PUMP QUESTIONS AND ANSWERS

Covering the Construction, Application, Operation, Installation, Maintenance, and Troubles of Centrifugal, Reciprocating, Regenerative, Rotary, and Vertical Turbine Pumps

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To

Albert H. Borchardt, Assistant Vice-President
Worthington Pump and Machinery Corporation

PREFACE

"Question 1-1: What is meant by pumping?

"*Answer:* Pumping is the addition of energy to a fluid to move it from one point to another and not, as is frequently thought, the addition of pressure. Since energy is capacity to do work, the addition of energy to a fluid causes it to do work, such as flowing through a pipe or rising to a higher level."

This is the first question presented in the first chapter of this book. It provides the clue to the reasons that motivated the authors to prepare the manuscript of the book and the publishers to print and distribute it. Just as water, in the form of oceans, seas, lakes, ponds, and rivers, covers the major portion of the surface of our globe, so does the transportation of fluids from one point to another play a major part in modern civilization. The claim that every single industrial process that underlies this modern civilization involves the transportation of fluids is hardly subject to challenge. Therefore a pump, which is the mechanical means of achieving this transportation, must be a piece of machinery that is an essential part of our everyday life.

The choice of the question-and-answer form for this book was made in the belief that it is the most compact means of covering the subject to be treated and that it does away with a great deal of transition material that is used merely to give cohesion to the text. The contents of this book appeared first on the pages of the magazine *Power* in the form of articles that ran consecutively from March, 1945, to March, 1948, inclusive. They were prepared under the active guidance and with the cooperation of F. A. Annett, Associate Editor of *Power*, who edited these articles and to whom the authors are sincerely indebted.

The book was written by a group of men, each of whom contributed material pertaining to that type of pumping equipment in which he has specialized for a number of years and, therefore, with which he is intimately familiar. It is made up of five major sections, dealing respectively with centrifugal pumps, vertical turbine pumps, regenerative pumps, rotary pumps, and reciprocating pumps.

The section on centrifugal pumps opens with a description of these pumps and of their parts, as well as of the various classifications under which they are generally subdivided, and treats of the materials used in the construction of centrifugal pumps. The discussion of centrifugal-pump and system characteristics is followed by an analysis of the operating characteristics and of the application of these pumps. The installation, operation, and maintenance of centrifugal pumps are given very complete treatment. There is presented a complete discussion of centrifugal pump troubles, their symptoms, and their causes. A final chapter on centrifugal pumps treats of the very important subject of priming.

The sections dealing with vertical turbine pumps, regenerative pumps, and rotary pumps follow this same general outline.

In the section dealing with reciprocating pumps, two chapters are dedicated to the construction of simplex and duplex direct-acting pumps and of horizontal and vertical power pumps, respectively. They are followed by a chapter describing the characteristics of steam and power pumps. Because of the importance played by the design and operation of the reciprocating-pump liquid valves, an entire chapter has been devoted to this subject. Similarly, a chapter has been given to the description of rod, piston, and plunger packing, and another to the analysis of cushion chambers. This section ends with a chapter on the installation and operation of reciprocating pumps and another on remedies for steam- and power-pump troubles.

Shakespeare said, "An honest tale speeds best being plainly told." The authors attempted to follow this advice to the best of their abilities in the preparation of this book. They felt that a clear, concise presentation, in simple terms, avoiding pedantic complications of language that serve only to confuse the reader, would make this book more useful to a greater number of people—and this, after all, should be the aim of every book. To what degree they have succeeded in this attempt, they will have to leave to the judgment of the readers.

THE AUTHORS

HARRISON, N.J.
July, 1949

CONTENTS

CHAPTER 1

CENTRIFUGAL PUMPS AND THEIR PARTS

Question 1-1: What is meant by pumping?

Answer: Pumping is the addition of energy to a fluid to move it from one point to another and not, as is frequently thought, the addition of pressure. Since energy is capacity to do work, the addition of energy to a fluid causes it to do work, such as flowing through a pipe or rising to a higher level. In the closely allied field of centrifugal compressors, "pumping" describes an unstable surging that occurs in many compressors at partial capacity.

Question 1-2: What is a centrifugal pump?

Answer: It is one that employs centrifugal force for pumping liquids. Liquid coming in at the center (eye) of the impeller (Fig.

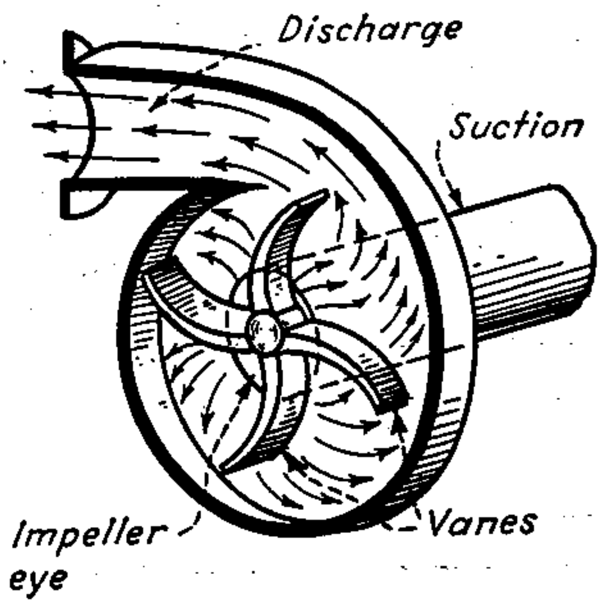


FIG. 1-1. Diagram showing the operation of a centrifugal pump.

1-1) is picked up by the vanes and accelerated to a high velocity by the rotation of the impeller and thrown out by centrifugal force into an annular channel, or volute, and to the discharge, as indicated.

Question 1-3: What are the essential structural elements of any centrifugal pump?

Answer: They are the rotating element, comprising basically a shaft and impeller, and the stationary element, made up of casing, stuffing boxes, and bearings (Fig. 1-2). All other parts are refinements of construction, supplementing the function of the main elements.

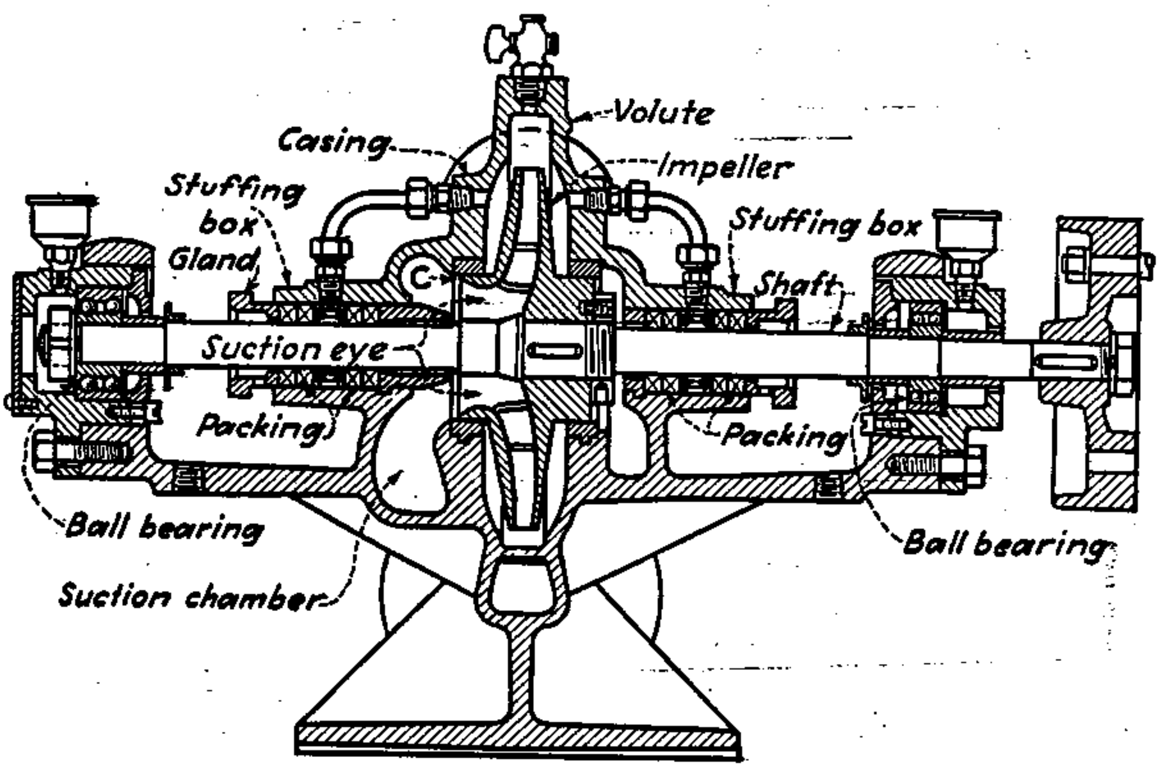


FIG. 1-2. Axial cross section of a single-stage, single-suction, split-case, side-suction, side-discharge centrifugal pump.

Question 1-4: How are centrifugal pumps classified?

Answer: They are classified according to:

1. Type of energy conversion: (*a*) volute, (*b*) diffuser or turbine.
2. Number of stages: (*a*) single-stage, (*b*) multistage.
3. Impeller type: (*a*) single- or double-suction, (*b*) open, semi-open, or closed, (*c*) single- or double-curvature vanes.
4. Axis of rotation: (*a*) horizontal, (*b*) vertical, (*c*) inclined.
5. Casing: (*a*) solid or split, (*b*) location of suction and discharge nozzles.

6. Method of drive: (*a*) direct-connected (coupled or close-coupled), (*b*) geared, (*c*) chain-driven or belted.

There are other classifications, such as classes of service, features of construction, and rotation.

Question 1-5: What is a volute-type centrifugal pump?

Answer: A pump named from its spiral-volute form of casing (Fig. 1-3), which acts as a collector for the fluid discharged by the

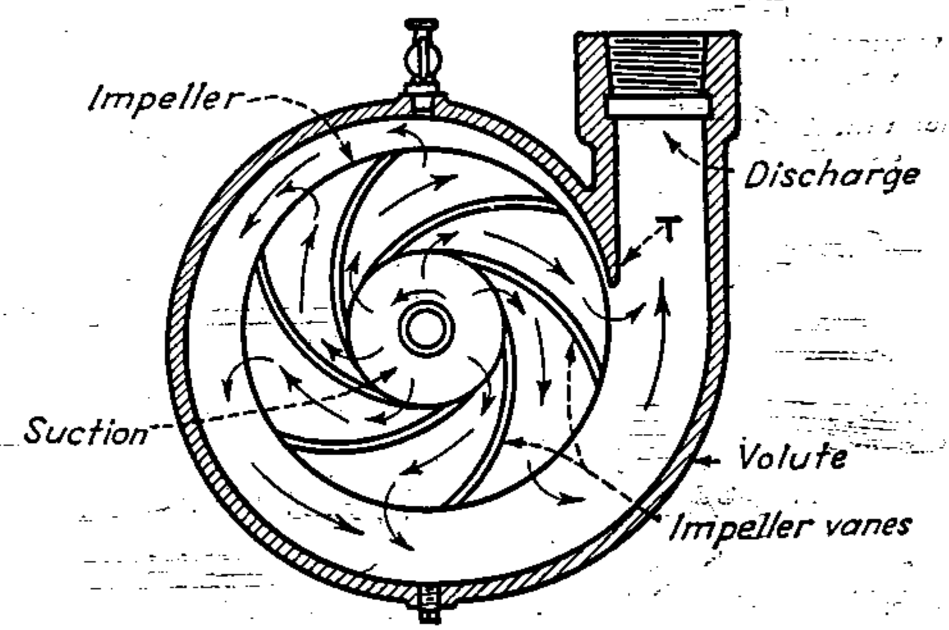


FIG. 1-3. Radial diagrammatic section through a volute-type centrifugal pump.

impeller. Wall *T*, dividing the beginning of the volute and the discharge nozzle, is called the "volute tongue" or "cut water."

Question 1-6: What is a diffuser-type centrifugal pump?

Answer: In this pump diffusion vanes (Fig. 1-4) convert the velocity energy imparted to the liquid by the impeller into pressure energy. The diffusion-vane assembly is known as a "diffuser."

Question 1-7: What is a turbine-type pump?

Answer: The name "turbine pump" has been applied to two different designs. One, a centrifugal design, is now generally called a "diffuser type" (Fig. 1-4), except for the "vertical turbine pumps," formerly called "deep well turbine pumps," which are

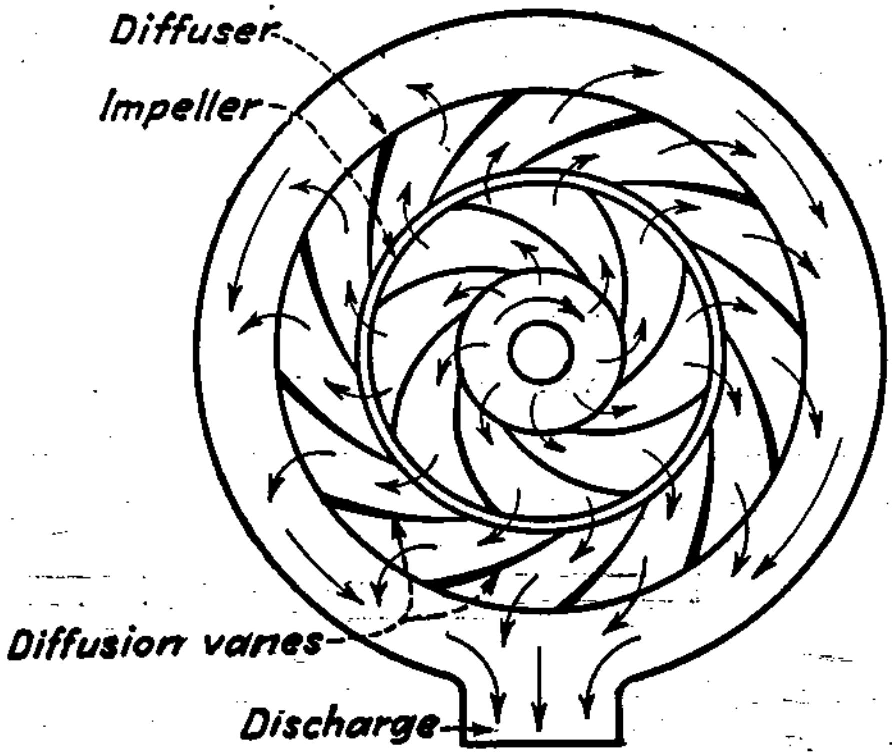
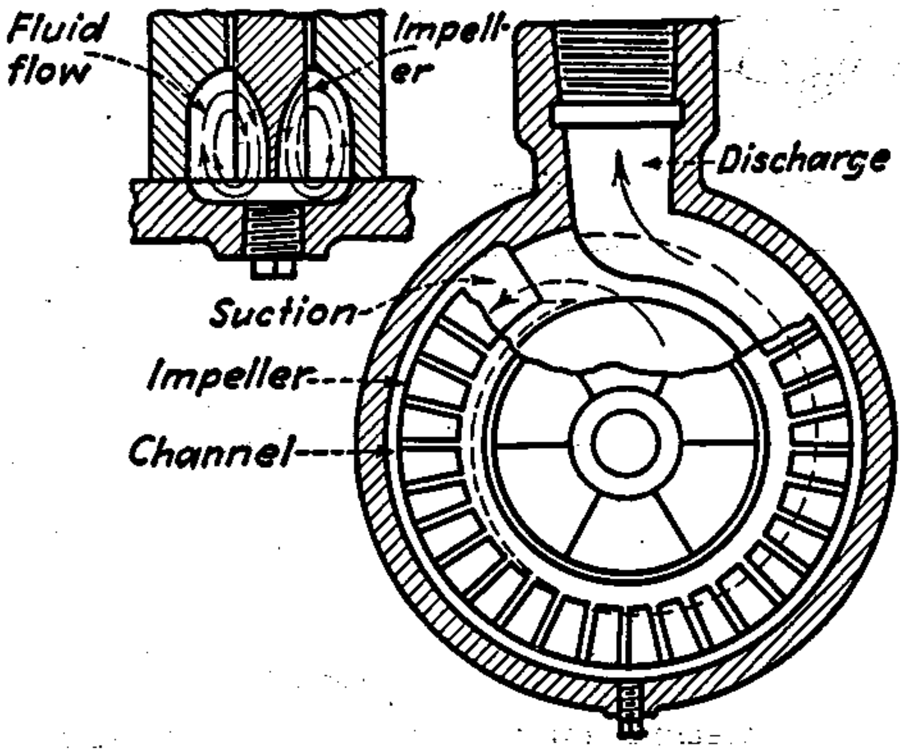


FIG. 1-4. Radial diagrammatic section through a diffuser-type centrifugal pump.



