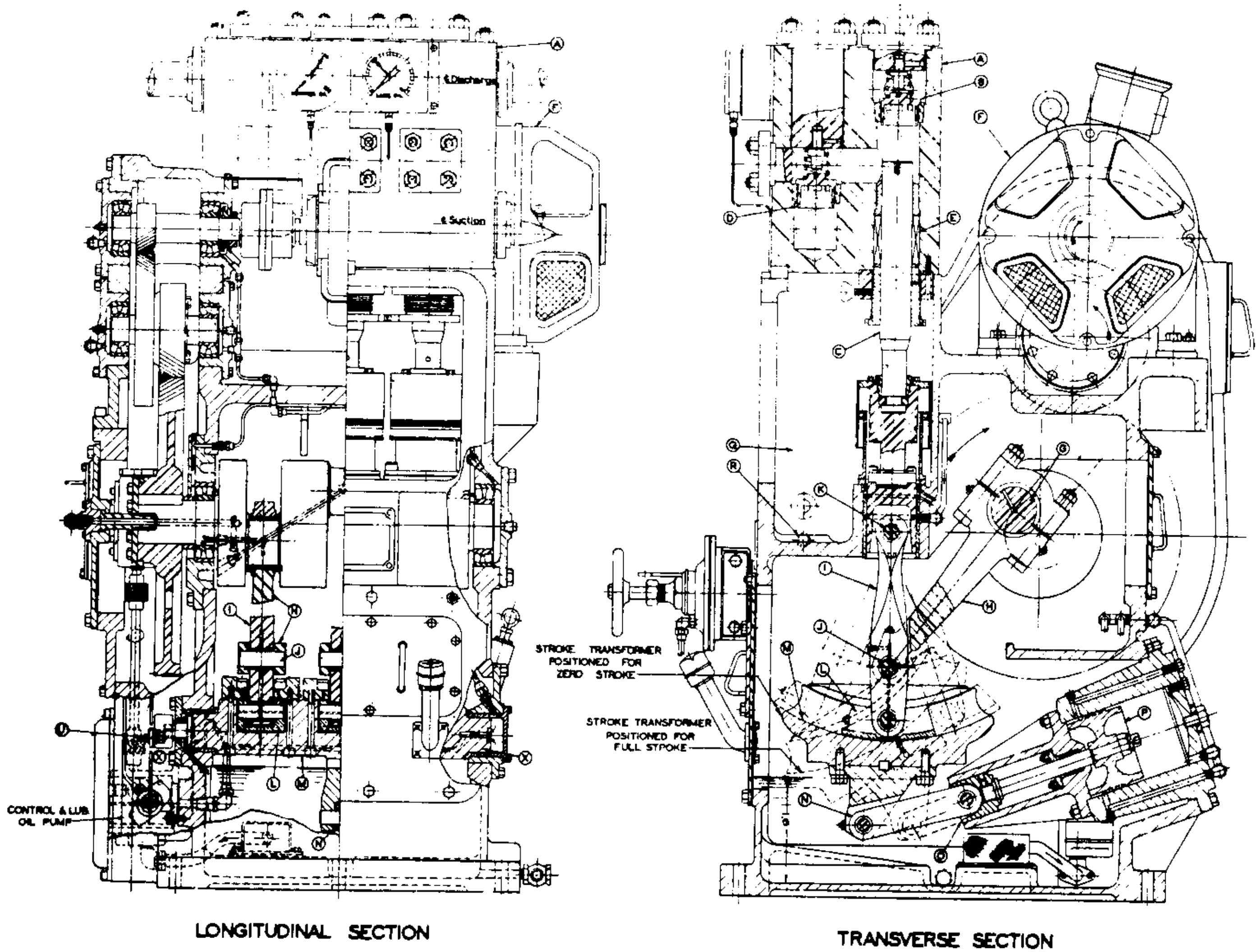


Figure 9470-9

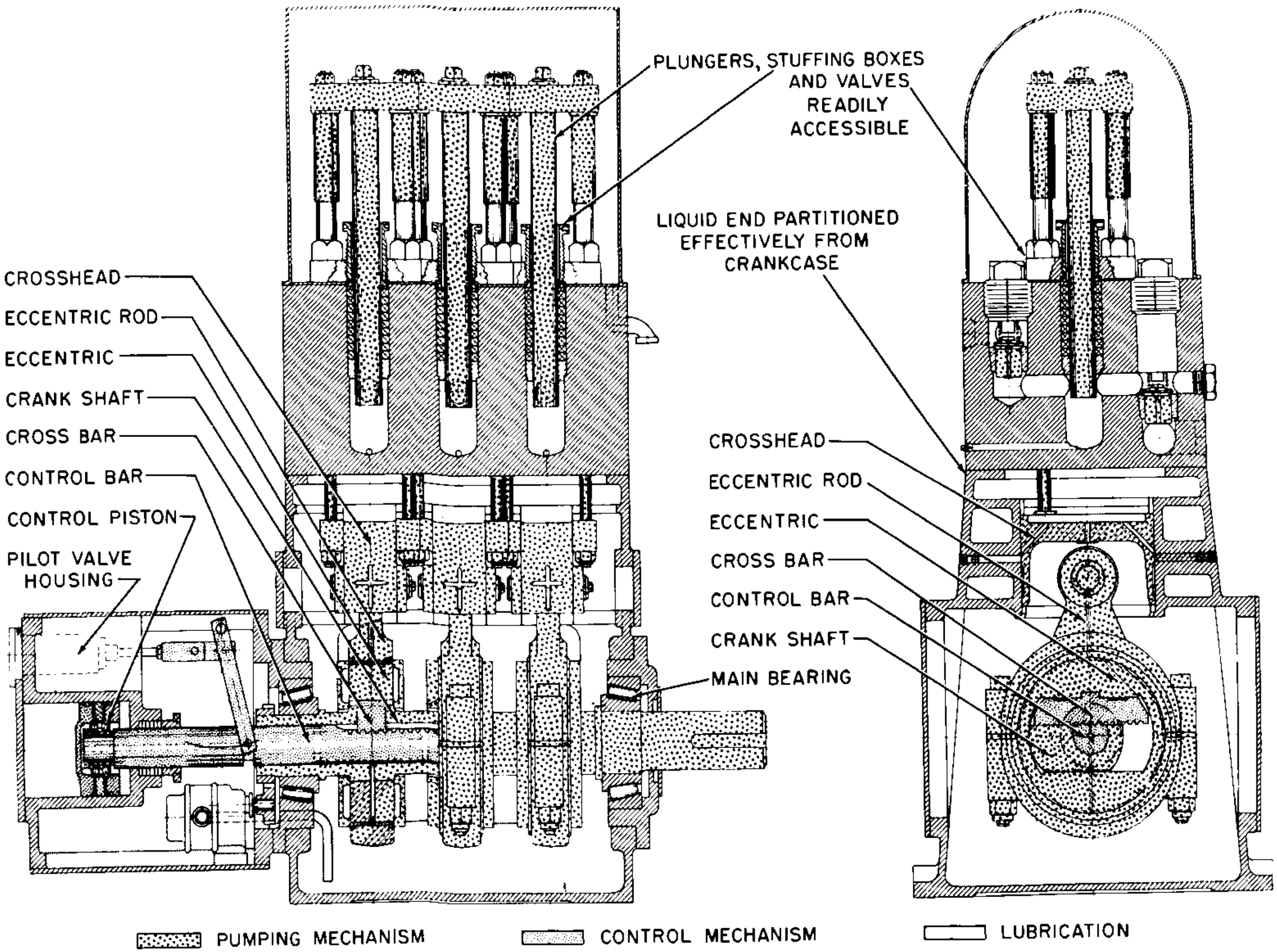


Figure 9479-10

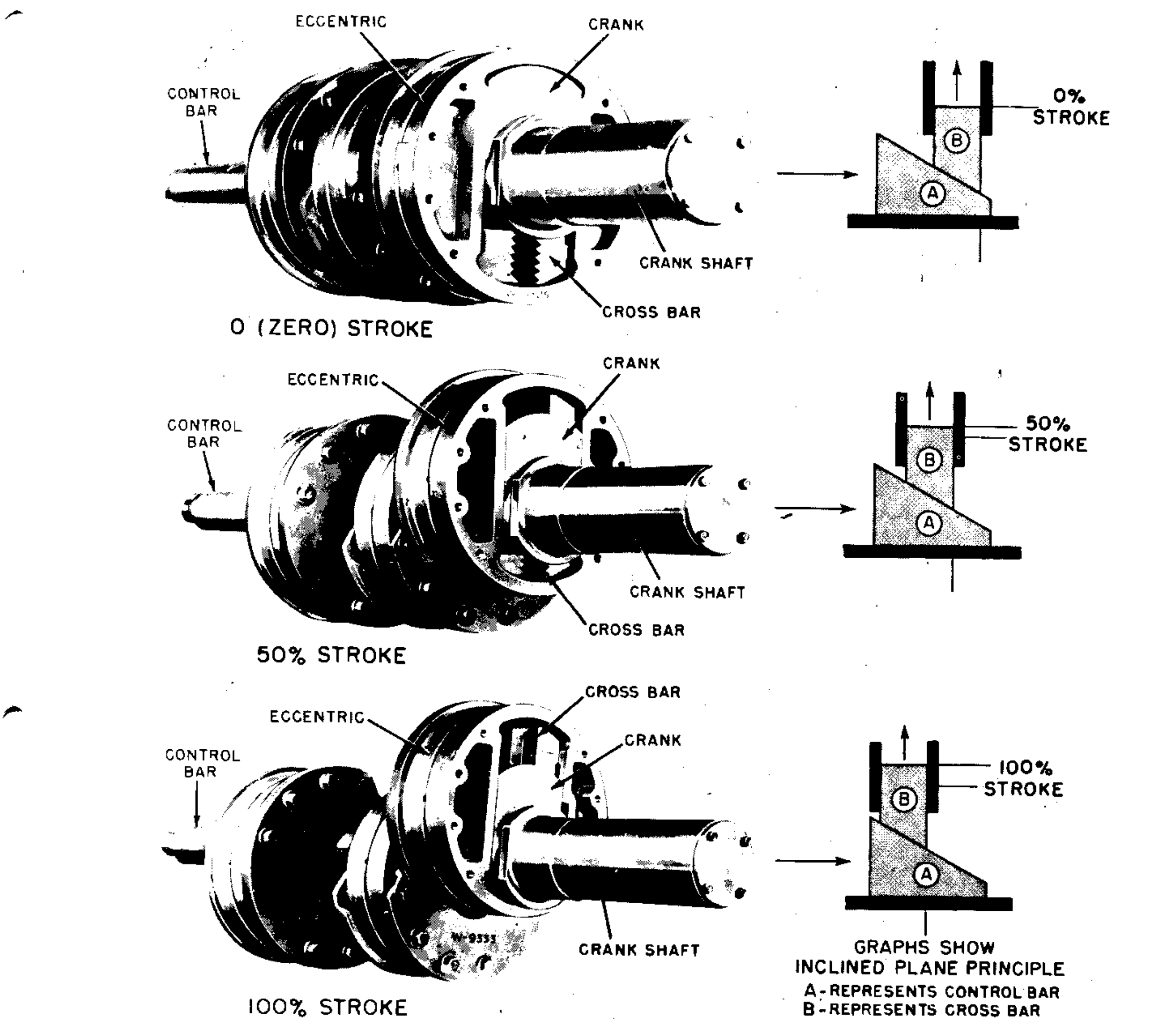


Figure 9470-11

b. High back pressure.