FIGS. 1-5 and 1-6.

FIG. 1-5. Axial section of impeller periphery and waterway of a regenerative or so-called turbine-type pump showing how liquid is recirculated through the impeller vanes as it travels from suction to discharge.

FIG. 1-6. Radial section of a regenerative pump showing circular passage of liquid from suction to discharge.

separately discussed in Chap. 10. The second so-called turbine pump is not actually a centrifugal type. It is now generally called a "regenerative" type but has also been called "vortex" and "turbulence." In this type of pump the impeller acts on the liquid on its periphery for almost a full revolution (Fig. 1-6) with the liquid being circulated in and out of the impeller vanes (Fig. 1-5) so the true path of the liquid is a circular spiral. This regenerative pump is discussed in Chap. 11.

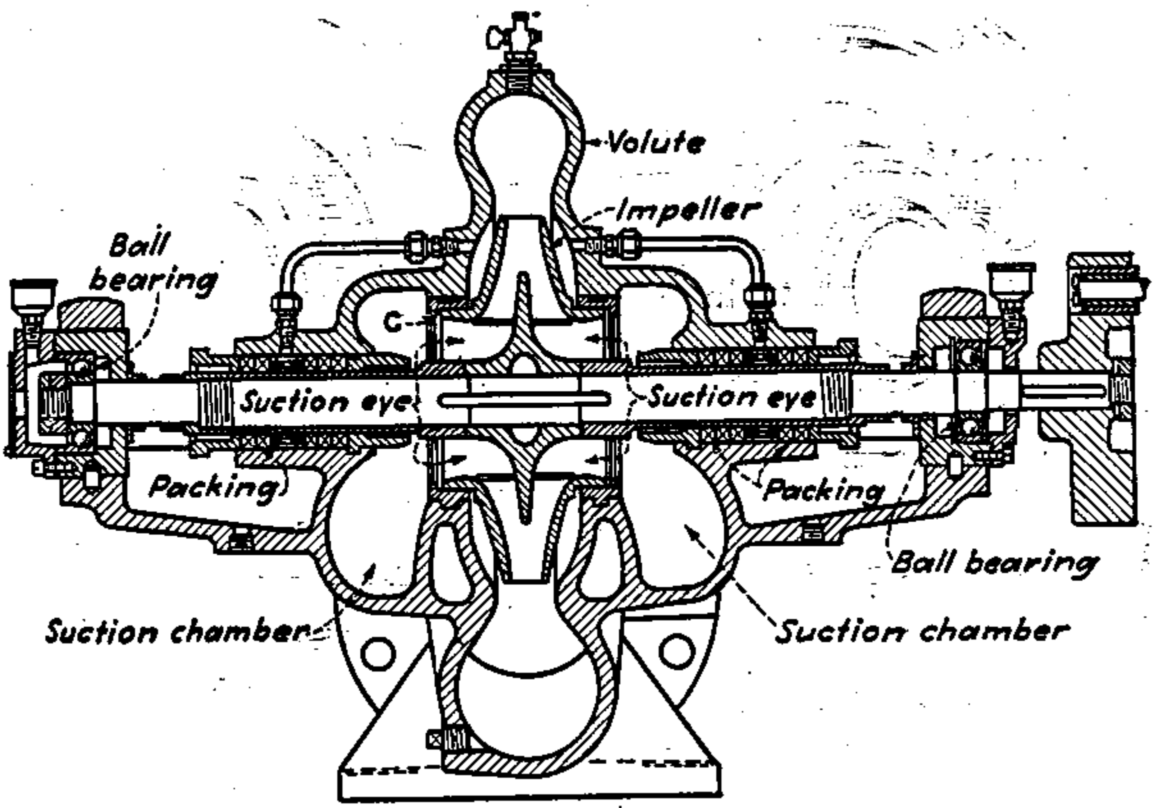


FIG. 1-7. Axial cross section of a single-stage, double-suction, split-case, side-suction, side-discharge centrifugal pump.

Question 1-8: What is a single-stage pump?

Answer: It is one in which the total head is developed by a single impeller (Figs. 1-2 and 1-7).

Question 1-9: What is a multistage pump?

Answer: If the total head required is too high for a single impeller to produce, two or more impellers (or pumps) may be used in series, the second impeller taking its suction from the discharge of the first impeller (Fig. 1-8). If all impellers acting in series are in a single casing, the pump is a multistage design.

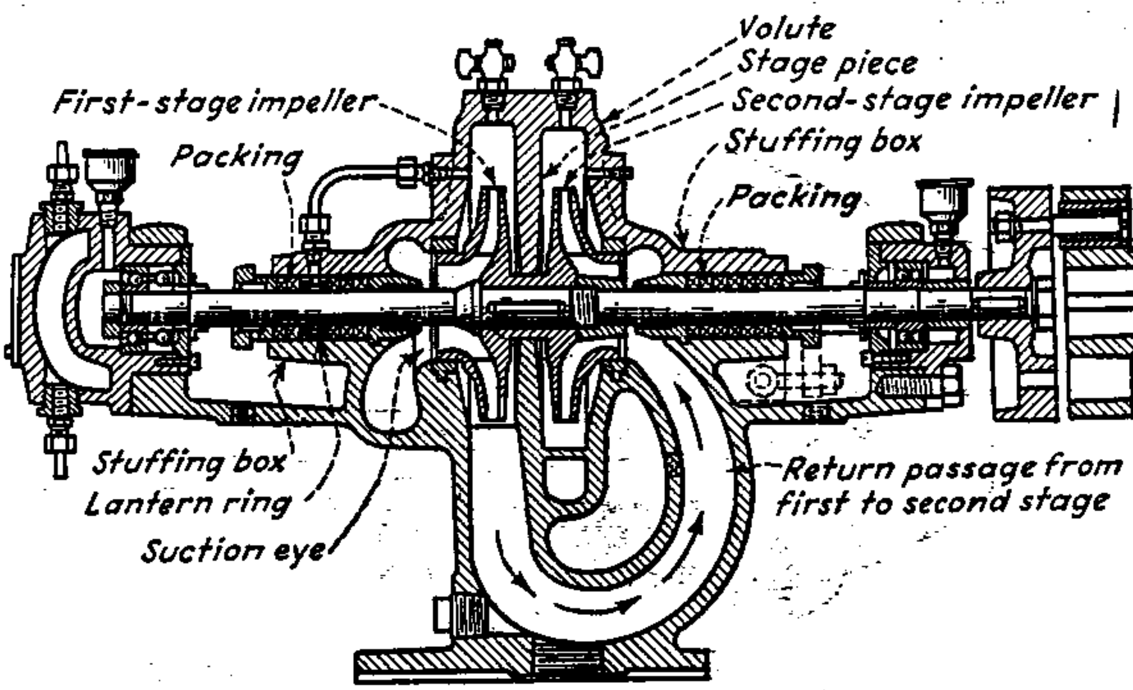


FIG. 1-8. Two-stage pump having single-suction impellers facing opposite directions.

Question 1-10: What is a bi-rotor pump?

Answer: Bi-rotor and even tri-rotor designs were fairly common in the early development of centrifugal pumps. These pumps were, in effect, two one-half- or three one-third-capacity pumps in one casing, operating in parallel.

Question 1-11: What is a single-suction impeller?

Answer: It is one in which the liquid pumped enters the impeller from one side (Fig. 1-2).

Question 1-12: What is a double-suction impeller?

Answer: It is, in effect, two single-suction impellers cast back-to-back so that liquid enters the impeller simultaneously from both sides (Fig. 1-7). Normally, the suction chambers, one at each side of the impeller, connect to a common suction passage and nozzle. Some large pumps have two separate suction passages and nozzles connected to opposite sides of a double-suction impeller. This construction was used in the early development of double-suction pumps for units quite small in size.

Question 1-13: What are the respective advantages of double-suction and single-suction impellers?

Answer: In general service, a double-suction impeller is favored in single-stage, horizontally split casing designs. It is theoretically

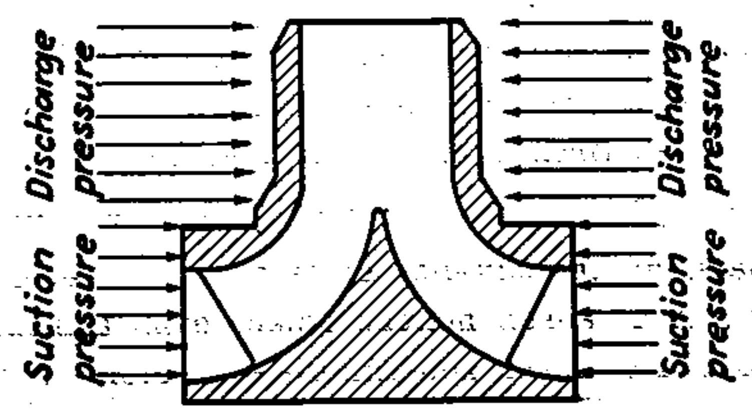


FIG. 1-9. Axial hydraulic forces on both sides of a double-suction impeller equalize each other.

in axial hydraulic balance as indicated in Fig. 1-9, making an oversized thrust bearing unnecessary. The greater suction area in a double-suction impeller permits the pump to operate with less net absolute suction head for a given capacity than with a single-suction impeller. For manufacturing reasons, the single-suction impeller is more practical than the double-suction in small pumps because the waterways are double size and not divided into two; but they require a larger thrust bearing to take the unbalanced axial thrust (Fig. 1-10), or balancing ports.

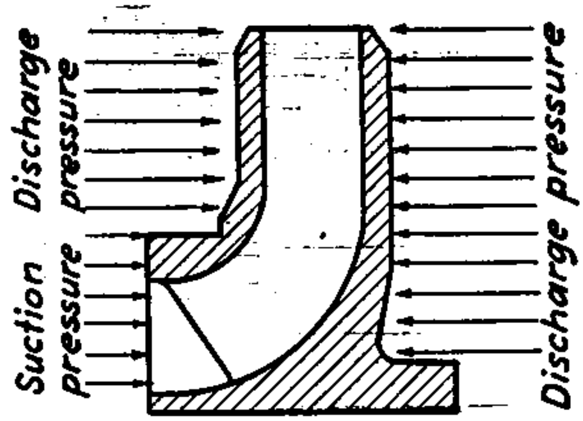


FIG. 1-10. Axial hydraulic forces on the two sides of a single-suction impeller without back wearing ring and balance holes are unequal, resulting in an axial thrust.

End-suction pumps with single-suction overhung impellers (Fig. 1-15) have advantages that cannot be obtained with a double-suction impeller, and thus, in general, pumps with radially split casings have single-suction impellers. Because with an overhung impeller the shaft does not extend into the suction eye, single-suction impellers are almost always used for pumps handling liquids containing suspended matter, such as sewage.

Question 1-14: What is an open impeller?

Answer: Strictly speaking, an open impeller has vanes only, attached to a central hub without any form of side wall or shroud. With long vanes this design is structurally weak, and the vanes have to be strengthened by ribs or a partial shroud (Figs. 1-11 and

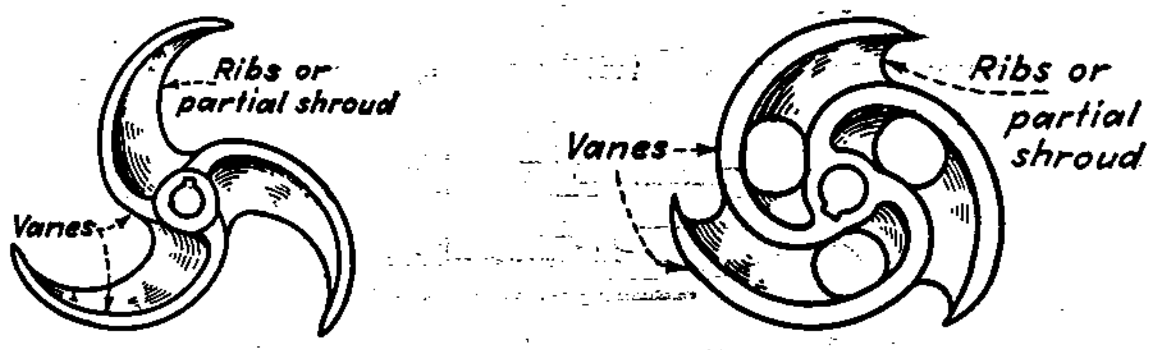


FIG. 1-11. FIG. 1-12.

FIGS. 1-11 and 1-12. Three-vane open impellers with ribs or partial shrouds to give strength.

1-12). Generally, open impellers are used in small, inexpensive pumps or in those handling abrasive liquids. The impeller rotates between side plates in the volute-casing walls or between the stuffing-box and suction-head covers (Fig. 1-15). Because of the clearance between impeller vanes and side walls, water slippage occurs, similar to that in a reciprocating pump, which increases with wear. Restoring the original efficiency requires replacing both the impeller and the side plates, which costs more than replacing sealing rings in a closed-impeller pump.

Question 1-15: What is a semiopen impeller?

Answer: One that has a shroud, or wall on one side only (Figs. 1-15 and 1-16). This shroud may or may not have pump-out vanes (Fig. 1-16) to reduce pressure on the stuffing box of single-suction pumps and to prevent solid material from lodging behind the impeller.

Question 1-16: What is a closed impeller?

Answer: This design, used almost universally in centrifugal pumps handling clear liquids, has shrouds on each side that totally enclose the waterways from the suction eye to the periphery of the impeller (Fig. 1-13). Water slippage does not occur in this design as with an open or semiopen impeller, but a running joint is needed between the impeller and the casing to separate the pump discharge

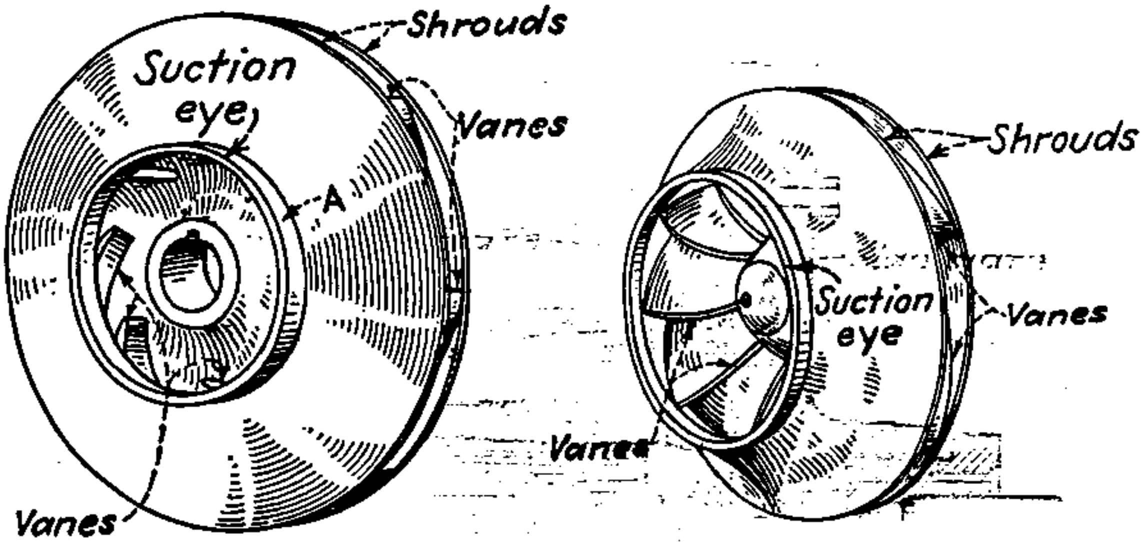


FIG. 1-13. FIG. 1-14.