c. Piston rings in steam end worn.

d. Excessive friction loss such as improperly packed stuffing boxes or binding of liquid piston packing.

e. If a pump races without appreciably increasing the discharge pressure, such racing is caused by a leaky piston; leaky, broken, or stuck valve in the water end; or by air being admitted through open or leaky valves in the suction line. Stop the pump as soon as practical in order to ascertain and correct trouble.

f. Should a pump which has been running properly suddenly lose pressure on one stroke, look for a broken valve which should be replaced at once. Great loss in efficiency results from leaky suction, discharge valves, and leaky pistons. Previous experience with a particular type of pump may be taken as a guide in deciding where to look for trouble. Under ordinary conditions, the first investigation should be of the most accessible parts. (See articles 9470.62 and 9470.63 in regard to proper adjustment of valve guards and valves so as to prevent leakage.)

3. **Loss of capacity.** Loss of capacity and failure to develop pressure are closely related troubles, due to the fact that a drop in one will generally be reflected in a drop of the other. Some additional causes of failure to develop required capacity are as follows:

a. Insufficient speed of pump.

b. Suction lift is excessive.

c. Air leakage into pump through stuffing box or leaks in suction line.

d. Entrained vapors in liquid pumped.

4. **Pounding.**

a. Pounding in the liquid end can be caused by improper cushioning, too heavy valves, or a loose piston. This pounding can be stopped by:

(1) Adjusting the steam cushioning valves.

(2) Examine the pistons to see that they are tight on the rod, and the piston rod for lost motion where it is secured to the crosshead.

(3) Look for a loose nest of valves or loose zinc plates, if fitted.

b. If the pump is not fitted with an air chamber on the suction side and pounds, a snifting valve installed on the suction side usually stops the pounding.

c. An air chamber should be fitted in suction lines when there are high velocities.

d. To stop pounding in the water end of a pump having a considerable suction lift:

(1) Slow down the pump, as this is usually due to water hammer or ram effect in the suction piping.

(2) See that air chamber is properly charged.

5. **Groaning.** Groaning in the water end is generally due to the packing being too tight, but may be the result of misalinement, a broken follower, or a stuck valve. Therefore, the pump should be stopped and examined at once. Failure to investigate usually results in at least a scored cylinder.

6. **Erratic operation.** When a pump operates erratically, sticks in any part of the stroke, or stops frequently, with the throttle valve opened the proper amount, the cause is in the steam end and is probably due to one or more of the following defects:

a. Lost motion in the operating gear due to wear. This should be remedied by rebushing and, if necessary, renewing the pin at the affected part.

b. Leakage of steam by the main or the auxiliary valve due to wear of either. To remedy this, spot in the defective valve on its seat.

c. Leakage of steam by, or sticking of, the valve chest piston. To stop this it is necessary either to rebore the valve chest cylinder and renew or to refit the piston rings.

d. Excessive leakage by the steam piston rings. To prevent this leakage it may be necessary to rebore the steam cylinder or to renew the rings, or both. In some cases where split rings are fitted, the leakage can be partly stopped by taking out the rings and peening them so as to increase the wall pressure. When cylinders are rebored, oversize pistons should be fitted.

e. Small ports and passages in the valve chest stopped up with scale. This frequently happens on new ships as a result of failure to blow out all scale from steam lines before connecting up.

7. **Knocking in steam end.** Knocking in a steam cylinder is indicative of a loose piston assembly or a piston loose on the rod. The pump should be stopped at once and the exact trouble discovered by measurement and examination; then rectified.

8. **Groaning in steam end.** Groaning in the steam cylinder is usually due to steam piston being cocked on the piston rod, to broken rings, or to the cylinders being out of alinement. The trouble, unless immediately rectified, will result in scoring the cylinder walls. If a pump has been idle for a long period, rust some times forms and may cause groaning when the pump is started.

9. **Worn piston rings.**

a. Should the pump stop when going very slowly, it may be due to steam passing worn piston rings. The appearance of a good bearing on the piston ring does not necessarily indicate a good fit. They should be removed from the cylinder so that measurements of both rings and cylinder may be checked for excessive clearance.

b. Should a pump stop or stick, do not jack it with steam turned on; to do so may result in serious damage to it or injury to personnel. (See article 9470.81.)

10. **Hydraulicking.**

a. Water and such liquids as are usually handled by pumps are incompressible. If a pump is fully primed and the discharge outlet closed tight and steam admitted to the steam cylinder, the pump piston and rod would not move if it were not for the slip or leakage from one side of the pump pistons to the other, equalizing the pressure. Under these conditions, on the down stroke or the stroke where the pump rod enters the liquid cylinder, the liquid pressure in the cylinder will build up in the ratio of the area of the steam cylinder to the area of the pump rod. That is termed hydraulic action in pumps.

b. Assuming the size of a steam cylinder as 11-inches diameter and pump rod 2-5/8-inches diameter, and the steam pressure 575 psig, the resulting liquid pressure

$$= \frac{\dfrac{\pi (11)^2 \times 575}{4}}{\dfrac{\pi (2.625)^2}{4}} = \frac{95.03 \times 575}{5.42} = 10{,}081 \text{ psig}$$

If such hydraulic action is allowed to occur, it will obviously break the liquid cylinder. This usually occurs when the pump is secured and the steam throttle valve leaks or is not carefully closed.

c. To prevent hydraulic action always open steam cylinder drain valves when pump is shut down and leave them open until pump is again started and cleared of condensation. Check to make sure that relief valve in discharge line functions properly. Make sure that relief valve is located between pump and the discharge shutoff valve in the discharge line.

11. **Vapor lock.** Fuel oil transfer pump suction lines in particular may have vapor pockets, due to the lines having to go over obstructions. High viscosity oil in a long line has a high friction loss and it is often necessary to heat the oil to lower its viscosity and the friction loss. This may be accompanied by considerable gasification. With a vacuum on the suction line, this gasification will cause vapor locks wherever pockets exist. This condition also exists when a new oil supply has been taken aboard.

Part 3. Maintenance (where 3-M system is installed, conduct preventive maintenance in accordance with 3-M cards)

9470.31 LUBRICATION

The pins of the valve operating gear should be oiled frequently. Never use oil in the steam or water cylinders, the valve chest, nor on piston rods.

9470.32 JACKING

All pumps should be moved daily and logged.

9470.33 GLAND LEAKAGE

Great care should be exercised in setting up on glands. If the two sides are set up unequally, the gland becomes tilted and scores the rod, and sometimes breaks the gland itself. If a gland is set up too hard, the packing will probably score the rod. If a gland continues to leak after it has been given a few turns on the nuts, the best thing to do is to let it alone until the pump can be shut down. Then an examination should be made to see if the throat bushing is too large, possibly due to the rod having been turned down, or to see if only new packing is needed. Particular attention should be given to packing for cylinders operating at low pressure or below atmospheric pressure to prevent not only loss of vacuum, but the admission of grease into the cylinder and of air into the feed system.

9470.34 STEAM VALVE CHESTS

1. Preventive maintenance of steam valve chests operating under high steam pressures is most important to avoid serious steam cutting and wear. Routine inspections will disclose early signs of leakage or wear and need for corrective action. Careful attention to lapping out these early cuts will ensure a tight metal-to-metal fit of the valve chest and steam cylinder.

2. Attention to condition of piston rings on the main and auxiliary valve with timely replacement will preclude broken rings and steam cutting in the grooves.

3. Prior to making up the metal-to-metal joint, it should be blued and tested for good overall contact. If it is not a good metal-to-metal fit of if scores or leakage are evident, remove the studs from the steam cylinder and lap the steam chest in against the cylinder seat.

4. Strict cleanliness must be observed in making up the joint so that no particles of dirt or scale will prevent a good metal-to-metal contact of the ground faces. A thin film of copaltite should be applied. The holding-down bolts should be taken up gradually and alternately in a diagonal sequence. After steam is applied and the pump is in operation, the holding-down bolts should be taken up again.

5. Do not install gaskets on steam chest joint. Such a gasket tends to blow out around the steam passage lands, thus aggravating the cutting action of the steam.

6. When continued lapping of valve chest and steam cylinder surfaces have worn them down to the extent that they must be built up to original dimensions, then both should be overlayed and brought back to design dimension. When this is necessary, follow the instructions in article 9470.75.

9470.35 ADJUSTMENT OF STROKE

1. Pumps should be made to run with the full length of stroke which will ensure that the piston travels a little beyond the counterbore.

2. Full stroke is that which is stamped on the nameplate. (See article 9470.2.2.) A stroke indicator, consisting of a sheet metal pointer secured to the piston rod crosshead and two marks on one cylinder tie rod, is usually provided. The upper mark lines up with the pointer on the crosshead when the pump is at the upper end of its stroke, while the lower mark indicates the proper position of pointer with pump at the lower end of its stroke. If such stroke indicator is not already fitted, it should be installed by the ship's force, as it materially assists in the proper setting of the valve gear, and permits a ready checkup of pump stroke. This ensures steam economy and better operation of the pump. When full stroke cannot be obtained, it invariably means that something is wrong with the adjustments of the pump. A short stroke results in incomplete cushioning and the formation of shoulders in the cylinders and valve chests with resultant breakage of rings and followers. A long stroke is usually indicated by a heavy metallic knock in the steam cylinder and should be immediately corrected. It should not be necessary to keep moving the tappet collars continually while the pump is running; if found necessary to do so, something is wrong with the pump and it should be dismantled and the interior parts examined. Do not alter the settings of the tappet collars without orders from the engineer officer.

3. The proper valve setting must be carefully determined to make a pump take full stroke. Refer to manufacturer's drawings and instruction books for detailed information as to the method to be followed for specific makes and types of valve gear. The following general method is included as a guide when more detailed information is not available: Place piston and auxiliary valve on the center or half stroke. Then move each collar so that they will be about one-half the width of the steam port away from the tappet. Start the pump. If the stroke is too short, the collars should be screwed farther apart. Care should be taken to move both collars the same amount, otherwise stroke will be longer on one end than on the other. When the final adjustment has been made, lock the collars securely in place.

4. a. To adjust the steam valves of a duplex pump, place one piston on its top striking point and, by removing the steam chest cover of the valve chest of the other piston, adjust and secure the valve of the other piston to give an excess on one-eighth inch of full port opening at the top end. To set the other valve, repeat the operation with the other piston.

b. Run the pump slowly with the throttle cracked, against little or no pressure, and with the cushioning valves wide open if fitted. The pistons should be striking on the cylinder heads; if they do not, there is some undue friction, possibly due to tight piston rod and plunger packing. Close down on the cushioning valves until the pump is running at full stroke of both steam pistons with smoothness of reversal and no striking. Should it be impossible to obtain this smoothness of reversal, it may be necessary to alter slightly the adjustment of the valve operating collars.

c. To shorten the length and prevent striking of the piston against the cylinder heads, set the collars close together; to lengthen the stroke, set the collars farther apart. Care should be taken to move both collars the same amount, otherwise stroke will be longer on one end than on the other.

5. Some pumps may, due to design characteristics or wear, override and pound at full speed after having been adjusted to give full stroke at low speed. The only remedy for this is to have different valve adjustments for different speeds.

6. After a pump runs on short stroke for a considerable length of time, difficulty may be encountered in making the pump take a full stroke, due to shoulders having been worn in the steam valve seat and pump cylinder lining.

These shoulders will have to be removed before full stroke can be obtained.

9470.36 PROTECTION AGAINST CORROSION

On salt water pumps having cast iron bodies and parts in contact with salt water, special care shall be taken to guard against corrosion. These parts shall be examined every six months, scaled, and wire brushed insofar as practical. Reduction of corrosion can be effected by fitting zinc plates or rings on water end valve chest covers and on cylinder covers.

9470.37 EXAMINATION OF WATER VALVES

For satisfactory and economical running of pumps, it is essential that the valves should be absolutely tight. The liquid end valves in all pumps should be examined quarterly and all foreign matter on valves, valve stems, and valve springs removed.

9470.38 TESTS AND INSPECTIONS (RECIPROCATING PUMPS)

1. **Weekly.** Operate all pumps by steam or power. If power is not available, move by hand.
2. **Quarterly.** Check settings of relief valves.
3. **Annually. (See also 9470.52, para. 2)**
   a. Inspect liquid end, valves, valve stems, and springs.
   b. Inspect steam valve gear for wear.

NOTE: These tests and inspections are the minimum necessary to give adequate assurance of safe and reliable operation of equipment. Indications of low discharge pressure or other manifestations of improper operation should indicate more frequent or extensive tests and inspections.

Part 4. Repairs

9470.51 GENERAL

If a pump does not function properly and is in need of repair, do not dismantle the steam end until a thorough investigation shows that the trouble is not at the water end. Most pump troubles are due to fouled water cylinders, worn valves, or conditions in the pipe connections external to the pump.

9470.52 DRAWINGS AND DATA

1. When repairing or making an examination of a pump, it is essential that assembly and detail drawings and available dimensional data thereto be at hand. Frequently after overhaul, important dimensions become altered, such as the width of and distance between steam ports, the length of rods and steam valves, and the diameter of pistons. This may cause poor operation which will continue in spite of other major repairs unless the real cause of trouble is recognized and rectified.

2. Whenever reciprocating pumps are opened up for repairs (at least annually), micrometer measurements shall be taken of the cylinders and valve chest on the fore and aft and athwartship diameters at the top, middle, and bottom, and the results recorded on Form OPNAV 4700-2G (NAVMAT Notice 4700) card with an accompanying diagrammatic sketch showing measurements obtained and the date on which they were made.

9470.53 SCORED WATER CYLINDER

A scored water cylinder need not necessarily be rebored or renewed. Scores should be smoothed up by stoning to reduce leakage and wear on the packing. When the scoring is extensive and the packing rapidly wears out, reboring or renewal of the liner should then be considered.

9470.54 LOOSE PISTONS

1. Steam and water pistons frequently work loose on the rods. This fault may be due to poor fitting so that the shoulder on the rod brings up against the piston without giving a proper bearing surface for the tapered part of the rod. The piston should fit within 1/32 inch to 1/8 inch of the shoulder when set up hand tight, depending on the taper of the rod, and should then be forced tightly against the shoulder by the securing nut. A small amount of foreign matter on the tapered part of the rod will prevent the piston from being brought home firmly. If a piston which has given trouble is properly refitted to the rod, the trouble will usually disappear.

2. If a jamb nut or split pin is not fitted, one should be installed.

9470.55 TEST FOR TIGHTNESS

The following method of testing the tightness of the steam piston in the cylinder has been used with success:

1. Remove the cylinder head. Shore the piston to prevent forward or upward motion.

2. Connect a steam hose to the lower drain cock and gradually raise the pressure to the working pressure of the pump or the hose, whichever is less. If the rings are not tight, steam will leak past them. It must be remembered that this test shows defects for only that portion of the cylinder occupied by the piston. If measurements show that there is a great difference in size in various parts of the cylinder, the test should be repeated several times with the piston in a different part of the cylinder during each test.

9470.56 PISTON TOO SMALL

On reboring a water cylinder, the piston and follower may be given so much clearance as to require renewal. As a temporary substitute, until a new piston and follower can be obtained, one of the following procedures may be employed:

1. Install upper and lower rings of metal packing and insert soft packing between them.

2. The piston and follower may be turned down, threaded, and a ring with same thread screwed tightly on. The outside diameter is then machined to fit the cylinder.

9470.57 BUILDING UP PISTONS

Pistons may be built up by flowing on metal by the oxyacetylene method, electric welding, or metal spray. The piston is then machined to proper fit.

9470.58 FITTING SOFT PACKING

When packing a pump with Tuck's, flax, or other soft packing, soak the packing in hot water approximately 12 hours before fitting and installing. Should an urgent demand for the pump make this impractical, the packing should be put in to fit loosely and allow for swelling. A failure to do this will surely cause the pump to groan or result in a scored cylinder.

9470.59 LAMINATED PHENOLIC PACKING

Laminated phenolic packing type rings are increasing in usage, particularly for boiler feed service. The Naval Ship Systems Command is conducting laboratory and service tests on various types and makes to determine their suitability and to develop data as to their proper application and installation. The use of laminated phenolic rings as a substitute for soft packing may be authorized by NAVSHIPS upon request, provided the design of the piston is satisfactory for or can be altered to accomodate the phenolic type rings.

9470.60 METAL RINGS

Metal rings are sometimes very narrow and cut the cylinder. If wider rings are fitted, the trouble generally stops. Sharp edges of rings should be rounded with a file. The use of white metal rings is considered obsolete and unsatisfactory.

9470.61 BREAKING OF FOLLOWERS

1. Breakage of followers and bolts may be due to misalinement. Another cause may be screwed plugs in the piston working loose and coming adrift. Whenever a piston is removed for examination or repairs, check for tightness of plugs. Plugs should be prick-punched to prevent their backing out.

2. However, if this trouble in general in any one set of pumps on board ship, it may be that the followers are too weak, in which case the Naval Ships Systems Command should be advised so that the design may be improved and alterations authorized.

3. In pumps where the follower is too small in diameter, soft packing may roll up between it and the cylinder causing the follower to jam and break.

9470.62 VALVES IN WATER CYLINDER

1. For satisfactory and economical running of pumps, it is essential that the valves in the water end be absolutely tight. Cast disc valves may be faced off in a lathe and then ground in on their seats by a simple device which consists of a rod of sufficient length, slotted for a piece of metal which fits into a slot across the top of the valve. An ordinary bit stock can be used to do the grinding.

2. It is sometimes desirable to take a cut off the valve seat without removing it. A simple cutter for this can be made with an extension for a bit stock similar to the grinding device. When flat valves are fitted, the seats may be trued up by using a small surface plate, and spotting in the section on the surface plate.

3. At each examination try all sheet metal valve discs with a straight edge to see if they are true. Solid (cast) valves, if scored or warped, should be trued up in a lathe. Kinghorn valve (thin metal) discs, figure 9470-9, if dished or warped should be discarded and new ones installed, as they are liable to break after having been dished. The life of rubber valves can sometimes be prolonged by trimming and turning the valves and by inserting brass backings.

4. See that valve springs have the proper tension; this should be just great enough to ensure a quick closing of the valve and it should be possible to lift the valve easily by hand. See that the springs are well secured by split pins. Adjust the valves to give the proper lift; lift should be such that the circumferential opening is slightly greater than the clear opening through the seat. The lift ought never to be more than one-fourth of the diameter of the opening.

5. Keep valves clean; a light mineral oil makes a good cleanser and a lye or soda solution is good for removing caked or gummed oil from valves.

6. New valves may differ in thickness from the old, thus giving too much or too little lift; too little lift will usually cause the valve to squeal.

7. In some pumps the valve seats are secured only by a taper fit. In such cases the valve seats should be forced home by a jack resting on the end of a reseater which in turn rests on the face of the valve seat. Should the seat work loose, slightly peen over the edge of the metal. In pumps that have the valve seats screwed into the pump diaphragm, they should always be screwed in with white lead, otherwise it will be almost impossible to get them out.

8. Several valves are sometimes secured in one plate. The plate is secured in the water valve chamber with a ground joint which is difficult to make tight. If it is not tight, water will leak under the joint and score it. This joint can be reseated with ship's valve reseating machine or by being ground in with a valve grinding mixture. In difficult cases a copper gasket may give better results.

9470.63 ASSEMBLING VALVES

Great care should be exercised in assembling pumps after overhaul; otherwise, the entire work on valves, etc., may be ·t. Valves, seats, stems, and springs should be marked before removal and replaced in the same place from which they were removed.

9470.64 SNIFTER VALVE

When it is thought that a snifter valve is the cause of faulty pumps performance, it should be borne in mind that the valve is simply a check valve with an adjustable amount of opening, and its functioning is usually to prevent overloading and to permit the entrance of a certain amount of air at each stroke and thus do away with the necessity of charging the air chamber with high pressure air. A globe valve should be installed between the check valve and the pump so that the amount of air can be regulated.