FIG. 1-13. Closed impeller with straight (single-curvature) vanes.
FIG. 1-14. Closed impeller with francis screw (double-curvature) vanes.

and suction chambers. This running joint, generally formed by a relatively short cylindrical surface on the impeller shroud, as at *A* (Fig. 1-13), rotates within a slightly larger stationary cylindrical surface *C* in the pump casing (Figs. 1-2 and 1-7).

Question 1-17: What is a straight-vane impeller?

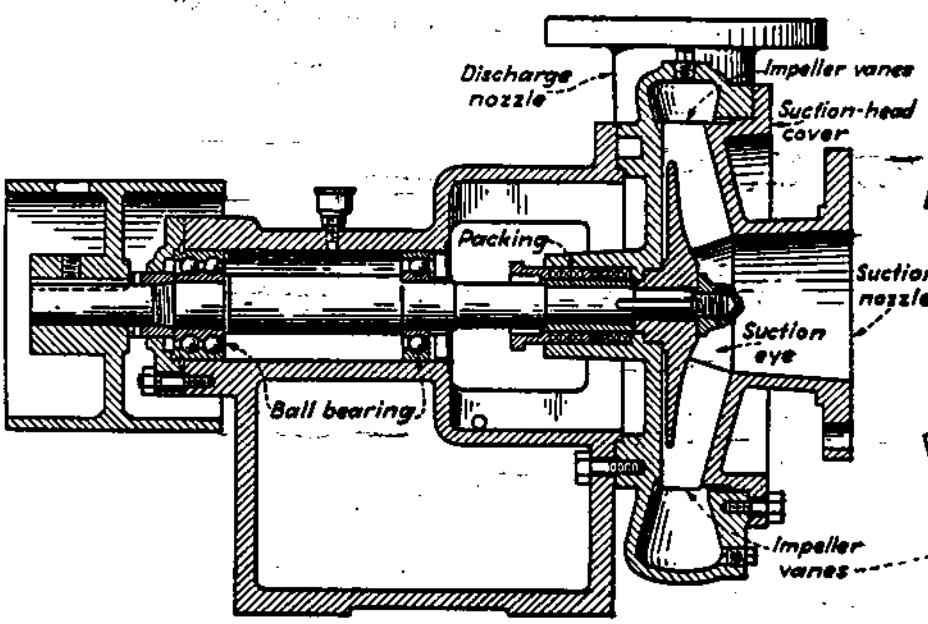
Answer: It is one in which the vane surfaces are generated by straight lines parallel to the axis of rotation. Vanes of this design are also called "single-curvature vanes" (Figs. 1-11, 1-12, and 1-13).

Question 1-18: What is a francis-vane impeller?

Answer: Its vane surfaces have a double curvature, and it is often called a "francis-screw-vane" or "screw-vane" impeller (Fig. 1-14).

Question 1-19: What is a mixed-flow impeller?

Answer: An impeller in which the flow is both radial and axial. It is generally restricted to single-suction designs having a specific speed above 4200. Lower specific speed types are called "francis-screw-vane." Mixed-flow impellers having a small radial-flow component are generally classed as propellers. The definitions of francis-screw-vane impellers, mixed-flow impellers, and propeller have never been rigidly fixed. Thus, what one person may call a



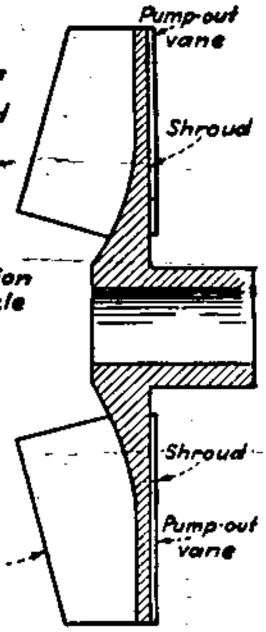


FIG. 1-15. FIG. 1-16.

FIG. 1-15. Axial section of an end-suction type of pump with semiopen impeller.
FIG. 1-16. Section of semiopen impeller with pump-out vanes on back of shroud.

high-specific-speed, mixed-flow impeller, another person may call a propeller type. Likewise, a low-specific-speed, mixed-flow impeller might be called, equally as accurately, a francis-screw-vane impeller.

Question 1-20: What is a propeller or axial-flow impeller?

Answer: A true propeller or axial-flow impeller is one having a flow solely parallel to the axis of rotation, in other words, strictly axial (Fig. 1-17).

Question 1-21: What is a non-clogging impeller?

Answer: Usually impellers of centrifugal pumps have quite sharp suction-vane edges and there are four or more vanes, six to eight being quite common. Only relatively small solids could pass through the waterways of such impellers and if rags or stringy materials reached the impeller, they would catch on the sharp suction-vane edges. Special non-clogging impeller designs with blunt suction-vane edges and with large passageways between the vanes are therefore used in pumps handling liquids, like sewage, containing rags, stringy material, and solids. For pumps up to 12- to 16-in. discharge, these impellers have only two vanes; larger sizes

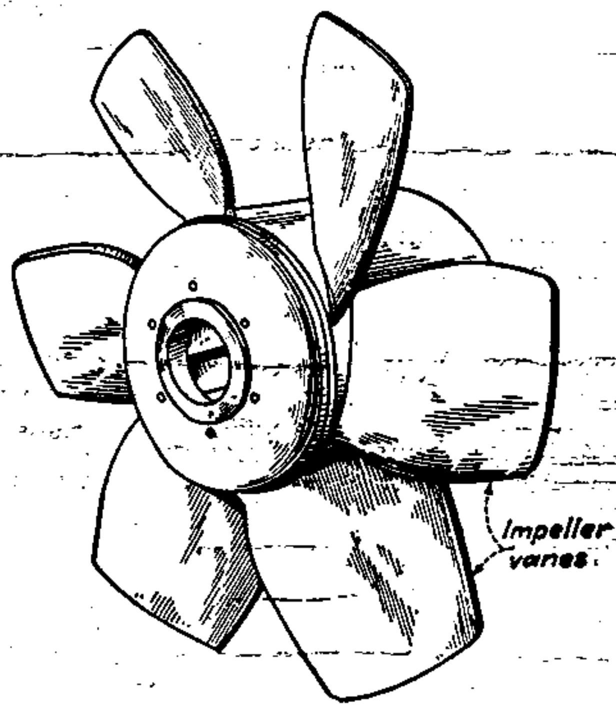


FIG. 1-17. Axial-flow impeller.

usually have three or four vanes. Generally non-clogging impellers are of the single-suction, overhung type.

Question 1-22: What is an overhung impeller?

Answer: If the impeller is supported by bearings on one side only and the shaft does not extend beyond the suction side of the impeller hub, it is called an "overhung impeller." This design makes possible the end-suction pump (Fig. 1-15).

Question 1-23: What is the suction eye of an impeller?

Answer: The inlet just before the point where the vanes start is called the "suction eye." In a closed-impeller pump, the diameter of the suction eye is taken as the smallest diameter of the shroud (Fig. 1-18). When determining the area of the suction eye, deduct that occupied by the impeller hub.

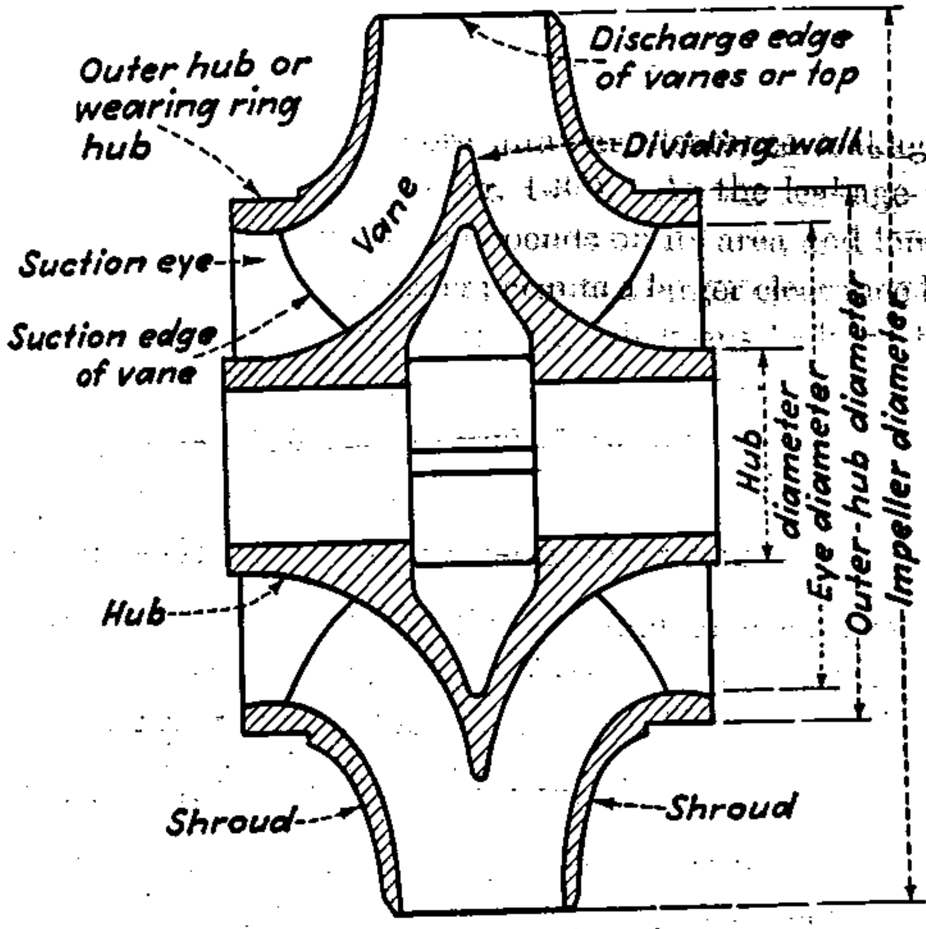


FIG. 1-18. Impeller nomenclature.

Question 1-24: What is the hub of the impeller?

Answer: In a centrifugal-pump impeller, it is the central part (Fig. 1-18) generally bored out for the pump shaft. It also frequently refers to the impeller part that rotates within the wearing ring when it is the outer impeller hub or wearing-ring hub of the shroud.

Question 1-25: What are a solid casing and a split casing?

Answer: "Solid casing" implies a design in which the discharge water passages leading to the discharge nozzle are all contained in one casing. To install or remove the impeller, at least one side must be open. Strictly speaking, a solid casing cannot be used, and casings normally called "solid" are radially split designs (Fig. 1-15).

"Split casing" implies that the casing is made of two or more parts fastened together.

Question 1-26: What is a horizontally split casing?

Answer: This term indicates a casing construction divided by a horizontal plane through the shaft center line (Fig. 1-19). This designation is not entirely satisfactory, and many design engineers prefer the term "axially split" casing. As both suction and discharge nozzles are normally in the lower half, this construction permits removal of the top half of the casing to inspect the interior of the pump without disturbing the bearings or piping.

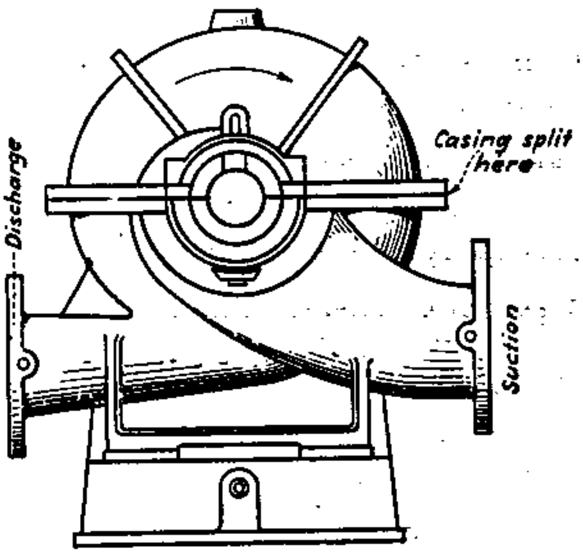


Fig. 1-19.

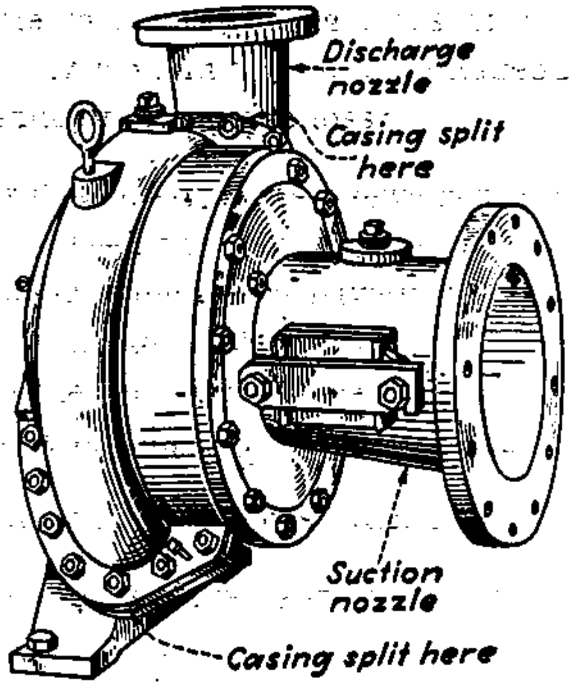


Fig. 1-20.

Fig. 1-19. End view of horizontally split-case centrifugal pump.
Fig. 1-20. Horizontal-shaft end-suction pump with casing split on diagonal plane through axis of shaft.

Question 1-27: What is a vertically split casing?

Answer: Like "horizontally split," "vertically split" is bad terminology. It is intended to refer to a casing split in a plane perpendicular to the axis of rotation (Fig. 1-15). The term "radially split" is now used by some designers to designate this type of construction.

Question 1-28: What is a diagonally split casing?

Answer: A horizontal-shaft pump with a casing split along a plane passing through its axis but inclined diagonally from the horizontal (Fig. 1-20). This construction, used primarily where vertical discharge is necessary, permits convenient opening of the casing for internal inspection of the pump.

Question 1-29: What is a twin volute?