9470.65 RELIEF LINE

If the relief line from a feed pump is piped back to the suction line to a point close to the pump, water may be blown away from the suction when the relief valve lifts, causing the pump to become vaporbound. This trouble can be stopped by piping the relief line to the feed tank.

9470.66 BLOWING GASKETS

1. For all cold water pumps a sheet packing of rubber with cloth insertion is suitable for the water end bonnet. For the steam end of all pumps and water ends of hot-water pumps, compressed asbestos packing is used for bonnet gaskets for steam pressures up to 300 psi. Above this pressure metal-to-metal joints should be used, or gaskets of soft iron or steel annealed dead soft. Maximum hardness of 80 Brinell may be used. When gasket blowouts occur, the trouble can usually be eliminated by turning a shoulder on the head and counterboring the cylinder, so as to make a male and

female joint; this shoulder will prevent the gasket from blowing out.

2. On the water cylinder the head with the stuffing box gives the most trouble by blowing gaskets, on account of an alternate increase and decrease of pressure on the gasket, due to the motion of the rod. This will be aggravated if the stuffing box is unduly tight. A very reliable joint for this end can be made by fitting the shoulder as above described and then using a copper gasket cut out in one piece and put on by disconnecting the rod at the crosshead.

3. In order to keep gaskets from blowing out through the forcing bolt holes, the forcing bolts should be kept in place.

9470.67 SCORES IN STEAM CYLINDER

Relatively small scores in a steam cylinder necessitate a reboring on account of the cutting action of steam. Leakage past the steam piston is readily located by the dullness and discoloration of the cylinder walls. Having once started, through the medium of a few scores, steam will gradually cut away cylinder walls until leakage by the piston becomes so excessive as to cause faulty operation of the pump. The spring in steam piston rings will take up some of the wear in the steam cylinder, thereby permitting some wearing away before reboring the cylinder is necessary.

9470.68 THE ALINEMENT OF PUMPS

1. Pumps must be kept in line, as failure to keep proper alinement is one of the greatest sources of trouble with pumps aboard ship and is one of the most serious defects. Pumps secured to a bulkhead are more likely to get out of alinement than those with an independent base and setting. A pump may have been well alined in the shop and then pulled out of line when it is bolted to the bulkhead, or, after installation, the bulkhead may have distorted sufficiently to cause misalinement. A pump run in this condition usually scores the rod and cylinders and breaks follower and bolts. The alinement of pumps should be tested occasionally by removing the pistons and rods and running a line through the cylinders; especially should this be done during the first year of a new ship in commission and also if a pump is giving trouble by scoring the rod or cylinders or breaking followers.

2. It is found in some cases that when steam cylinder foundation pad bolts are slacked off, the cylinder pad pulls away from the foundation as much as one-half inch, indicating settling of foundations and bulkheads. This is corrected by fitting shims between foundation and pump.

3. An easy method for a rough check of alinement is to pull the steam and liquid end rod packing, and check clearances at four points 90 degrees apart between piston rods and cylinder heads as soon as practical. Alinement should be checked at three positions—top, center, and bottom of the stroke. If throat bushings are not worn (out of round) and clearances are not uniform, realinement of pump should be undertaken. Misalinement may also be due to lack of provision for expansion of the pump with respect to foundations. This can be corrected by longating the foundation bolt holes in the steam cylinder foundation pad, or using foundation bolts one-eighth inch smaller in diameter than the holes.

4. A frequent cause of misalinement (and often not recognized) is the piping. The piping should line up naturally with the pump connections and should not be forced into position. Provision must be made in steam and exhaust lines to take care of expansion. All piping should be supported independently of the pump. Incorrectly installed piping puts a strain on the pump and forces it out of alinement. Special care should be taken in the installation of suction piping to prevent vapor pockets, particularly on fuel oil pumps.

9470.69 RUNNING A LINE

To run a line through the cylinders to facilitate the work of realinement, fasten one end of the line on a finger piece secured to the bottom of the water cylinder and the other on a temporary beam rigged above the steam cylinder; then center the line at the bottom and top of the water cylinder so that it becomes the axis of this cylinder. The steam cylinder can now be moved about and centered on the line without disturbing the centering of the line on the water cylinder.

9470.70 FITTING METAL RINGS

1. Troubles in the steam cylinder result chiefly from the rings. Leakage of steam past a ring dulls the rubbing surface of a ring which, when tight, has a highly polished glazed finish. Diagonally cut split rings seem to give better service than overlapping rings. The latter generally break at the corner after being in service a short time. However, this trouble can be reduced by rounding the corners of the shoulders.

2. The ideal piston ring would be one which would prevent completely the leakage of steam without exerting lateral pressure against the cylinder wall. This, however, is impractical so split rings are used. The method of fitting them to the cylinder varies with the different designs of pumps. Ordinarily the designs of rings shown on approved drawings or manufacturer's instruction book of the pump should be followed in making up and fitting new rings. In the absence of drawings or with continued failure of rings, the practice of the home naval shipyard or repair ship should be followed.

3. The following procedure is one which will usually be found to give satisfactory results. In order that the ring may be under tension and still remain in close contact with the cylinder wall, even after a little wear, and at the same time not allow excessive leakage through the joint, a ring should be turned before splitting to a diameter that is 0.010 of an inch per inch of diameter larger than the cylinder bore. The ring is then diagonally split and an amount taken out which is approximately twice the amount of difference between the diameter of the rough turned ring and the cylinder bore. A liner consisting of a piece of chart paper for the larger size and thinner paper for the smaller size pumps is now placed in the gap made by splitting. The ring is secured on the face plate of a lathe and turned to the exact diameter.

9470.71 TIGHTNESS OF STEAM VALVES

Flat steam valves should be kept true by scraping and the packing rings of piston valves kept free in the grooves. When scraping flat steam slide valves, the strokes of the scraper should cross, so that the scraper will not chatter on the narrow bridges between the ports, and thus cause steam leakage when the valve is in operation. It is highly important that all the steam valves be kept tight, for, if the auxiliary valve leaks, the main valve will become steam bound; that is, the steam pressures in the various passages and ports of the auxiliary valve chest and in the main valve chest cylinders will equilize and the pump will stop.

9470.72 TO CHECK VALVE MEASUREMENTS

As a result of the frequent spotting in of the main and auxiliary valves, the relative size and arrangement of port openings may change owing to irregular coring. An easy method to check the accuracy of the valve action is to cut paper patterns of the valve and valve seat faces by laying the paper on the valve and peening with a hammer. Sliding the pattern of the valve over that of the seat will show the exact laps, leads, and port openings, which can then be checked with the drawings.

9470.73 VALVE CHEST STEAM CYLINDER

The same care must be exercised in fitting the valve chest steam cylinder, cylinder rings, and piston as is used in fitting the steam piston and cylinder.

9470.74 CARE OF MECHANISM

It is most important that all wear be kept out of the steam valve operating mechanism. Failure to do this will cause the pump to operate in a faulty manner and perhaps to stick. Rebush and renew the pins as often as necessary. If the wear of the bushings occurs rapidly despite careful lubrication, the holes may be bushed with tool steel bushings.

9470.75 REPAIRS TO VALVE CHESTS, LINERS, AND PISTON VALVES

1. When piston valve chest joints and lands and other parts are badly damaged by wear and steam cutting, the following repair procedures are authorized. Refer to figure 9470-12 for identification of the areas to be repaired.

2. Joint face "A" between steam chest and cylinder may be built up by overlaying with one of the following electrodes in accordance with Specification MIL-R-17131. Both surfaces should be brought back to their design dimensions.

3. Main liner chest lands "B" may be built up with 25 percent chrome 20 percent nickel corrosion resisting (austenitic type) steel welding electrode.

4. Liners should be replaced rather than subjected to extensive repair. However, if spare liners are not available, liner may be chrome plated on the outside area "C". In such event liners should be shrunk into place. Pressing chrome plated liners into the chest will result in galling. Chrome plating the inside or wearing surface of liners area "D" is not authorized as a repair as this would do away with the graphitic lubrication provided by the original special alloy iron liners.

5. Main and auxiliary valves should be replaced when worn or steam cut. Building up ring grooves "E" with chrominum cobalt composition is authorized if replacement valves cannot be obtained in time to effect repair. The original valve material has a hardness of approximately 200 Brinell compared to approximately 500 Brinell for the liner. This is designed so that wear will be taken on the pistons rather than on the more expensive liners. Thus to overlay the ring grooves with a harder material than the liner will result in rapid wear of the liner and is to be avoided.

6. The end lands of main piston valves may be found to have insufficient clearance with the liner. The clearance on diameter between end lands "F" and the liner should be between .002 and .0025 inches per inch of diameter. Center lands should have approximately half of this clearance.

7. Replacement piston rings "G" should be of cast iron as originally installed in order to obtain the required finish on working faces.

8. Tail rods "H" should be made of K monel metal when replaced in order to avoid corrosion and the danger of rust working loose in the valve chest. Chrome plating of the original carbon steel rods is satisfactory as a repair.

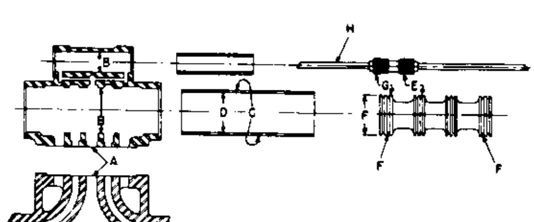


Figure 9470-12

Part 5. Safety Precautions

9470.81 SAFETY PRECAUTIONS

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

2. Boiler feed pumps shall not be used for purposes other than those connected with the service of the boilers or use of feed water, except in an emergency.

3. Before opening a steam cylinder or steam valve gear, see that drains are open and that steam and exhaust root valves are wired closed.

4. Before opening the water cylinder or valve chest of a pump handling water at a temperature in excess of 120° F., see that suction and discharge valves are wired closed, and cylinder and valve chests are drained.

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

SECTION II. CENTRIFUGAL PUMPS

Part 1. General

9470.101 INTRODUCTION

Centrifugal and axial flow pumps are now largely used for all nonviscous liquid services on naval ships. They have largely supplanted direct-acting steam pumps for such services. Advantages of centrifugal pumps include simplicity, compactness, weight saving, and adaptability to high-speed prime mover. However, centrifugal pumps have the disadvantages of lack of suction power and must be primed to operate on a suction lift. They are also sensitive to variations in head and speed. They are usually designed for a specific set of operating conditions and may give unsatisfactory performance when subjected to operation at great variance with the design rating. Hence it is important that the principle of operation, the suitable application, and limitations of the various types be thoroughly understood.