Answer: A single volute (Fig. 1-21) completely surrounds the impeller. A twin volute (Fig. 1-22) is, in effect, two half-capacity

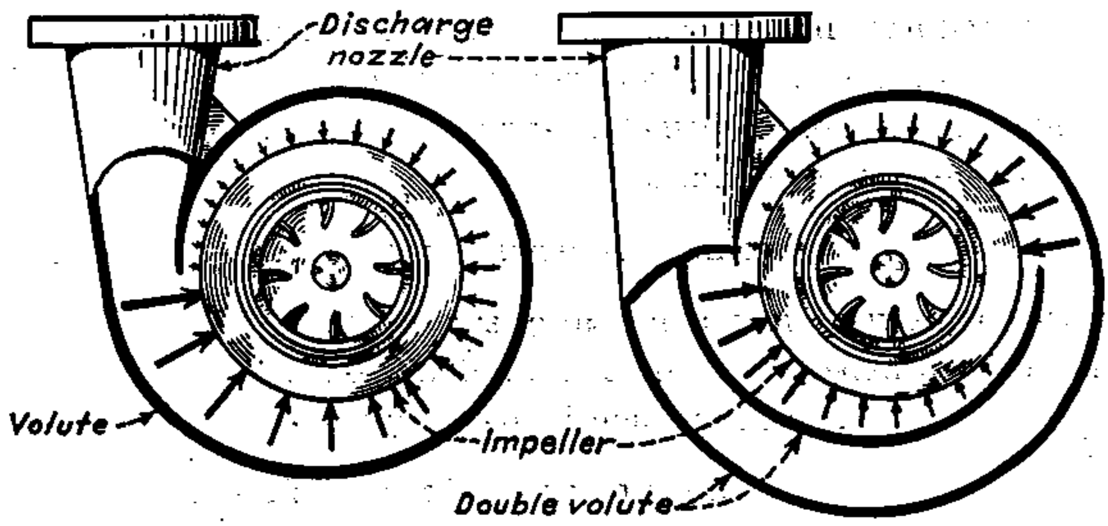


FIG. 1-21. FIG. 1-22.

FIG. 1-21. In pumps with a single volute, radial hydraulic forces do not balance.
FIG. 1-22. In a twin- or double-volute pump radial hydraulic forces balance.

volutes, each taking the discharge from 180 deg of the impeller in a way that approximately balances the radial thrust. It is sometimes used in single-stage pumps, particularly those of large capacity for high heads, and in multistage pumps developing high heads per stage.

Question 1-30: What is a twin-volute diffuser?

Answer: It is, in effect, a twin volute but cast separately from the casing. It resembles a diffuser with only two vanes (Figs. 1-23 and 1-24) and is used mainly on multistage pumps where the diffuser also contains the return channel to the stage that follows.

Question 1-31: What is radial thrust?

Answer: In a single-volute centrifugal pump (Fig. 1-21), fluid discharged by the impeller into the volute produces hydraulic forces that act radially on the impeller, as indicated by the arrows. These forces are not equal around the impeller and consequently produce a radial thrust that is transmitted to the shaft. At reduced capacities the volute area is not correct for the flow, and the hydraulic forces get considerably out of balance, producing a resultant force that acts radially on the shaft.

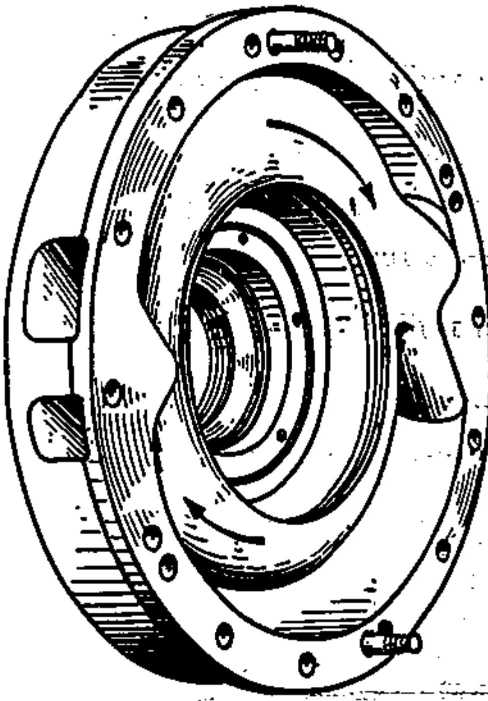


FIG. 1-23.

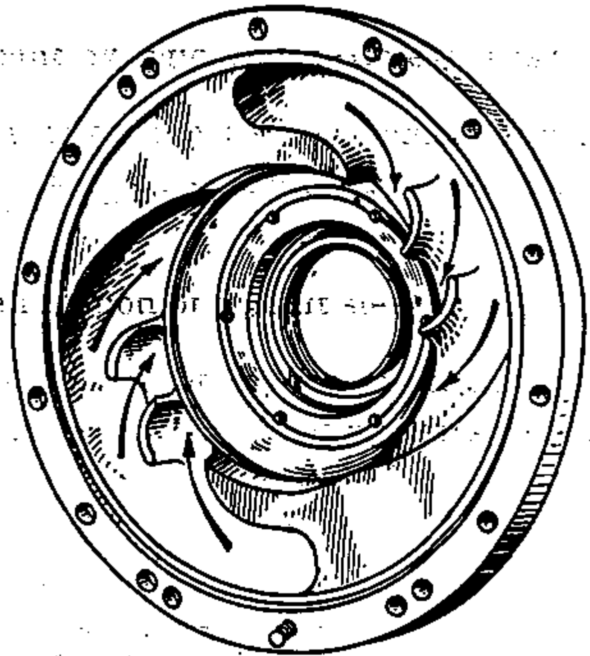


FIG. 1-24.

FIG. 1-23. View of last-stage twin-volute type of diffuser.
FIG. 1-24. Back view of intermediate-stage twin-volute diffuser showing return channel to guide the water to the following stage.

Question 1-32: What is a stop piece?

Answer: A stop piece is a projecting fin or section in the suction waterways of a centrifugal pump or its interstage passages guiding the flow into the impeller. In multistage pumps there is often incorporated in the interstage passages two or more vanes which may be either short or long and either straight or curved but generally of thin section (Fig. 1-24). These guide vanes function wholly or in part as stop pieces. In double-suction pumps the suction passageways on each side are now generally of the volute type (Figs. 2-6 and 2-7), and the stop piece (*A*, Fig. 1-25) divides the two sections of each volute passageway. From the outside, its location is

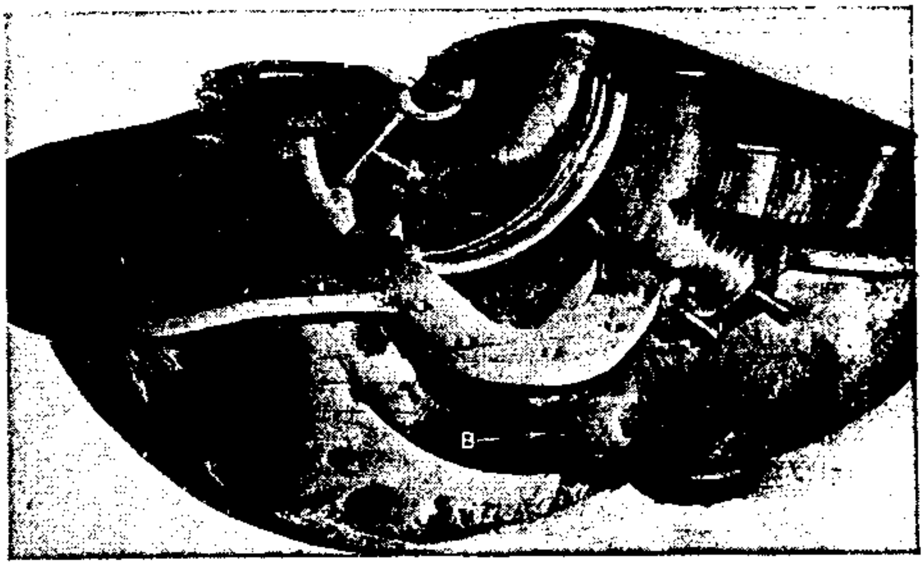


FIG. 1-25. Inverted top-half casing of a bottom-suction double-suction pump (similar to Fig. 2-7) showing at *A* the stop-piece section of suction waterway.

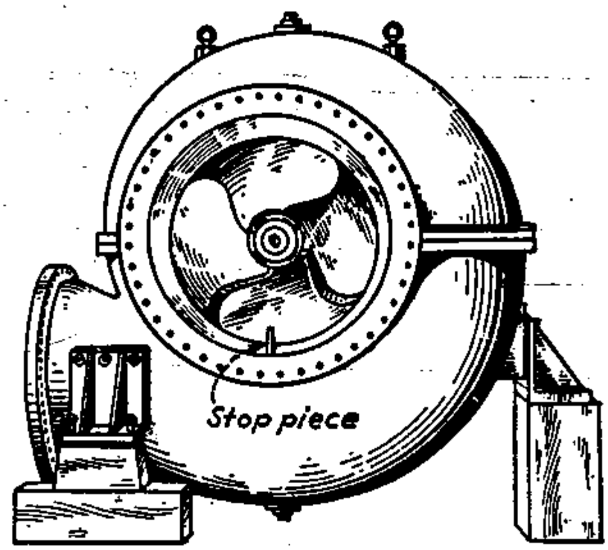


FIG. 1-26. Stop piece for end-suction pump is radial fin.

evidenced as at *B* (Fig. 1-25). In a few designs of end-suction pumps, a radial fin (Fig. 1-26) in the suction nozzle is used as a stop piece; in other end-suction designs no stop piece is used (Fig. 1-15). Occasionally the suction piping design used with end-suction pumps without stop pieces results in trouble with prerotation.

Question 1-33: What is prerotation?

Answer: Depending upon the individual design of the suction piping and suction passages or suction inlet of a pump, the column of liquid may spiral in the suction waterways some distance back from the impeller. This phenomenon is called "prerotation." It results from several causes, such as improper entrance conditions and shape of approach, and can decrease the net effective suction head and pump efficiency. Various means are used, either in the construction of the pump or in the pump-approach design, to avoid this phenomenon. There are cases, however, where the effects of prerotation have been used in special applications to produce desired effects in pump performance.

Question 1-34: Why do some end-suction, single-stage pumps with solid casings have casing suction heads (Fig. 1-27) for impeller removal, others have stuffing-box heads (Fig. 1-28), and still

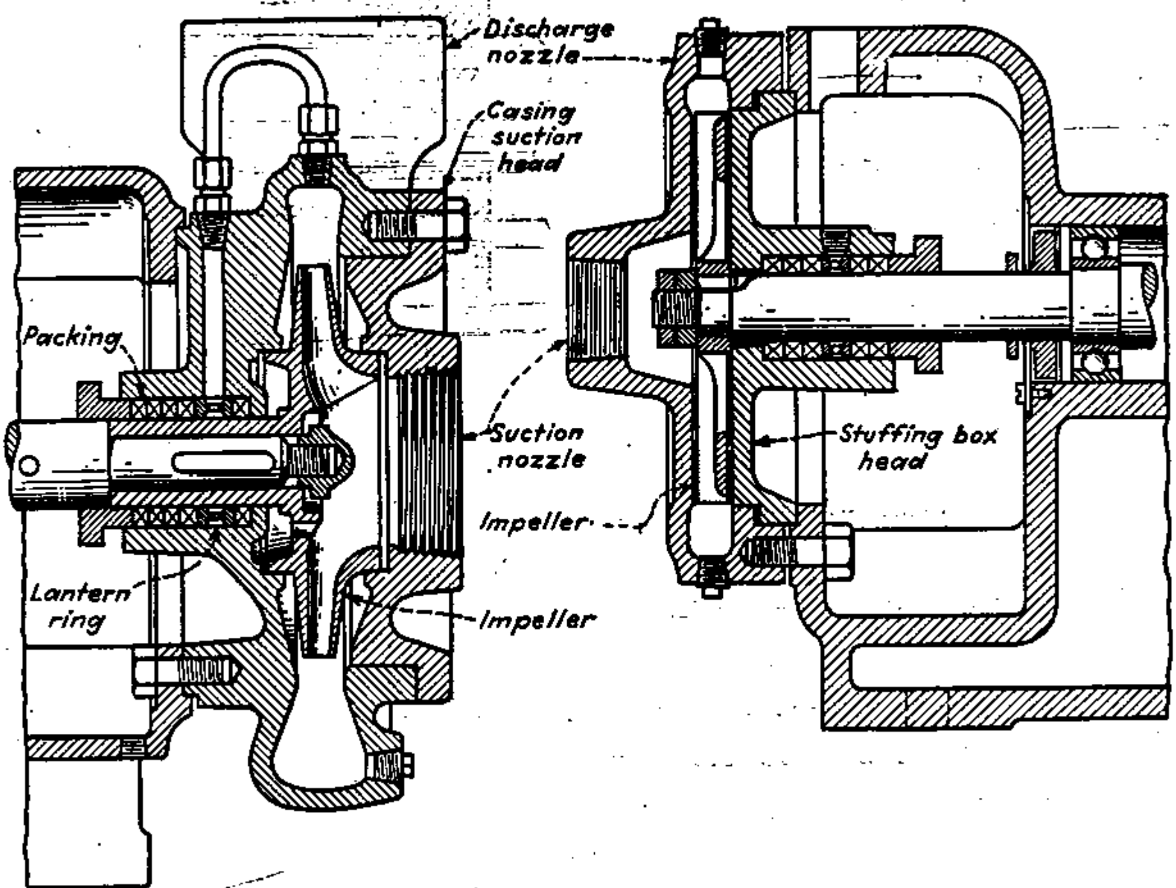


FIG. 1-27. FIG. 1-28.

FIG. 1-27. End-suction pump with removable suction head.
FIG. 1-28. End-suction pump with separate stuffing-box head.

others have both a casing suction head and a stuffing-box head (Fig. 1-29)?

Answer: For end-suction pumps up to 4- and 5-in. discharge and for the cheaper "contractor" designs, the volute and one side can be cast in one piece. The choice of whether the stuffing-box

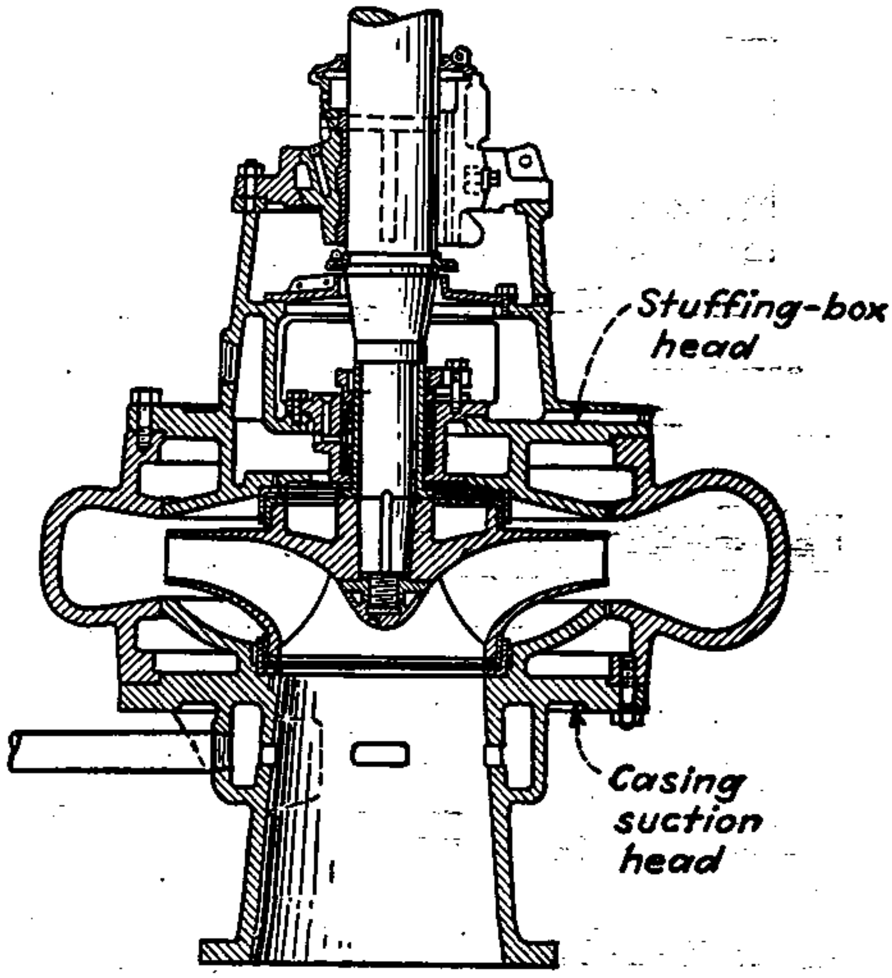


FIG. 1-29. End-suction pump with removable suction and stuffing-box heads.

side or the suction side is cast with the casing usually depends on the general design of the pump, with the aim to obtain the lowest cost. For some larger designs, like those for sewage, there is a demand for pumps of both rotations. A design with separate suction and stuffing-box heads (Fig. 1-29) permits using the same casing for either rotation if both flanges are identical. Vertical-shaft pumps can best be dismantled by removing the rotor and bearing assembly from the casing top, thus favoring a design with a removable stuffing-box head.

Question 1-35: What is a side plate?

Answer: It is a stationary part mounted within the casing (Fig. 1-30), used to provide a close-clearance guide to the water flowing through an open impeller.

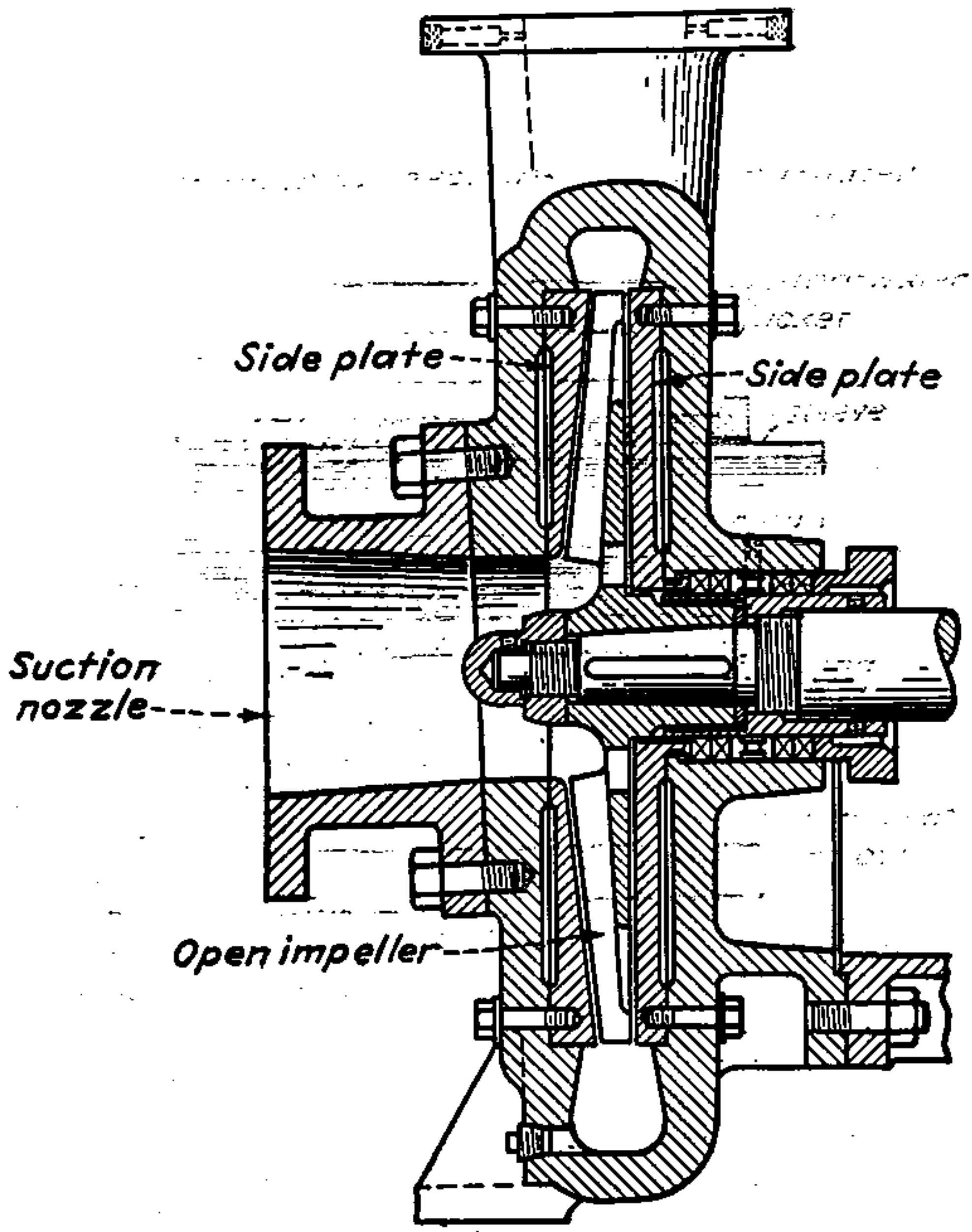


FIG. 1-30. Section of end-suction pump with open impeller and side plates.

Question 1-36: Why are wearing rings used?

Answer: In a centrifugal pump there must be a running joint, formed by a portion of the impeller and a portion of the casing, separating the chambers forming the suction and discharge waterways. The leakage of liquid through this running clearance joint will cause wear on the adjacent surfaces, the rate of wear being greater if there is grit or other foreign material in the liquid. In most pump designs, the casing casting proper is protected from this wear by attaching to the casing a removable stationary part or ring. Likewise the impeller is often made with an attached removable part forming the leakage joint surface. These removable casing and impeller parts are called wearing rings. Naturally they can be replaced when worn at a fraction of the cost of a new casing or impeller.

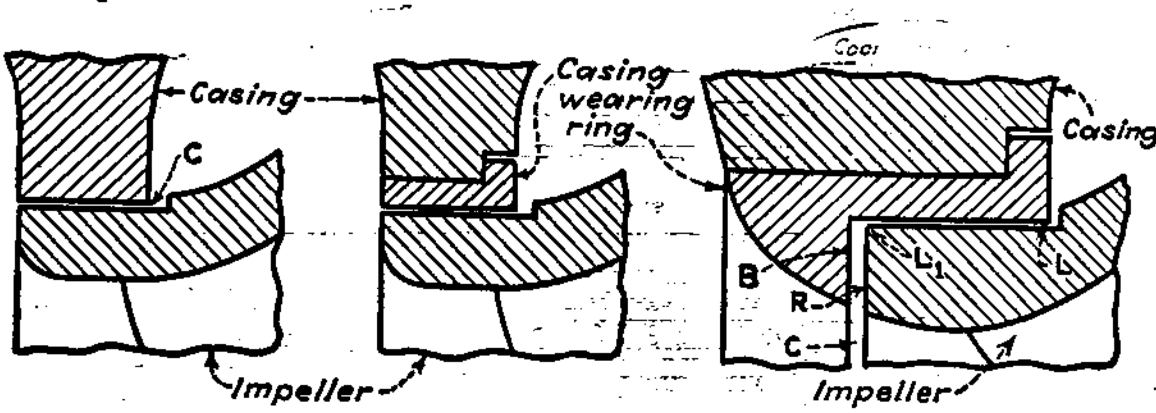


FIG. 1-31. FIG. 1-32. FIG. 1-33.

FIG. 1-31. Plain leakage joint between casing and impeller without removable part.
FIG. 1-32. Leakage joint similar to Fig. 1-31 but with removable casing ring.
FIG. 1-33. Leakage joint with L-shaped, or nozzle, casing ring.

Question 1-37: What is a casing wearing ring?

Answer: Sometimes called an "impeller guide ring," a casing wearing ring is inserted in the pump casing (Figs. 1-32 and 1-33) to protect the latter from wear.

Question 1-38: What is an impeller wearing ring?

Answer: To protect the impeller from wear at its running joint within the casing or casing ring, it is fitted with a removable ring called an "impeller wearing ring." (Figs. 1-34 to 1-38).

Question 1-39: What is a flat wearing ring?

Answer: It is a design in which the leakage joint is a straight annular clearance (Figs. 1-32 and 1-36). This type of ring has been replaced extensively by L-shaped and labyrinth designs.

Question 1-40: What are double wearing rings?

Answer: They are renewable rings on the impeller and in the casing (Figs. 1-34 to 1-38).

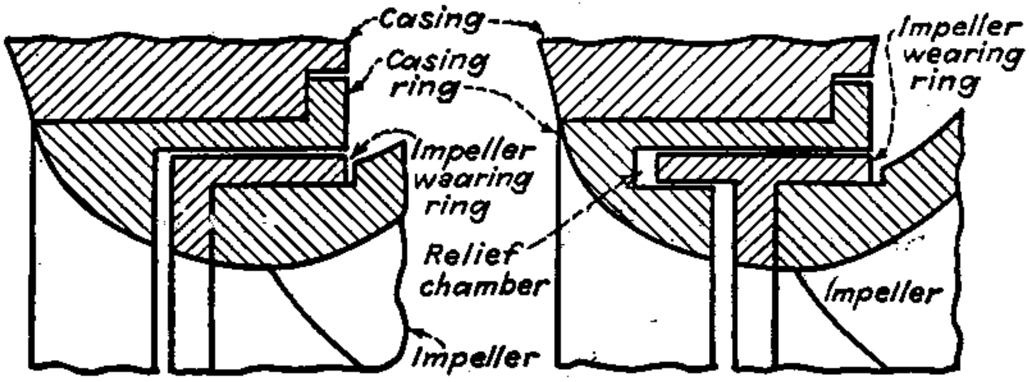


FIG. 1-34. FIG. 1-35.

FIG. 1-34. Leakage joint with L-shaped casing ring and L-shaped impeller ring.
FIG. 1-35. Leakage joint with rings of the intermeshing single-labyrinth type.

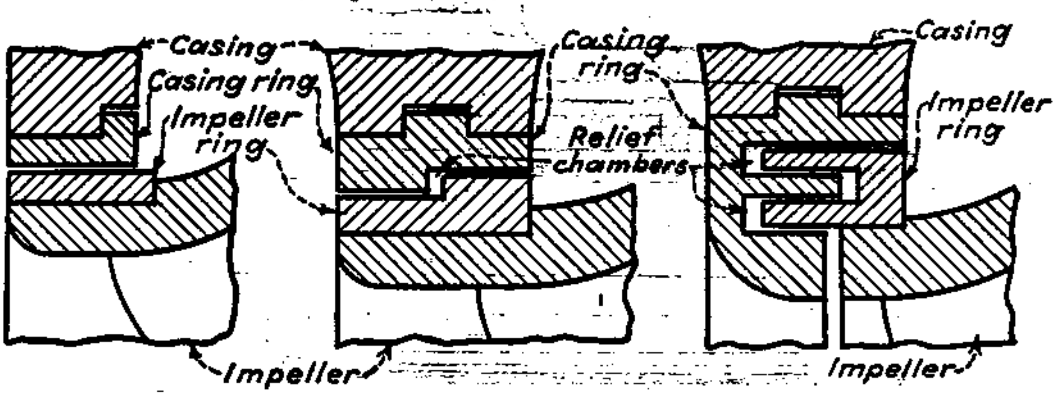


FIG. 1-36. FIG. 1-37. FIG. 1-38.

FIG. 1-36. Leakage joint similar to Fig. 1-31 but with double flat rings.
FIG. 1-37. Leakage joint similar to Fig. 1-36 but of the step type.
FIG. 1-38. Double-ring construction with labyrinth leakage joint.

Question 1-41: What is an L-shaped ring?

Answer: An L-shaped casing ring (Fig. 1-33), or nozzle type, as it is commonly called, gets its name from its "L" shape, with the short bar B parallel to the radial surface R of the impeller, which would otherwise be exposed to incoming flow. This construction avoids wearing-ring leakage coming into the suction stream at an angle almost perpendicular to it and causing eddy current and hydraulic losses. The leakage joint is along the long bar of the L from L to L_1, with large clearance C between the short bar of the

L to avoid losses because of high-velocity leakage at right angles to the suction flow line. An L-shaped ring is basically a flat ring with a protecting apron. The construction in Fig. 1-34 uses L-shaped wearing rings in the casing and on the impeller.

Question 1-42: What is a labyrinth ring?

Answer: It has two or more annular clearance leakage joints connected by relief chambers (Fig. 1-35). As the leakage through the clearance between two rings depends on its area and length, the leakage path of the labyrinth ring permits a larger clearance between the rotating and stationary rings and still keeps leakage to a low value. The step ring (Fig. 1-37), not commonly used, and the intermeshing ring (Fig. 1-38), often used, are other forms of labyrinth rings.

Question 1-43: What is an intermeshing ring?

Answer: A labyrinth ring with two annular clearance joints of different diameters involving a reversal of flow is often called an "intermeshing ring" (Fig. 1-38).

Question 1-44: What is a step ring?

Answer: It is the name applied to a wearing-ring design utilizing, in effect, two flat ring elements of slightly different diameters on the total width of a leakage joint. It is not very commonly used and could be classified as a subdivision of the labyrinth-ring group (Fig. 1-37).

Question 1-45: Is the location of the wearing rings restricted to the impeller suction eye, or can they be placed near the impeller discharge?

Answer: The amount of leakage through a clearance joint is controlled by its area, the length of the path, and the differential pressure. To obtain minimum area so that leakage also will be at a minimum, the clearance joint should have the smallest possible diameter. Thus, its logical location is at the impeller eye. There are applications where it is advantageous to locate the wearing ring at the impeller discharge, such as in vertical single-bottom-suction pumps handling water containing sand or grit, which will be idle at times. If the rings are located at the impeller eye, sand or grit may accumulate between the impeller and the suction-head wall. When such a pump is started, the sand will cause rapid wear of the sealing rings and may do other damage. Besides having the disadvantage of increased area and rubbing speed, wearing rings at the periphery of the impeller are objectionable in commercial lines of pumps, because in small sizes impeller diameters vary widely.

Question 1-46: If it is feasible to have wearing rings at both the impeller eye and periphery, why wouldn't it be advantageous to use both locations and thus reduce the leakage loss?

Answer: This construction has been used in a few sewage pumps to exclude solids from the space between the impeller and the side walls and to reduce sealing-ring wear, which is relatively rapid because of grit. It could not be advantageously used for high head pumps, as the outer rings increase the rotating surface area causing an additional disk-friction loss that offsets any saving in leakage.

Question 1-47: Are rings with axial clearances practical?

Answer: Rings with axial clearances have been used for centrifugal pumps but have not been as popular with designers or purchasers as those with radial clearances (Figs. 1-34 to 1-38). Because of variations in castings, it is generally desired that designs of double-suction or multistage pumps particularly permit shifting of the impellers on the shaft to center them closely in their volutes. Radial-clearance leakage joints permit this, but axial-clearance joints do not.

Question 1-48: What is the most desirable clearance-ring construction?

Answer: This varies with the kind of liquid handled, the pressure across the leakage joint, the rubbing speed between running and stationary clearance surfaces, and pump design. Generally, pumps have ring designs that their manufacturers have found best suited to the service for which the pump is used.

Question 1-49: How are casing rings held in the casing?

Answer: With axially split casings, it is general practice to use a tongue-and-groove joint for the ring in the lower half of the casing, so that bolting the two casing halves firmly locks the ring in place and prevents it from rotating (Figs. 1-2, 1-7, 1-8, and 1-40). In radially split casings, casing rings are pressed into the casing or the stage piece and are frequently bolted in place (Fig. 1-29).