9470.102 DESCRIPTION

1. A centrifugal pump consists essentially of a hydraulic machine which imparts energy to the liquid through centrifugal force. This force is produced by the rotation of an impeller at high speed within a casing. Liquid is led to the "eye" or center of the impeller and is thrown to the outer periphery of the impeller by the centrifugal force imparted by the impeller vanes. The liquid leaves the impeller with high velocity and kinetic energy and is collected by the casing volute or diffusion vanes. These and the diffusing discharge nozzle are carefully designed to convert the high velocity kinetic energy to low velocity potential energy or pressure at the discharge flange of the pump.

2. The axial flow or propeller pump has a flow parallel to the shaft. The liquid has velocity imparted by the action of propeller vanes and this velocity is converted to pressure by a diffusion casing and diffusion vanes which straighten the flow and reduce its velocity. Correctly, axial flow or propeller pumps are not centrifugal pumps, but are not considered as a separate class because they represent the high capacity, low head extreme type in the whole pump family.

3. Intermediate types between the pure radial or straight vane centrifugal impeller and the axial flow include the "francis vane" and "mixed flow" pumps.

4. Figures 9470-13 and 9470-14 show diagrammatic section through volute and diffusion type centrifugal pumps respectively.

9470.103 CLASSIFICATION AND TYPES

There are many methods of classifying centrifugal pumps, including:

1. **Service application.** (See article 9470.104.)
2. Number of stages.
   a. Single stage.
   b. Two stage.
   c. Multistage.
3. Shaft position.
   a. Horizontal.
   b. Vertical.
   c. Inclined.

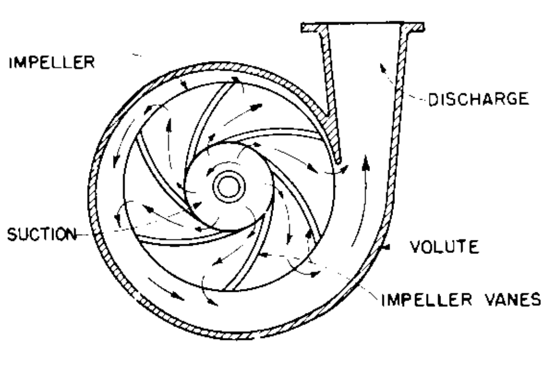


Figure 9470-13

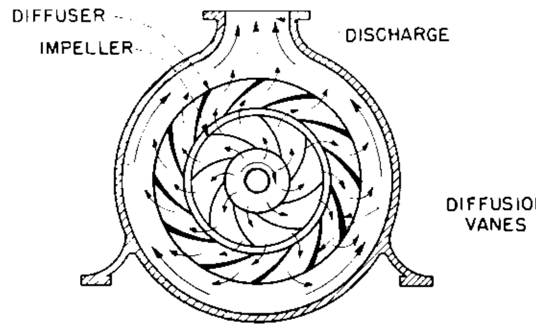


Figure 9470-14

4. Type of impeller.
   a. Single suction or double suction.
   b. Open, semiclosed, or closed.
   c. Radial, francis, mixed flow, or axial flow (propeller).
5. Type of casing.
   a. Volute, diffuser, or turbine.
   b. Solid, split, or barrel.
6. **Nozzle arrangement.** Side, end, top, or bottom and identified to both suction and discharge nozzles.
7. **Materials.** All iron, bronze-fitted, all-bronze, steel, stainless steel, titanium and other special alloys.

NOTE: The first two material classes are seldom used in naval shipboard service due to their inherent lack of shockproofness and accelerated corrosion when handling sea water.

8. **Type of drive.** Flexible coupled, rigid coupled, close coupled, gear or belt driven.
9. Other classifications such as special construction, direction of rotation (or reversible) kind or power drive, etc.
10. Figures 9470-15, 9470-16, 9470-17, 9470-18, 9470-19, and 9470-20 illustrate the several types of impellers mentioned herein.

9470.104 SERVICE CLASSIFICATION

The various pumps used in naval ships have requirement and problems of operation and installation peculiar to each application. The more distinctive types are listed below according to application. As types of naval ships vary greatly, not all of these services will be found on all ships; also special machinery plants may require pumps not listed herein:

1. Boiler feed pumps.
   a. Main.
   b. Auxiliary.
   c. Port.
   d. Cruising.
2. Feed booster pumps.
   a. Main.
   b. Auxiliary.
   c. Cruising.
3. Condensate pumps.
   a. Main.
   b. Cruising.
   c. Auxiliary (or dynamo).
   d. Distiller.

Figure 9470-15

Figure 9470-16

4. Circulating.
   a. Main.
   b. Auxiliary (or dynamo).
5. Fire pumps–including combination services such as fire and flushing, fire and emergency bilge.
6. Miscellaneous circulating and supply pumps
   a. Distiller condenser circulating.
   b. Refrigeration condenser circulating.
   c. Diesel engine cooling water.
   d. Auxiliary machinery cooling water.
   e. General service.
7. Fresh water pumps.
   a. Ship's fresh water.
   b. Fresh water pressure system.
8. Distilling plant pumps.
   a. Distiller circulating.
   b. Distiller condensate (distillate).
   c. Evaporator feed.
   d. Brine overboard.
   e. Distiller fresh water.
   f. Evaporator first effect coil drain.
9. Gasoline pumps.
   a. Main.
   b. Airplane fueling and defueling.
10. Lubricating oil.
    a. Main lubricating oil.
    b. Main lubricating oil standby.
11. Cargo.

a. Fuel oil.
b. Gasoline.
12. Portable pumps.
a. Electric submersible.
b. Gasoline engine driven.

9470.105 GENERAL CHARACTERISTICS

1. Unlike positive displacement pumps, a centrifugal pump at constant speed will deliver liquid at any capacity from zero to a maximum capacity which is limited by size of the pump, the suction condition, and its design factors. Over its range of capacity, the pressure, efficiency, and power required vary according to relationships inherent in the particular design. The relation of capacity, total head (pressure developed) efficiency, and horsepower required is customarily expressed by means of a characteristic curve. A characteristic curve generally contains the following three curves:

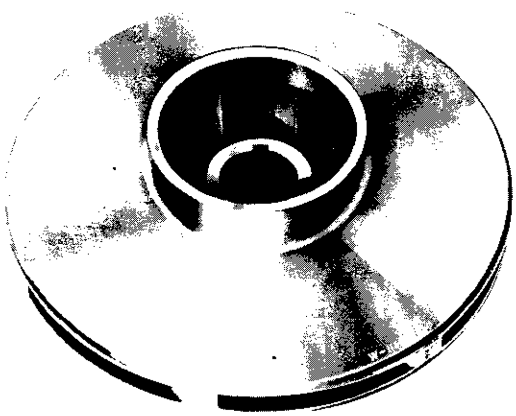


Figure 9470-17

Figure 9470-18

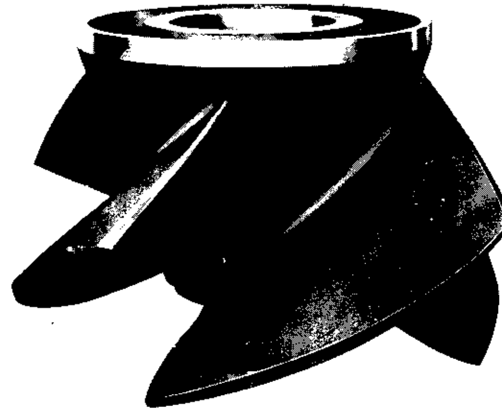


Figure 9470-19

Figure 9470-20

a. Head plotted against capacity.
b. Efficiency vs. capacity.
c. Brake horsepower vs. capacity.

All three curves are at constant rated speed. A typical characteristic curve is shown in figure 9470-21.

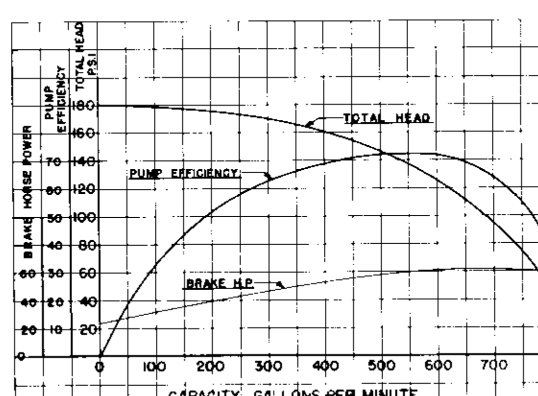


Figure 9470-21

2. In all centrifugal pumps the following laws of centrifugal apparatus hold true within practical operating limits of the pumps. For a given pump:

a. The capacity varies directly as the speed.

b. The total head varies as the square of the speed.

c. The horsepower varies as the cube of the speed.

3. It does not necessarily follow that when increased capacity is desired that it will require an increase in speed in direct ratio. This will depend on the pressure required. The performance acceptance test curves of the individual pumps should be consulted to determine the effect of changes in pump speed and pressure.

4. The following formula will be found useful for determining the approximate total head a centrifugal pump will develop:

a. $$H = f\left(\frac{DN}{1840}\right)^2$$

when H = total head in feet of liquid.
D = impeller diameter, inches.
N = pump revolutions per minute.
f = a speed constant.

f at shut off varies from 1 to 1.15

f at maximum efficiency varies from 1.10 to 0.75 and an average of 0.95 is accurate enough.

b. To convert H in feet to pressure psi, divide H by 2.31 for cool fresh water, or 2.24 for sea water.

c. For a multistage pump having equal impeller diameter, determine H for one stage and multiply by the number of stages, otherwise determine H for each stage and add the values of H.

d. Since the suction pressure is seldom zero, the discharge pressure to be expected should be computed from the total pressure by adding the positive suction pressure or by subtracting the suction lift or vacuum.