Question 1-50: How are impeller rings held on the impeller?

Answer: They may be pressed or screwed on the impeller hub, depending on pump design, intended service, and the expected frequency of ring renewal. It is also general practice to hold impeller rings in place by setscrews or other fastening devices. In some designs, the impeller rings are shrunk onto the impeller, but this is restricted to materials which permit such an assembly.

Question 1-51: What is dynamic balance?

Answer: A pump rotor is in dynamic balance if all the various centrifugal forces resulting from its rotation balance each other. Such a rotor runs at full speed without vibration.

Question 1-52: What is static balance?

Answer: A pump rotor is in static balance if, when supported on parallel and level knife edges or their equivalent, it will remain stationary in any position in which it may be placed.

Question 1-53: What is the critical speed of a pump shaft?

Answer: Any object of elastic material has a natural period of vibration. If a pump rotor or shaft rotates at a speed corresponding to its natural frequency, any minor unbalance is magnified. Such speeds are called "critical" speeds. The lowest is called the first critical; the next higher, the second, etc. When the rotor of a centrifugal pump operates at its critical speeds, it vibrates excessively, its shaft deflections exceed internal clearances, and the shaft may fail.

Question 1-54: What is the relationship between a critical speed of a pump shaft and its operating speed?

Answer: Generally the shaft dimensions of a single-stage pump cause the first critical speed to be considerably above its operating speed. However, centrifugal pumps may operate above their first critical speed without danger if the speed is high enough. This is done in multistage pumps by designing their shafts so that the first critical speed is from 60 to 75 per cent of the operating speed.

Question 1-55: What is meant by rigid and flexible shafts?

Answer: A rigid shaft has a first critical speed higher than its operating speed. A flexible shaft has its operating speed higher than one of its critical speeds.

Question 1-56: What is the function of a shaft sleeve?

Answer: A shaft sleeve (Fig. 1-39) protects the shaft, thus preventing corrosion, erosion, and wear from affecting its strength.

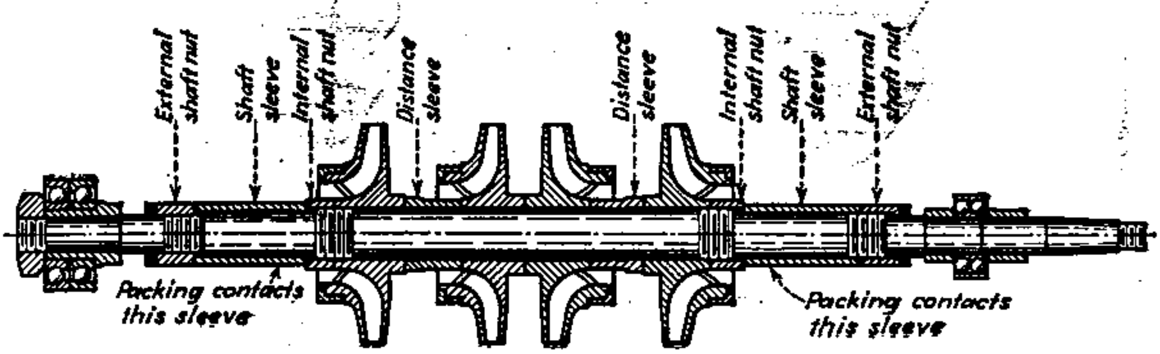


FIG. 1-39. Section of a four-stage pump rotor, identifying various sleeves and shaft nuts.

Many forms are used in centrifugal pumps; the commonest protects the shaft against wear where it extends through the stuffing box. On small pumps, a shaft sleeve may increase the outside diameter of the shaft to a point where it seriously reduces the impeller suction area and increases hydraulic and stuffing-box losses. Because of this, small pumps are very commonly built without shaft sleeves, making the shaft of a stainless steel which is sufficiently corrosion- and wear-resistant to give a satisfactory life.

Question 1-57: What is an interstage, or distance, sleeve?

Answer: It is a sleeve (Fig. 1-39) that protects the shaft between the impellers of a multistage pump. In some designs, impellers have long hubs that make distance sleeves unnecessary.

Question 1-58: What is the purpose of a stuffing box?

Answer: When pressure at its inner end is below atmospheric, the stuffing box prevents air leakage into the pump. When pressure at its inner end is above atmospheric, the stuffing box prevents liquid leaking out of the pump.

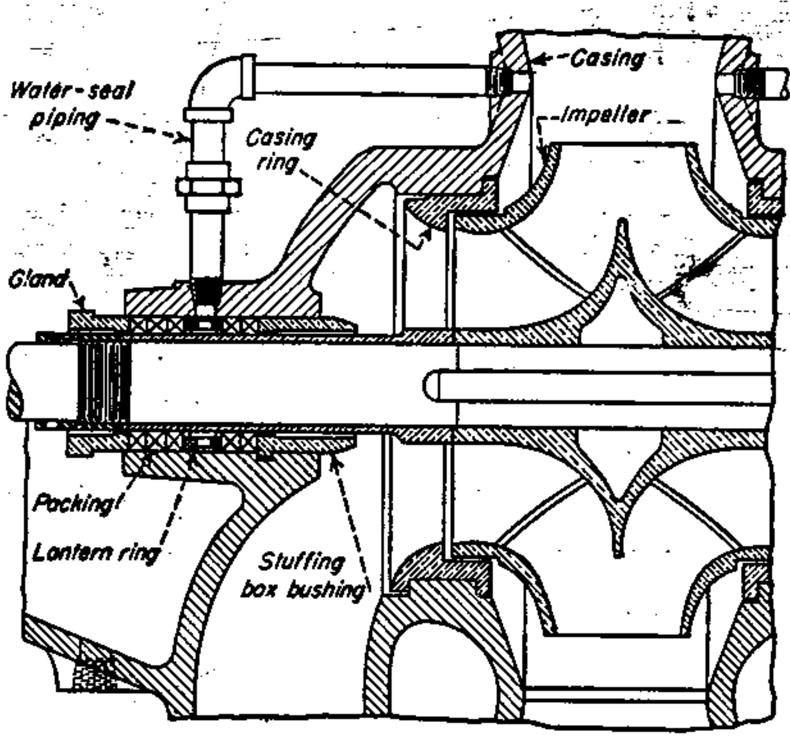


FIG. 1-40. Partial section of a double-suction pump to show stuffing-box bushing and other stuffing-box details.

Question 1-59: What is a stuffing-box bushing?

Answer: It is a removable piece that forms the stuffing-box bottom in some smaller sizes of centrifugal pumps (Fig. 1-40). It is not split, even though the pump casing in which it fits is split axially, and is generally held from rotation by a locked tongue-and-groove joint in the lower half of the casing. On large pumps, stuffing-box bushings are rarely used (Fig. 1-41). The use of stuffing-box throat bushings is based mainly on manufacturing considerations rather than on any particular operating advantage. They generally are used in small pumps, in which the diameter of the bottom of an integral stuffing-box would limit the boring-bar size to too small a diameter.

Question 1-60: What is the purpose of a stuffing-box seal or lantern ring?

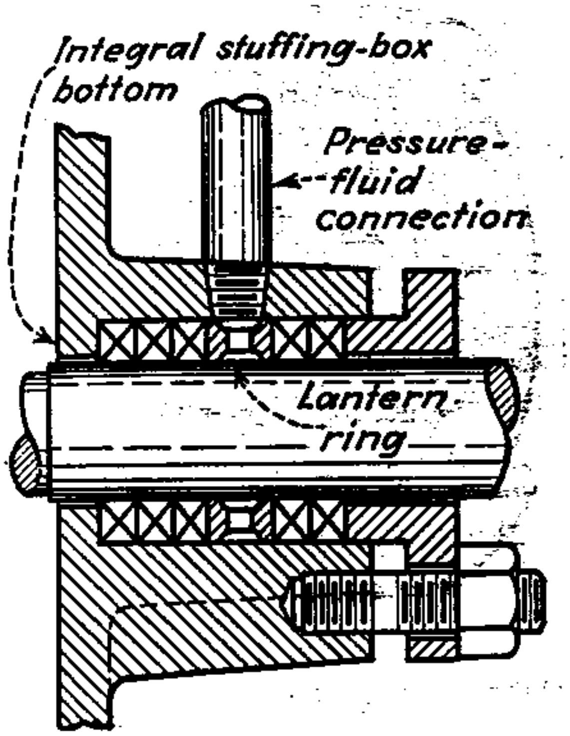


FIG. 1-41.

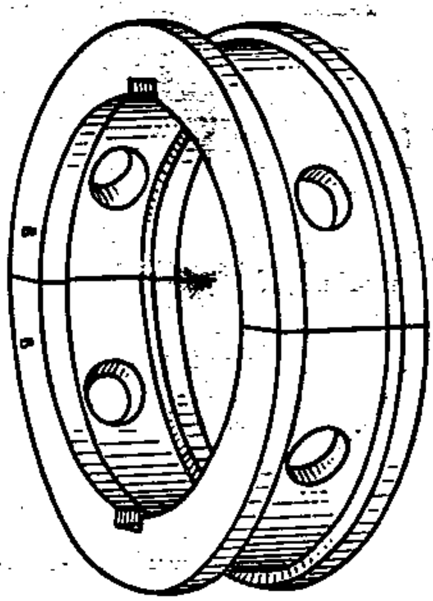


FIG. 1-42.

FIG. 1-41. Section of stuffing box with integral bottom.
FIG. 1-42. Typical lantern ring or seal cage design.

Answer: To prevent air leakage into the pump when operating on a suction lift, a water-seal cage or lantern ring (Fig. 1-42) is fitted into the stuffing box (Figs. 1-40 and 1-41). Water or sealing fluid under pressure piped to the space provided by the lantern ring (Fig. 1-40) makes an effective seal beyond which air cannot pass into the pump. This seal chamber also provides a reservoir of water or fluid for cooling and lubricating the packing. A seal may also prevent inflammable or chemically active and dangerous liquids from flowing out to the atmosphere.

Question 1-61: When is an independent sealing liquid needed in a lantern ring?

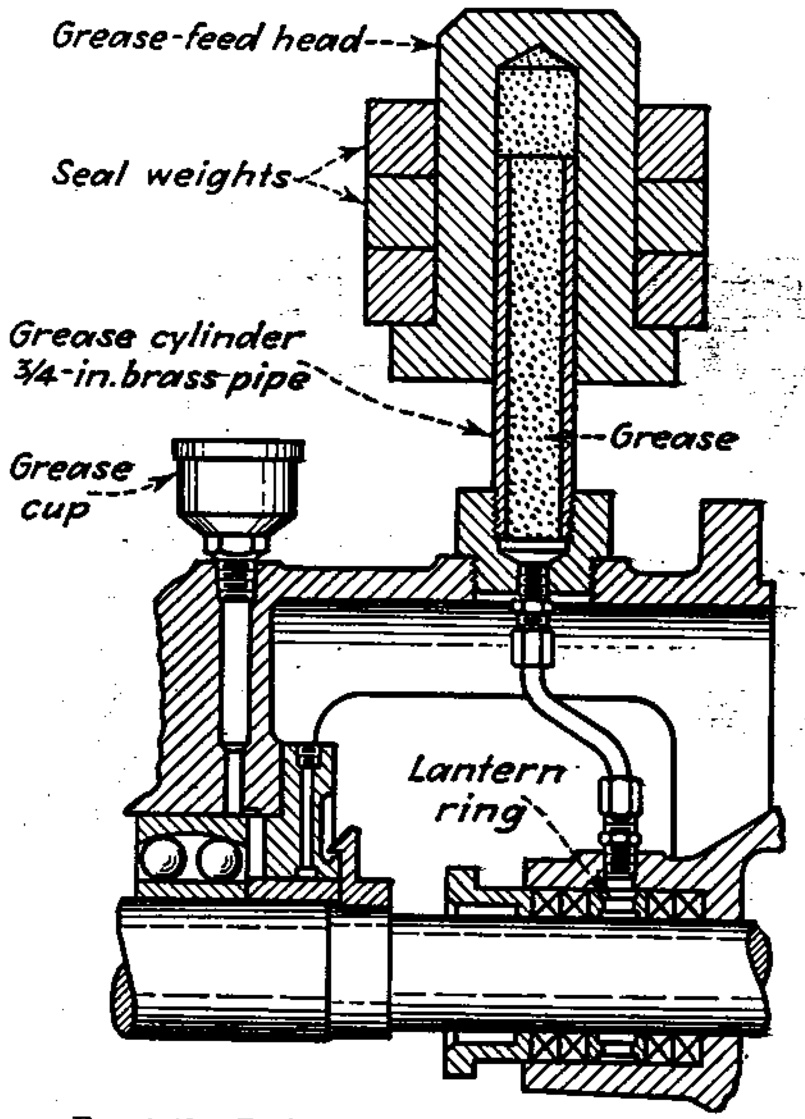


FIG. 1-43. Design of weight-loaded grease seal.

Answer: When a pump handles clean, cool water, the stuffing-box seal or seals are usually connected to the pump discharge (Fig. 1-40) in case of a single-stage pump or to the waterways of the first stage in case of a multistage pump. An independent water-seal supply should be used, if possible, when:

1. Suction lift exceeds 15 ft.
2. Discharge pressure is less than about 10 lb or 23 ft head.

3. Pump handles hot water (over 250°F) and adequate cooling is not otherwise provided.
4. Water is muddy, sandy, or gritty.
5. Pump is in hotwell service.
6. Acids, juices, molasses, or liquids other than water are handled, with no special provision in the stuffing-box design for the liquid pumped.

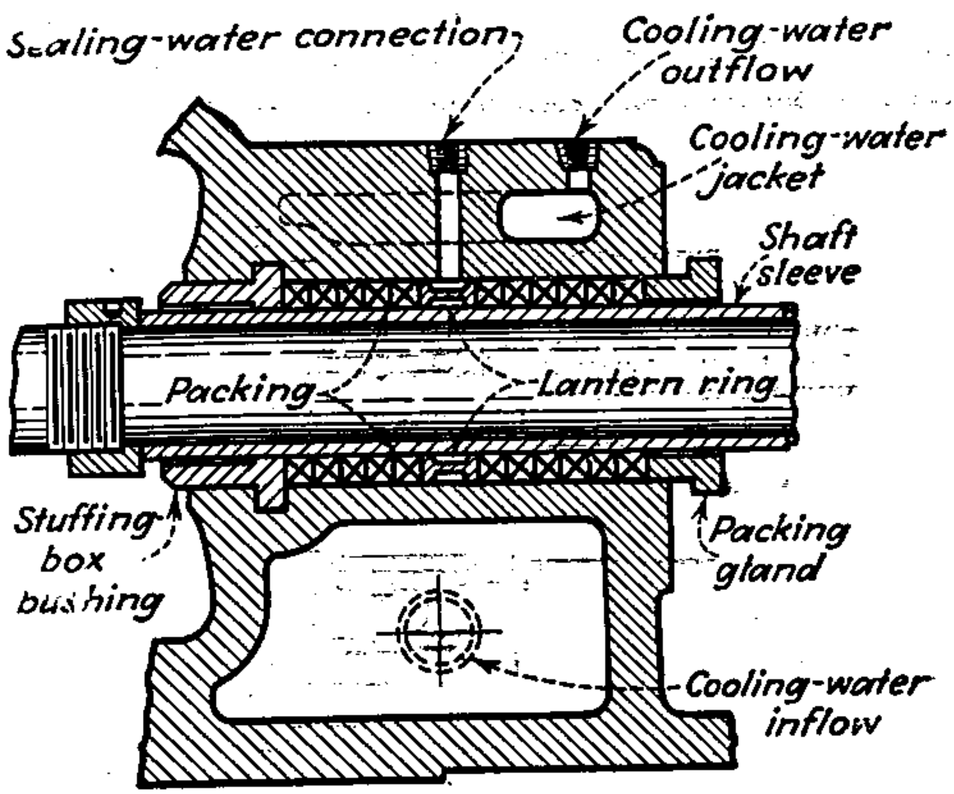


FIG. 1-44. Water-cooled stuffing box integral with casing.