5. The power required to drive any centrifugal pump is determined as follows:

$$WHP = \frac{GPM \times H \times sg}{3960}$$

where WHP = water horsepower output.
GPM = capacity in gallons per minute.
H = total head in feet of liquid.
sg = specific gravity.

$$\text{or } WHP = \frac{GPM \times P}{1714}$$

when P = total pressure in psi the

$$BHP \text{ required} = \frac{WHP}{\text{pump efficiency}}$$

9470.106 SYSTEM CURVES

A system curve is a plot of the pressure characteristics of the system against variations in flow. For any given capacity the system pressure is the summation of friction losses, pressure in receiver minus pressure at the source, plus static pressure. Typical system curves for a condenser circulating pump (in which the losses are practically all friction) for a main condensate pump, and for a boiler feed pump are shown in figures 9470-22, 9470-23, and 9470-24. The point of intersection of the pump head capacity curve and the system curve determines the capacity and pressure at which the pump will operate.

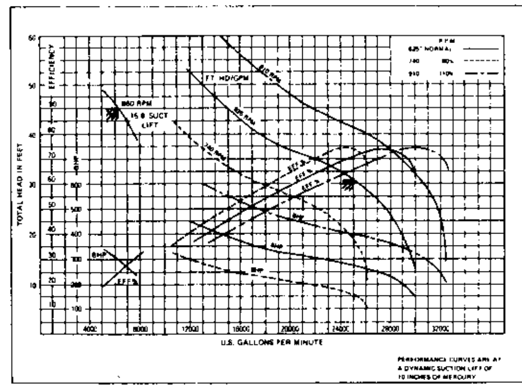


Figure 9470-22

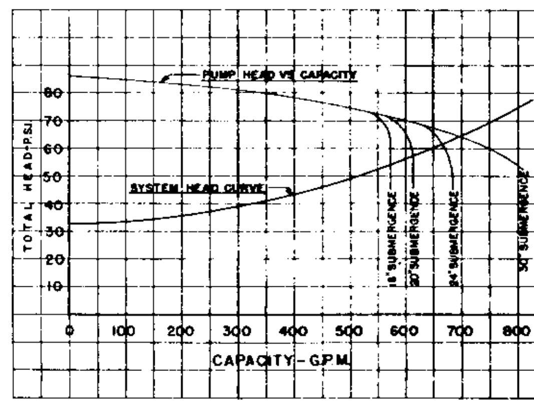


Figure 9470-23

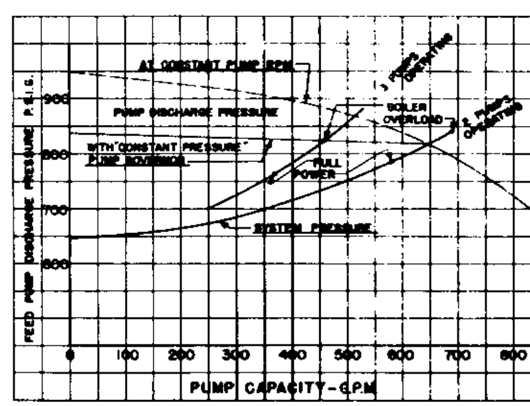


Figure 9470-24

9470.107 SUCTION LIFT

1. Suction lift exists when the total suction head is below atmospheric pressure. It is equal to the static lift plus the losses in the suction piping. When the liquid supply is above the pump center line, suction lift may result if the friction losses are greater than the static head.

2. The ability of a given pump to operate at suction lift depends on such factors as the viscosity, temperature and vapor pressure of the liquid and various design features of the pump itself including the suction passages, the impeller, and its speed. When a high suction lift is required, various means are available to the pump designer to obtain such performance. Liquid velocity in the impeller eye is a major factor but not an exclusive criteria. Pumps for high suction lift are necessarily larger and operate at lower speeds than pumps which can be given ample suction heads.

3. For any given pump the capacity and efficiency are not affected until the suction lift exceeds a certain value. When this value is exceeded, the pump capacity and efficiency drop, the pump will cavitate, and in severe cases may lose prime and seize. Figure 9470-25 shows how suction lift affects performance. This curve is for a fire and bilge pump.

9470.108 CAVITATION

Cavitation in a centrifugal pump occurs when the absolute pressure in the liquid at any point in the pump drops to or below its vapor pressure. When such a condition exists, vapor bubbles form and later collapse as soon as subjected to higher pressure at some other point in the flow. Cavitation in a pump may result from improper pump design, high suction lift, improper pump selection for the specific suction conditions to be encountered, and improper suction piping arrangement. Cavitation results in noise and vibration of the pump, reduction in maximum capacity and efficiency, and in pitting and corrosion of the pump parts, especially the impeller. Where cavitation cannot be avoided, it is necessary to provide special materials for the impellers and other parts to resist cavitation pitting. Naval condensate pumps, for example, operate nearly all of the time in cavitation and the relationship between capacity and submergence makes them selfregulating as to capacity. (See article 9470.149 and figure 9470-26.)

9470.109 NET POSITIVE SUCTION HEAD

"Net positive suction head" or NPSH is the suction pressure measured from absolute zero minus the vapor pressure expressed in feet of liquid under the actual pumping conditions. It represents the amount of energy available at the

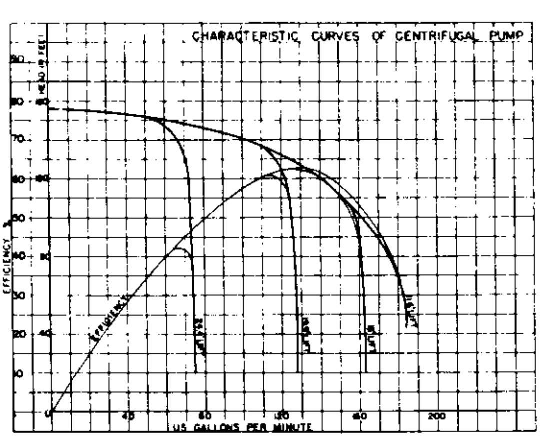


Figure 9470-25

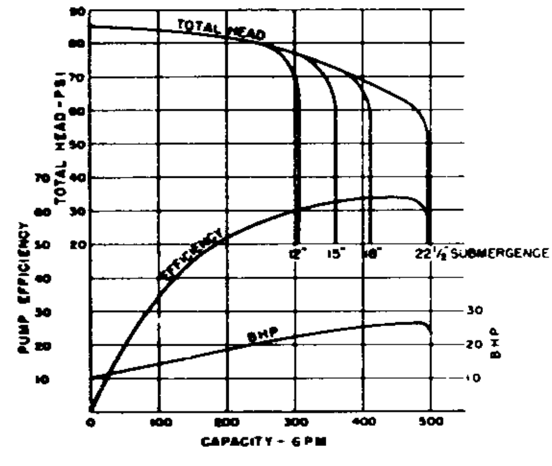


Figure 9470-26

pump suction to get liquid to flow into the pump. Two types of NPSH must be considered in selecting and installing pumps:

1. Available NPSH represents that of the system and is based on knowledge of the suction piping, location of pump with respect to the supply, pressure, temperature, etc.

2. Required NPSH is a condition of the pump design, usually as based on actual tests of nearly identical pumps, and indicates the minimum suction head minus the vapor pressure which is necessary to overcome inlet losses within the pump and ensure rated performance under satisfactory operating conditions. Both available and required NPSH vary with pump capacity.

9470.110 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 for a booster pump to be installed between the source of supply and the service pump. Examples of such installations on naval ships include fresh 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 the feed booster pumps installed between deaerating feed heaters and feed pumps. The main points to watch out for in such installations are:

1. The booster pump (acting as the first stage) should have identical or slightly greater capacity rating.

2. The maximum capacity of the first pump should equal or exceed the second.

3. The booster pump should always be started first and secured last.

9470.111 PARALLEL OPERATION

Pumps may be installed to operate in parallel when it is desired to have several smaller units to handle more efficiently varying capacity demands over a wide range, or when maximum reliability is required. Best examples are main and auxiliary condensate pumps, feed booster pumps, and boiler feed pumps. For satisfactory and stable operation in parallel, pumps should have constantly rising head from rated capacity to shutoff. Where pumps of different capacity ratings are to operate in parallel, as for example 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 essential in order 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.

9470.112 BOILER FEED PUMPS

1. Boiler feed pumps, in general, are of the high speed, multi-stage type, driven by steam turbines. Most pumps are horizontal; however, vertical steam turbine and motor driven pumps are used where space is limited. Figure 9470-27 shows a typical destroyer feed pump.

2. Chapter 9560 describes the different types of feed systems in general use. The pressure closed system, figure 9560-10, is installed on most ships of the active fleet. Feed pumps for this system handle water at 225° to 250° F. temperature and with suction pressures of approximately 50 psi gage. This is equivalent to an available NPSH of 110 to 85 feet respectively. The required NPSH is in the order of 20 feet. Suction pressure is usually supplied by a feed booster pump since the feed pumps operating at high speed would not have sufficient available NPSH. In some auxiliary ships the feed heater is located at a sufficient height above the feed pump to ensure adequate suction head without a feed booster pump. Boiler feed pumps require an NPSH of sufficiently high margin to take care of momentary or sudden increase of pump capacity.

3. Feed pumps require recirculation from discharge to source of supply, preferably the feed tank or deaerating heater, to protect the pump from flashing when operating at low capacity. See figure 9470-28 for effect of low capacity on temperature rise within the pump.

4. Modern feed systems involve the pumping of extremely pure unbuffered feed water of low pH values. In order to withstand the corrosive action of such feed water, boiler feed pumps are now constructed of corrosion resisting chrome or chrome nickel alloy steel.

9470.113 FEED BOOSTER PUMPS

1. Feed booster pumps are generally vertical, single or two-stage pumps driven by steam turbine or electric motor. Their construction is very similiar to condensate pumps. Feed booster pumps have low suction and impeller eye velocities and operate at low rpm because of the low NPSH available. Such pumps take suction from the deaerating feed tank under a submergence of 6 to 12 feet and this static head is practically the equivalent NPSH.

2. Feed booster pumps should be fitted with suction vent connections and with a recirculation line from discharge back to the feed heater. This line should be left open when starting or securing and when 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.

9470.114 CONDENSATE PUMPS

1. Main condensate pumps are vertical, one or two-stage pumps driven by steam turbine or electric motor. Condensate pumps take suction under extremely low submergence from the hot well of the condenser. First-stage impellers are usually single top suction type and the suction nozzle and piping are arranged to have a continuous downward slope from condenser to impeller eye. Every precaution is taken during the machinery design to ensure that the impeller is submerged even at maximum angle of ship's roll. Otherwise a severe water hammer may occur which has resulted in distortion of impeller shrouding between vanes. First stage impeller eye is vented back to the vapor space of the condenser.

2. Sealing water for condensate pump stuffing boxes should be taken from a remote source to ensure a continuous supply of clean water under adequate pressure to prevent leakage of air into stuffing box. Condensate pumps should not be operated at low capacities for very long without recirculation since their low margin of NPSH available over that required is insufficient to withstand temperature rise of more than a degree or two without flashing.

3. A typical two-stage condensate pump is shown in figure 9470-29.

9470.115 MAIN CONDENSER CIRCULATING PUMPS

Pumps for main circulating service on high-speed combatant ships are generally of the vertical propeller type driven by direct connected or geared steam turbine and are of capacities of 10,000–40,000 gallons per minute at total heads of approximately 30 feet. For such installations circulating pumps are usually arranged to operate when maneuvering, backing, or going ahead at low speed. At high speeds scoop circulation is adequate. For auxiliary ships, main circulating pumps are usually vertical, centrifugal, double-suction type driven by electric motor. Pump size is usually under 10,000 gpm.

9470.116 FIRE PUMPS

Fire pumps are usually horizontal, singlestage, double-suction type, either electric motor or steam turbine driven. Emergency fire pumps are usually diesel engine driven. Fire pumps are installed in numbers, locations, and sizes as necessary for best ship protection. Capacities may cover a range of 100–250 gpm for small ships, and up to 2,000 gpm for large ships.

9470.117 MISCELLANEOUS CIRCULATING AND SUPPLY PUMPS

1. The majority of pumps for such applications as flushing, auxiliary circulating, refrigerator circulating, distiller circulating, fresh water, etc., have capacities up to about 2,500 gpm at total heads under 100 psi. The horsepower seldom exceeds 25. Such pumps are usually of the single-inlet, centrifugal, volute type, close-coupled to the driving unit, which is usually 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 end bell or bearing bracket of the motor (figure 9470-30). The pump casing is fitted with a packing box in way of shaft and a cored type packing gland with a drip connection is usually provided. In addition the shaft is always fitted with a water flinger between the packing gland and the motor bearing bracket, the object of which is to prevent the water leakage from the packing box from following the shaft and entering the motor bearing and the motor housing. Frequently the pump casing is provided with supporting feet in addition to being bolted to the motor

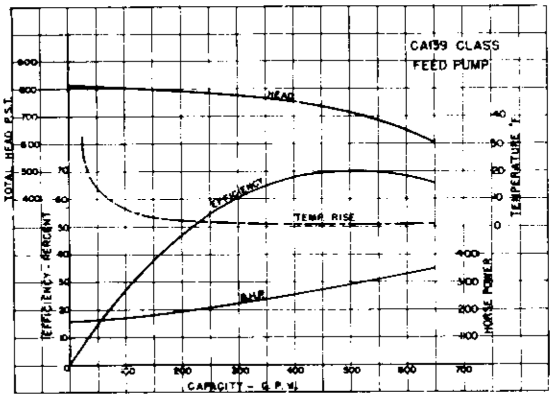


Figure 9470-28

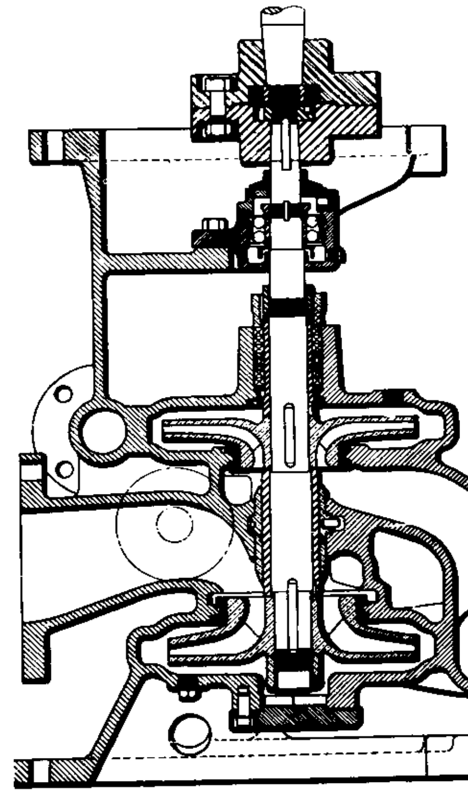


Figure 9470-29

although this construction is not essential where the shaft overhang beyond the motor is short.

2. In 1945, the Bureau of Ships introduced a line of Navy standard close-coupled pumps. At present this line includes 12 different size casings with motors to suit. The standard pumps are interchangeable as to size and location of suction and discharge connections and foundation bolting. Further motors are interchangeable as to shaft extension and bolting to pump casing so that motors of different manufacture are interchangeable on pump casings of the same size. The Navy standard close-coupled pumps will eventually be stocked in adequate quantities to permit their use as replacement for worn out nonstandard pumps. However, no large scale replacement program will be authorized except in instances where a specific unsatisfactory installation warrants general replacement.

9470.118 SHIP'S FRESH WATER PUMPS

Fresh water pumps are generally of the close-coupled type. They have a special feature of being automatically controlled by a pressure switch to maintain pressure in the fresh water system within predetermined limits. For this reason fresh water pumps should have a rather steep head-capacity curve so that pump is still delivering an appreciable flow as it builds up to the stop pressure.

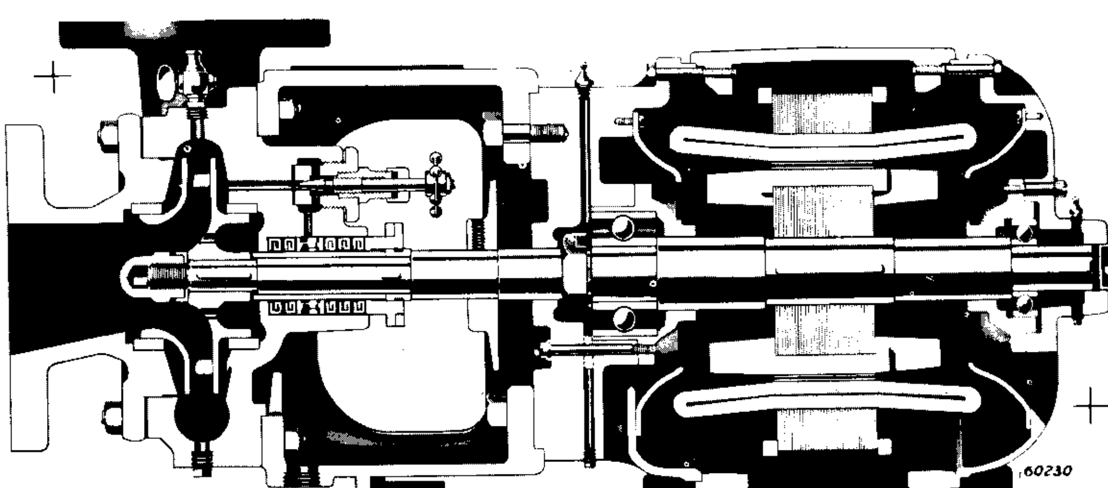


Figure 9470-30

9470.119 DISTILLING PLANT PUMPS

1. The following basic pump services are required for low pressure distilling plants:

a. Distiller circulating pump.
b. Distiller condensate pump.
c. Brine overboard pump.
d. Evaporator coil drain pump.

Additional services required for certain sizes and plant installations are:

e. Evaporator feed pumps.
f. Distiller fresh water pumps.

2. Pumps a., e., and f. above are usually horizontal close-coupled type. Pumps b., c., and d. are condensate type pumps and may be any one of the horizontal close-coupled, vertical-coupled, or vertical close-coupled types. Recent installations favor use of vertical pumps for distiller condensate, brine overboard, and coil drain service to ensure adequate suction submergence and to keep the motors out of unsatisfactory locations where they are subject to splash and drip.

3. Distiller condensate, brine overboard, and coil drain pumps should be either selfventing by installation of a suction pipe which rises continuously without pockets from the pump suction to the evaporator shell or condenser. Where this is impractical, a separate suction vent connection should be provided, led to a point on the condenser or evaporator shell well above the highest water line.

9470.120 LUBRICATING OIL AND CARGO OIL PUMPS

1. Lubricating oil or cargo oil pumps involve the pumping of viscous liquids which have the following effect on the performance of a centrifugal pump.

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

b. Efficiency is lowered and the horsepower for a given capacity is greater.

2. These changes in performance are primarily due to the increased disk friction losses of the impeller. Figure 9470-31 shows the effect of pumping oils of various viscosities on a main cargo pump. Nevertheless, centrifugal oil pumps have advantages in weight and space for the large pumping capacities now provided in modern tankers. Main lubricating oil pumps of the centrifugal type have been installed in several recent ships. Their viscosity range is usually between 130 and 500 SSU; therefore, the performance difference compared to water is not so marked. However, such pumps are required to handle large percentages of entrained air and special features are required to prevent difficulty due to air separation in the impeller eye or vane inlets.

9470.121 REGENERATIVE PUMPS

Regenerative pumps, also known as "peripheral" or "turbine" type, are occasionally used in naval service for applications involving low capacity and relatively high pressure. They are particularly useful on fresh water system in small ships since they have a steep head curve. The development of pressure does not follow the laws set forth in article 9470.105 and the flow through the pump is of a spiral path between the vanes on the impeller periphery and the

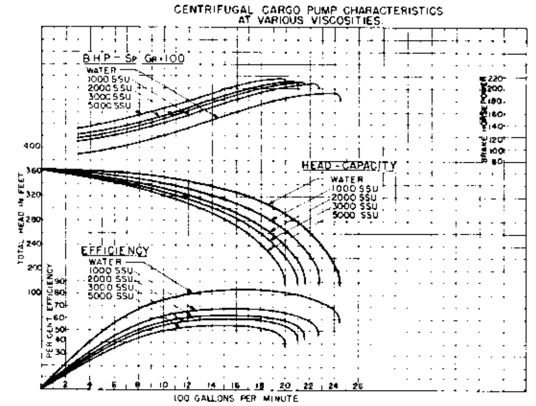


Figure 9470-31

casing raceway. Due to the close clearances only clean water services are recommended and their clearances must be carefully maintained. This type of pump is able to handle large percentage of vapors. Typical construction of a regenerative pump is shown in figure 9470-32

9470.122 PORTABLE SUBMERSIBLE PUMPS

1. Motor-driven, portable, submersible pumps of the centrifugal type are provided for all ships of the fleet and fleet auxiliaries and are intended primarily for damage control purposes, Some pumps of this type are stowed semipermanently, with suction and discharge hose connected, for use as drainage pumps. The number of these pumps allowed per ship is covered in each ship's allowance list.