Question 1-62: When is a grease seal used?

Answer: Grease is often used as a sealing medium for a stuffing box when clear water or a suitable clear liquid is not available or cannot be used. Grease sealing of the stuffing boxes is very commonly used for sewage pumps and low head drainage pumps, even if operating on suction lifts. Grease seals are also used frequently on chemical pumps. The grease is usually fed into the seal by spring-loaded grease cups, but weighted grease sealers (Fig. 1-43) are sometimes employed.

Question 1-63: What is the purpose of water-cooling stuffing boxes?

Answer: To reduce the temperature of the liquid leaking through the packing or to remove some of the heat of friction developed at the packing or both, pumps handling hot liquids and/or operating with high stuffing-box pressures are generally provided with jacketed stuffing boxes (Figs. 1-44 to 1-46) so a cooling medium, usually water, can be circulated through the jacket.

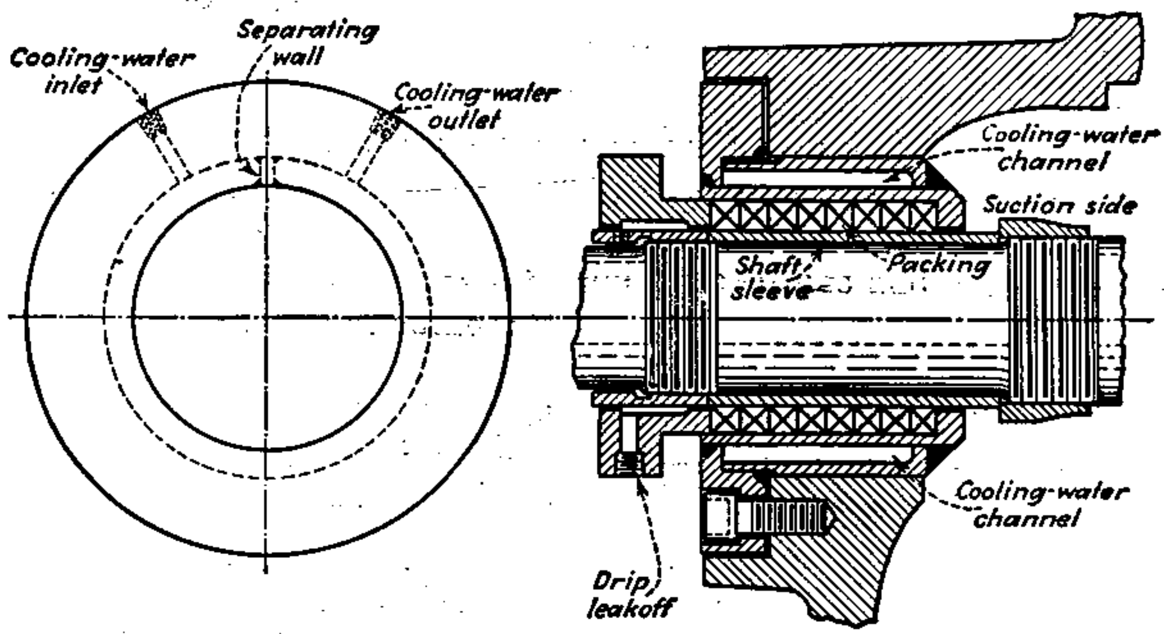


FIG. 1-45. Water-cooled stuffing box inserted in casing.

Question 1-64: What is a smothering or quenching gland and where is it used?

Answer: In many pump applications, stuffing-box leakage to the atmosphere is objectionable because it would create an explosion hazard or be dangerous to personnel. This is frequently true of pumps handling hydrocarbons at temperatures above their flash point or vaporizing temperatures. When leakage cannot be cooled sufficiently by the water-cooled stuffing box, provision is made for circulating a liquid—either water or another hydrocarbon—at low temperature through a channel between the packing gland and shaft sleeve. This liquid mixes intimately with the leakage, lowering its temperature or, for volatile liquids, absorbing or smothering the leakage.

Question 1-65: What is a split gland and when is it used?

Answer: Split glands are made in halves, so that they can be removed from the shaft without dismantling the pump, to allow more working space during repacking. Their halves are generally held together by the gland bolts or separate bolts, but other designs are also used. Figure 1-47 shows one half of a split gland that is held together by separate bolts.

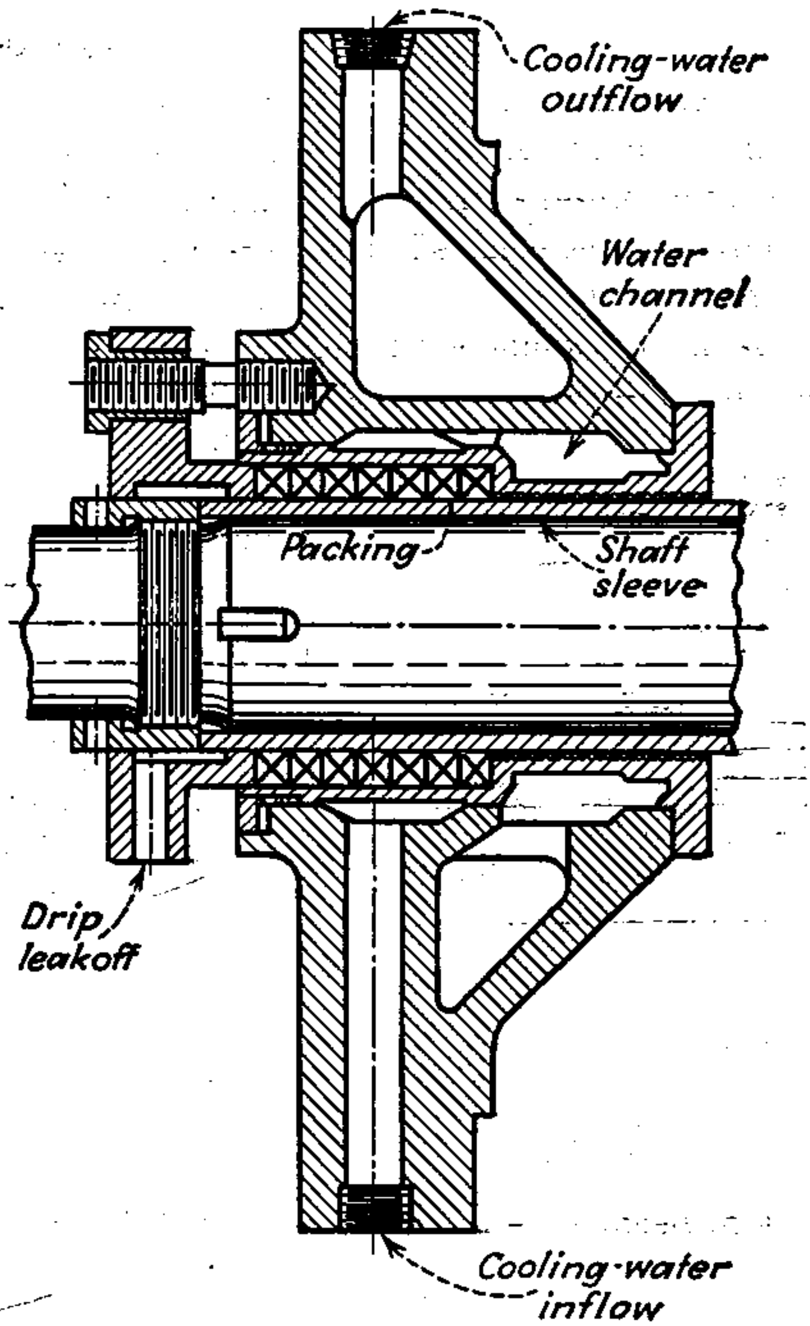


FIG. 1-46. Water-cooled stuffing-box arrangement for precooling stuffing-box leakage before it reaches packing.

Question 1-66: What is a stuffing-box pressure-reducing device?

Answer: When pressures exceed the value against which the stuffing box can be safely packed, a pressure-reducing device may be provided. Essentially, it consists of a bushing (Fig. 1-48) or a

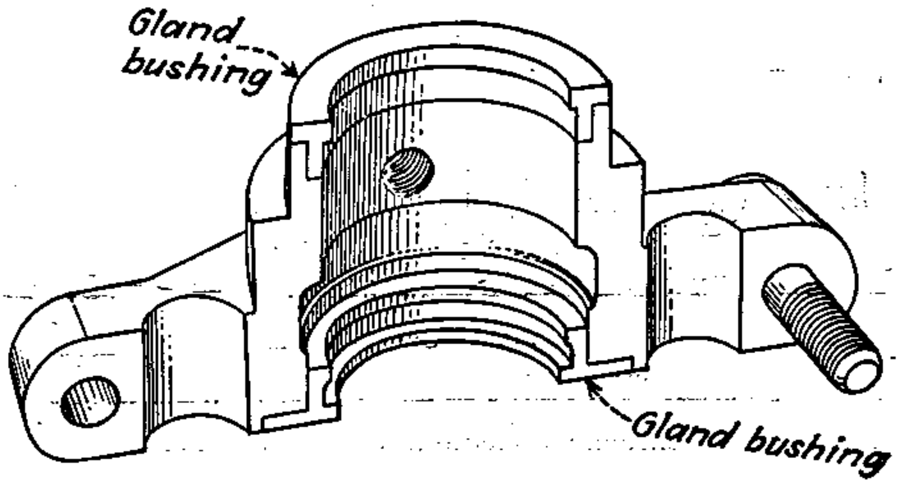


FIG. 1-47. Half of split gland provided with gland bushings.

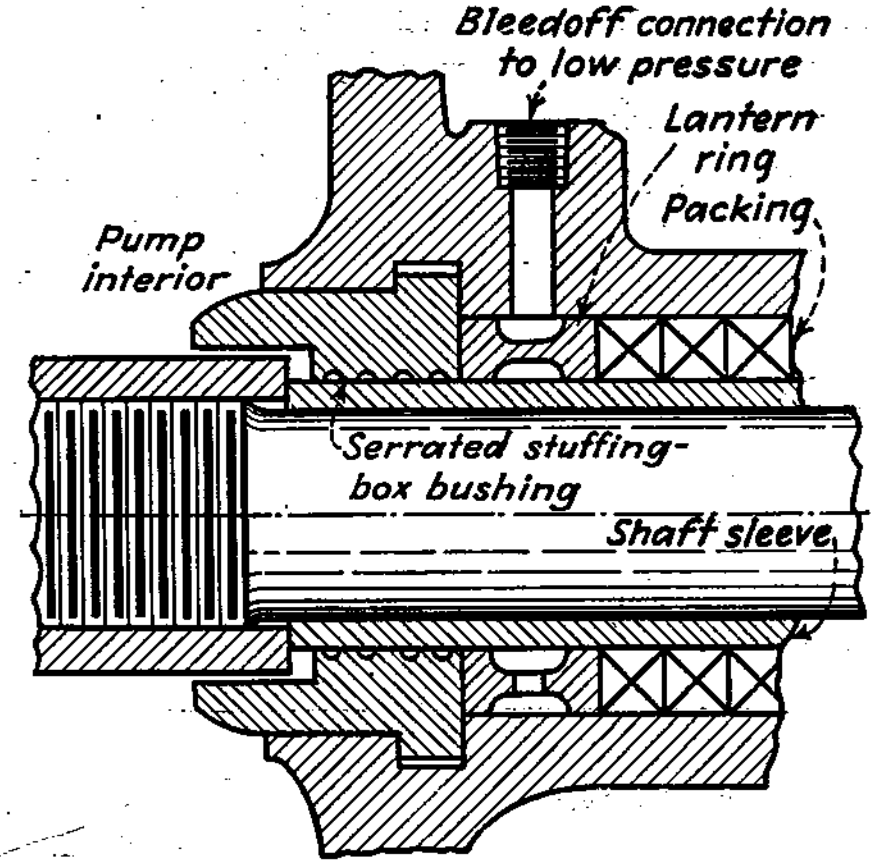


FIG. 1-48. Stuffing-box design with serrated stuffing-box bushing acting as pressure-reducing device.

meshing labyrinth located between the pump interior and the stuffing box proper, ending in a relief chamber. The latter is connected to a region of lower pressure, and leakage past the pressure-reducing device is returned to this point.

Question 1-67: What is a mechanical seal?

Answer: All mechanical seals, while differing in various physical respects, operate fundamentally on the same principle. In each, the sealing surfaces (Fig. 1-49) are located in a plane perpendicular to the shaft and usually consist of two polished lubricated surfaces running on each other (at S), one surface being connected to the shaft and the other to the frame. The polished or lapped surfaces of dissimilar materials are held in continual contact by springs. The surfaces form a fluid-tight seal between rotating and stationary members, with small frictional losses. In Fig. 1-50, a mechanical seal is installed on each side of the impeller at S.

Question 1-68: Why are mechanical stuffing-box seals used?

Answer: For many service conditions, a conventional stuffing box with composition packing is impracticable for sealing a rotating shaft, particularly where the seal must be leakproof. A mechanical seal provides a leakproof design with low power losses for a wide range of fluids, such as corrosive acids and gritty or inflammable liquids.

Question 1-69: What bearing designs are used on centrifugal pumps?

Answer: Practically every design has been used. Even the same basic pump design is often built with two or more bearing designs, as service conditions or the purchaser requires. For double-suction, single-stage, general-service pumps, two external bearings, one on either side of the casing, are generally used. Formerly oil-lubricated sleeve bearings were used, but most pump manufacturers now use anti-friction bearings extensively, which are either grease- or oil-lubricated.

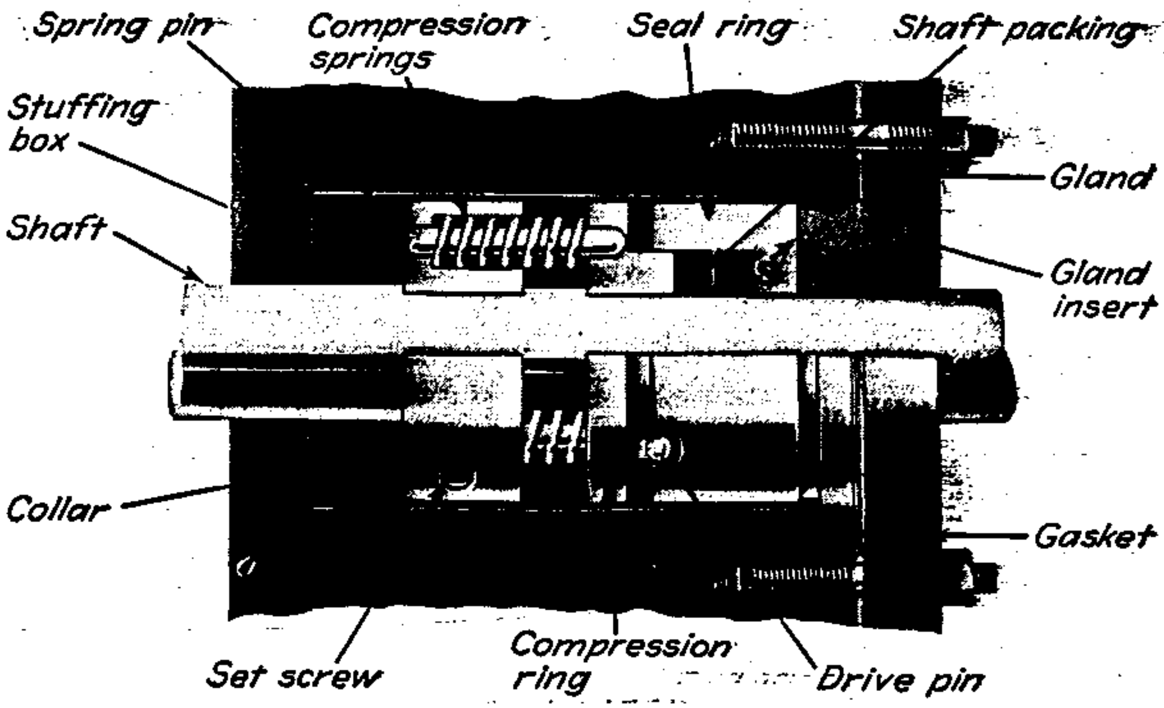


FIG. 1-49. Section of a mechanical seal.