2. Portable submersible pumps are furnished with either direct current or three phase alternating current motors as required. The pumps have the following minimum performance:

Capacity (gpm)	Total Head (feet)
140	70
180	50

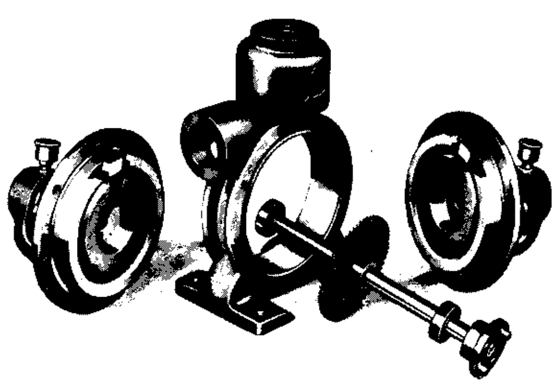


Figure 9470-32

The total heads include 20 feet suction lift to which must be added hose friction and vertical discharge lift to determine the suitability of the pump for the required operation. Where the total head required exceeds the head available in the pump, two pumps may be coupled for series operation.

3. Each pump is furnished with 75 feet of flexible cable having the controller installed in the cable 30 feet from the pump. Suction and discharge connections are arranged for Navy standard 2-1/2-inch fire hose. All pumps are designed to permit passage through a hole 10 inches in diameter. Each pump is provided with a suction strainer which should always be attached when the pump is in use. Each pump is also provided with a foot valve for attachment to the suction hose when the pump is to be operated above the water pumped. Priming is required when the pump is operated unsubmerged and this may be accomplished by lowering the pump into the water to start it, or by filling the pump casing and suction hose through the discharge hose connection. An eve is provided on the discharge end of the pump for attachment of a line for handling the pump.

9470.123 PORTABLE ENGINE-DRIVEN PUMPS

Gasoline engine-driven portable pumps classified as P-250: (250 gpm at 100 psi) are furnished to ships of the fleet and fleet auxiliaries. These pumps are dual service pumps which can be used as either a fire or drainage pump. As a drainage pump, they may be used in dewatering flooded spaces. Operation, care, and maintenance of these pumps are fully described in chapter 9930, Naval Ships Technical Manual.

Part 2. Operation

9470.131 USE OF MANUFACTURERS' INSTRUCTION BOOKS

The instructions contained in this chapter for the operation, maintenance, and repair of pumps are general. All pumps cannot be covered because of the great number of diverse makes and types and designs encountered in the naval service. For all but the simplest types of pumps, manufacturers' instruction books are furnished with the plans. These instruction books contain detailed information concerning the operation, maintenance, and repair of the specific pump installed and should be studied carefully before attempting to operate or service the unit. If any conflict exists between the instructions given in this chapter and the manufacturers' instructions, the Naval Ships Systems Command should be consulted.

9470.132 PREPARING PUMP FOR OPERATION

Prior to starting a centrifugal pump for the first time or after an overhaul, proceed as follows:

1. Check alinement at coupling and if necessary bring back into correct alinement. (See articles 9470.173 and 9470.185.) Before reassembling check direction of motor rotation to be sure it is correct as indicated by arrow affixed to pump casing.
2. Clean all external surfaces of the pump and driving unit to remove all dirt that may have accumulated during installation.
3. Check all auxiliary piping including lubricating lines, water cooling lines, vent lines, bypass piping, gage connections, priming connections, etc.
4. Examine all bearings to be sure they are properly lubricated.
5. Make sure stuffing boxes are properly packed.
6. Stuffing box glands should not be drawn too tight. A slight flow of water through the stuffing box is necessary for proper lubrication of the packing.

9470.133 STARTING

To start a centrifugal or axial flow (propeller) type pump, proceed as follows; subject to more detailed instructions which appear later in this section:

1. Check lubricating-oil level in pump tank or bearing housings. Fill oil cups or reservoirs, if fitted. If lubricated by a detached pump, open and adjust all oil delivery and return valves.
2. Open steam and exhaust casing drains of turbine or driving engine.
3. Open valves on pump to gland water seals, where fitted. Slack up on packing in pump-stuffing boxes to permit a reasonable amount of leakage.
4. Open steam and exhaust root valves.
5. If pump is driven by a reciprocating engine, jack it into position for starting. Remove starting bar.
6. Open turbine or engine exhaust valve.
7. Lift all relief and sentinel valves by hand.
8. Open pump suction valve.
9. Prime the pump. (See article 9470.141.)
10. For main feed pumps, and other pumps where fitted open recirculating (bypass) line valves and gland leak-off line valve.
11. Start unit.
    a. Crack turbine throttle valve sufficiently to free lines, steam chest, and exhaust casing of water, being careful that turbine runs, if at all, at very slow speed. As soon as turbine is free of water, close drain, then open throttle fast enough to bring unit up to speed gradually.
    b. For reciprocating engines, crack the throttle to allow cylinder to warm up slowly; then open throttle slowly. Sometimes it may be necessary to open the throttle smartly in order to start the engine, but if this is done, care must be taken to choke down as soon as engine starts, and to run slowly until engine is warmed up. Close drains when engine is clear of water.
    c. For motor-driven pump, start motor. (See instructions for operation of controllers and motors, chapter 9063.)
12. Open vent valves or air cocks on pump casing to check that all entrained air has escaped.
13. After all air has escaped, close vent valves or air cocks on pump casing.
14. For condensate pumps or other pumps vented back to a vacuum, open the valve in the vent connection.
15. When pump is up to proper speed, open the discharge valve. For axial flow (propeller) type pump, it is necessary to start with the discharge valve open to avoid either:
    a. Driving unit frequently too small to start pump against closed discharge, or
    b. Excessive discharge head may be developed.

16. Check lubrication system to see that all bearings are getting a proper supply of oil and that oil pressure and temperature are correct.

17. Check all gages to see that proper pressures are being developed.

18. For turbine-driven units, check revolutions per minute to see that rated speed is not exceeded by more than five per cent.

19. Set up on packing glands slowly until leakage is reduced to a mere trickle.

9470.134 STOPPING AND SECURING

To stop and secure, proceed as follows:

1. Close pump discharge valve.
2. Close steam throttle valve or stop motor.
3. Close exhaust valve.
4. Close vent valves.
5. For main feed pumps and others, where fitted, close recirculating (bypass) and gland leak off line valves.
6. Close valves on pump to gland water seals.
7. Close pump suction valve.
8. Open turbine or engine drains.
9. Close all delivery and return oil valves when pump is lubricated by a detached pump.
10. Close steam and exhaust root valves.
11. Close steam drains after turbine or engine is completely drained.

9470.135 CENTRIFUGAL PUMP TROUBLES

The following points should be checked in the event the pump does not operate normally or satisfactorily:

1. No liquid delivered.
   a. Pump improperly primed.
   b. Pump speed too low.
   c. Discharge head too high. Valves may be partly closed or other obstructions impose too high a system head.
   d. Suction lift too high. Unless pumps are specifically designed for high suction lift, the normal design is limited to approximately 15 feet of water suction lift when handling cold water.
   e. Impeller passages plugged up.
   f. Pump running opposite to correct rotation.
2. Not enough liquid delivered.
   a. Air leaks in suction pipe.
   b. Air leaks in stuffing boxes.
   c. Pump speed too low.
   d. Suction lift too high.
   e. Suction sea chest or connection to supply not submerged enough.
   f. Impeller passage partly clogged.
   g. Discharge head too high.
   h. Not enough suction pressure when pump handles hot or volatile liquids. (Indicated by noise and fluctuating pressure.)
   i. Mechanical defects including wearing rings worn, impeller damaged or eroded, stuffing box packing and/or sleeves worn and need replacing.
3. Not enough pressure.
   a. Pump speed too low.
   b. Air or gas in liquid pumped.
   c. Mecnanical defects as noted in 9470.135.2 above.
4. Pump works for a while and then fails to deliver.
   a. Air leakage in suction line.
   b. Air leakage in stuffing box.
   c. Stuffing box water seal plugged.
   d. Suction supply uncovered.
   e. Suction lift too high.
5. Pump takes too much power.
   a. Head lower than pump rating. This results in pump handling a capacity greater than rated and it will require more power (figure 9470-21).
   b. Pump speed too high.
   c. Specific gravity or viscosity of liquids other than water may be higher than pump designed to handle.
   d. Mechanical defects including rotor is binding, shaft bent, stuffing boxes too tight, wearing rings worn, misalinement.

9470.136 LUBRICATION

1. Lack of lubrication is one of the primary causes of pump failures. Pump bearings are lubricated by one of the following methods:
   a. Ring oiled sleeve bearings.
   b. Grease lubricated ball bearings.
   c. Oil lubricated ball bearings.
   d. Forced feed lubrication.
2. Check ring oiled bearings to see that oil rings are rotating freely and are carrying an adequate amount of oil. Check oil level frequently during each watch.
3. Reciprocating engine-driven pumps are usually lubricated by either sight feed drip cups or wick lubricators. See that oil cups are filled with oil and that adequate oil supply is being fed to bearings; check frequently during the watch.
4. Motor-driven pumps and some turbine-driven pumps fitted with ball bearings are usually fitted for grease lubrication. Before starting, see that all grease cups and bearing housings are filled with the proper amount of lubricant and that no-water or foreign matter is in the bearing housing. Grease lubrication is used for the two fold purpose of lubricating the bearing and excluding water and foreign matter from the bearing housing. Pump shafts are usually fitted with a water flinger between pump shaft stuffing box gland and bearing housing. See that such flingers are effectively preventing water from the pump glands from following along the shaft and entering the bearing housing. Occasionally it will be found that sleeves fitted on pump shafts do not fit the shaft tightly and water can leak under the shaft sleeves. If such leakage exists, care should be taken to prevent water from entering the bearing housing.
5. Turbine-driven pumps are usually fitted with a self-contained force lubricating-oil system supplied by an attached lubricating-oil pump of the gear or screw type.
6. Check the oil pressure and oil flow to or from all bearings, including the thrust bearing, being sure that the attached lubricating-oil pump has primed itself. See that cooling water is flowing through the oil cooler or cooling coils in the oil reservoir and that all air is vented from the water side of the oil cooler. It may be necessary to free the lubricating-oil system of air in order to maintain a steady oil pressure. This can be accomplished by opening air cocks on the high points of the lubricating-oil system. These should be closed immediately when oil appears. Check oil reservoir to see that it is free of water. Check bearing housings to see that no water from pump or turbine glands is getting into lubricating-oil system.