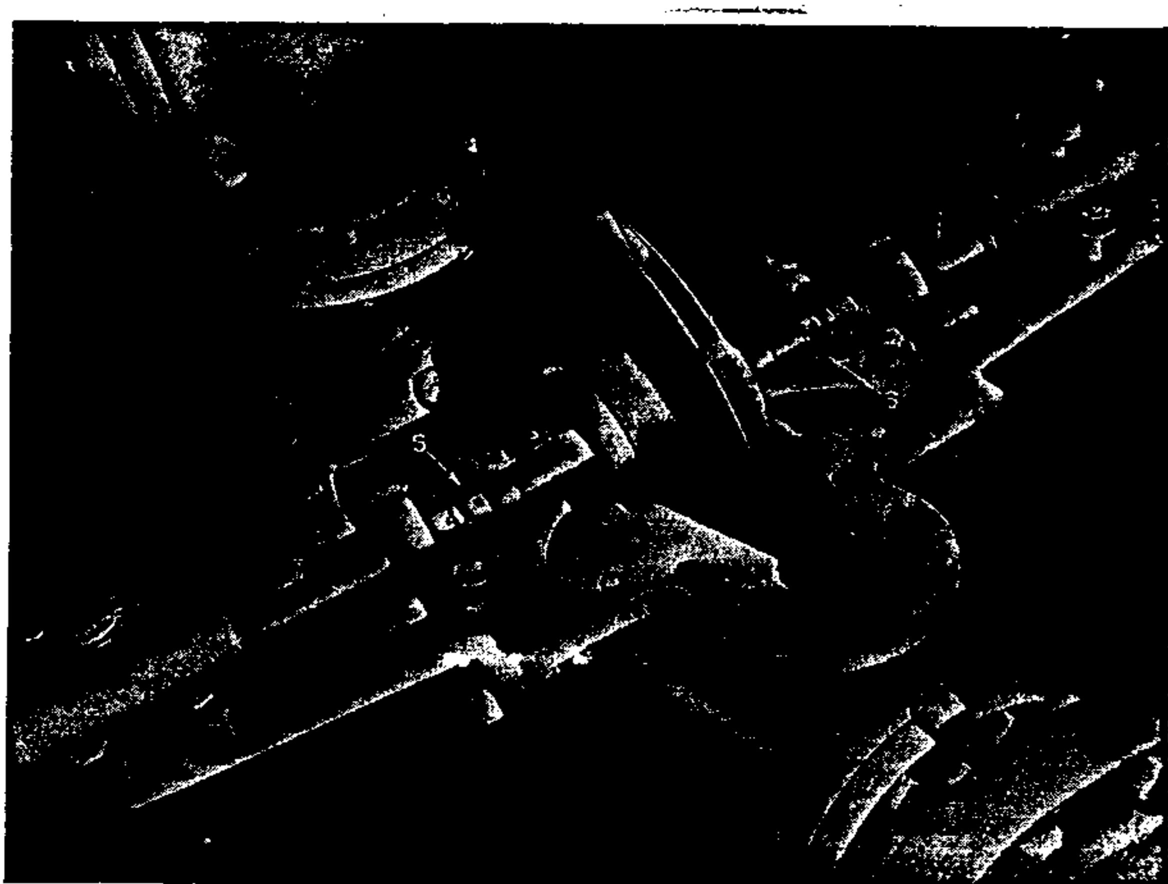


FIG. 1-50. Partially assembled pump with seals as shown in Fig. 1-49.

Question 1-70: What is a self-aligning bearing?

Answer: It is made to adjust itself automatically to a change in the angular position of the shaft. In babbitted bearings, the name applies to bearings in which the bearing shell has a spherical fit in the housing. In anti-friction bearings, it applies to those with a spherically ground outer race *R* (Fig. 1-51) to permit the bearing to align itself to the position of the shaft.

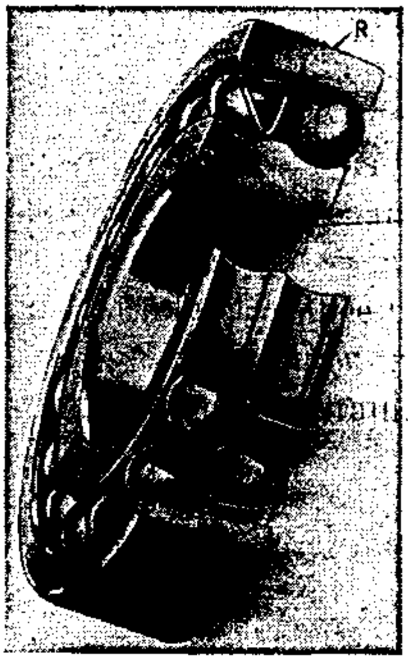


FIG. 1-51.

FIG. 1-52.

FIG. 1-51. Self-aligning double-row ball bearing.
FIG. 1-52. Single-row deep-groove ball bearing.

Question 1-71: What designs of anti-friction bearings are used?

Answer: Various ball-bearing designs are most commonly used. Roller bearings are used to a lesser degree, but spherical roller bearings (Fig. 1-57) are used quite extensively on large shafts, for which the choice of ball bearings is limited.

Question 1-72: What types of ball bearings are used?

Answer: The most common are single-row deep-groove (Fig. 1-52), double-row deep-groove (Fig. 1-53), double-row self-aligning (Fig. 1-51), and the angular-contact type made in double- and single-row designs (Figs. 1-54 and 1-56). All except the double-row self-aligning bearings can take both axial-thrust and radial loads. The ball thrust bearing (Fig. 1-58) was formerly used to take thrust only in combination with babbitted bearings, but now it is rarely used on centrifugal pumps. Sealed, adapter, and other ball bearings are used in special applications.

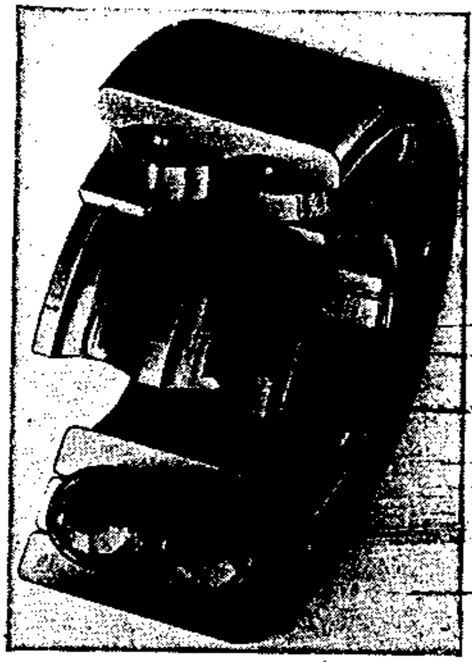


FIG. 1-53.

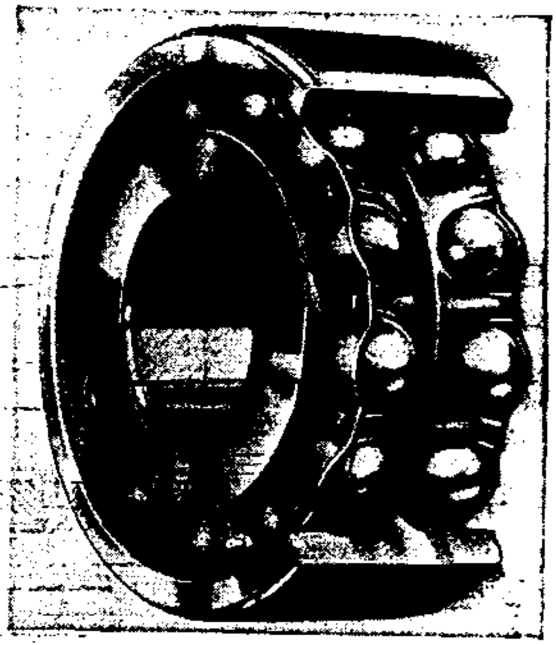


FIG. 1-54.

FIG. 1-53. Double-row deep-groove ball bearing.
FIG. 1-54. Double-row angular-contact ball bearing.

Question 1-73: When are self-aligning ball bearings used?

Answer: For heavy loads, high speeds, and long bearing spans with no end thrust, they are well adapted for line bearings on centrifugal pumps. The double row of balls (Fig. 1-51) runs in fixed grooves in the inner or shaft race, and the outer race is spherically ground; it therefore operates as a pivot to take slight vibration or shaft deflection. It has very little thrust capacity and is not used for combined radial and thrust loads in centrifugal pumps. With large shafts, the self-aligning spherical-roller design (Fig. 1-57) is used instead of self-aligning ball bearings (Fig. 1-51). Self-aligning spherical-roller bearings can carry a combined radial and axial load with considerable thrust load component.

Question 1-74: When are deep-groove ball bearings used?

Answer: Single-row deep-groove ball bearings (Fig. 1-52) are applicable to many services. They can take both radial and thrust loads. For conservative speeds, light thrust loads, and short spans, these bearings are satisfactory. Double-row deep-groove ball bearings (Fig. 1-53) are, in effect, two single-row bearings with greater capacity both for radial and thrust loads. They are used quite generally when loadings exceed single-row bearing ratings.

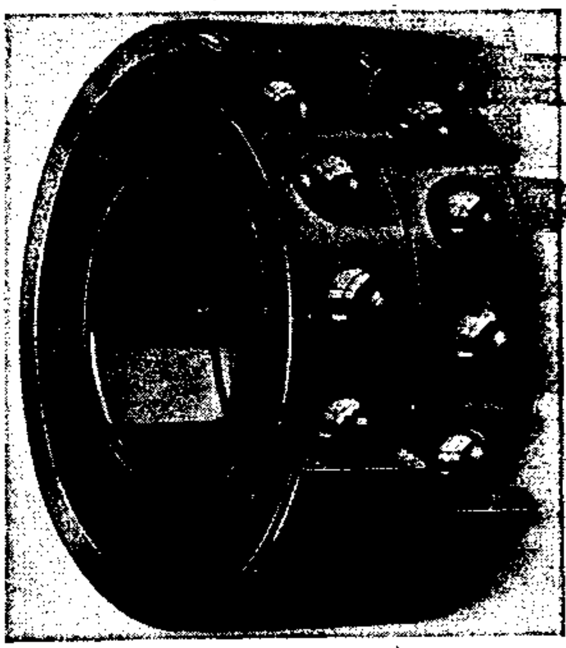


Fig. 1-55.

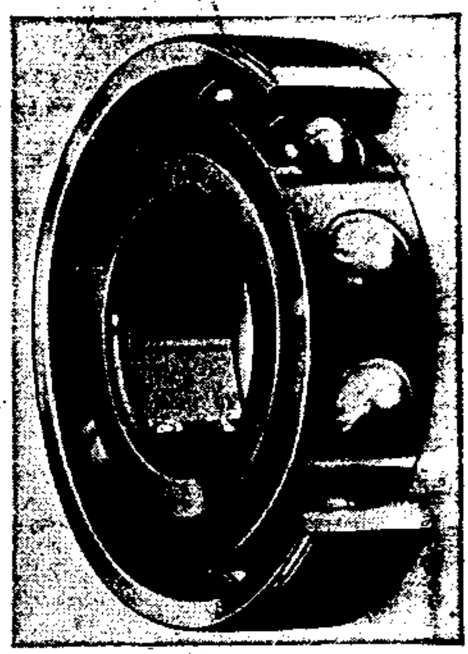


Fig. 1-56.

Fig. 1-55. Two single-row angular-contact ball bearings mounted back to back act as a double-row bearing.

Fig. 1-56. Single-row angular-contact bearing.

Question 1-75: When are angular-contact ball bearings used?

Answer: Single-row angular-contact ball bearings (Fig. 1-56) can be used only on centrifugal pumps when there will be a thrust at all times in one direction, and are limited primarily to vertical pumps. The double-row bearing (Fig. 1-54) or its equivalent, a matched pair mounted back to back (Fig. 1-55), are satisfactory for thrust loads in either direction. Some pump manufacturers standardize on this double-row angular-contact bearing for combined radial and thrust bearings and also use it for many applications where the double-row deep-groove design can be used.

Question 1-76: What dictates a designer's choice of lubricant for the bearings of a pump?

Answer: In laying out a line of centrifugal pumps, the choice of lubricant for their bearings is dictated by application requirements, by cost considerations, and sometimes by the preferences of a group of purchasers who use a major portion of a line of a particular design. For example, almost all oil refinery centrifugal pumps have oil-lubricated bearings because refinery engineers insist on this feature. On the other hand, in the marine field preference is for grease-lubricated bearings. For highly competitive lines of small pumps, cost is a main consideration, and the lubricant is chosen to obtain the lowest cost consistent with the bearing design and the requirement for high reliability of the pump in service.

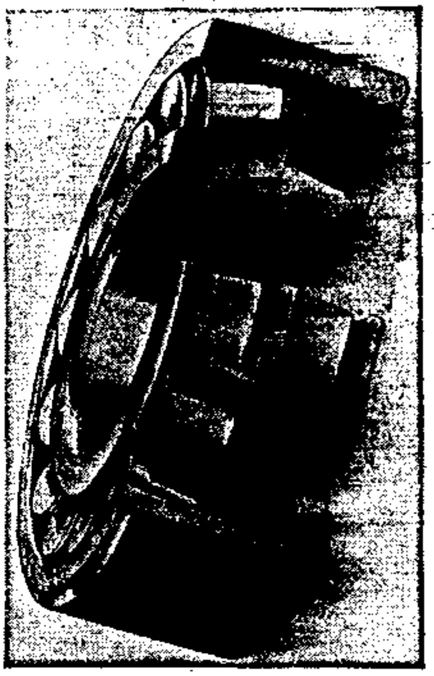


FIG. 1-57.

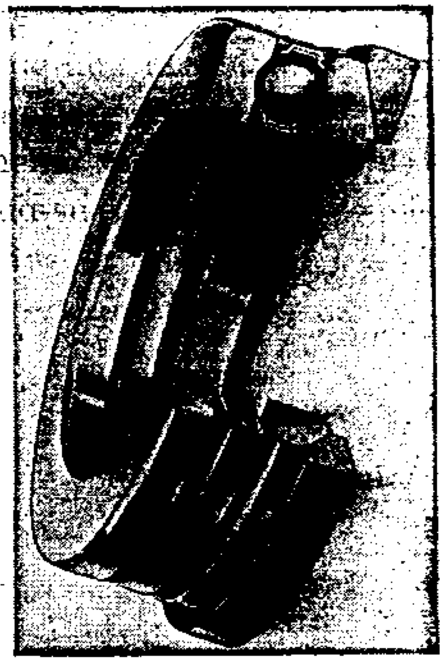


FIG. 1-58.

FIG. 1-57. Self-aligning spherical-roller bearing.
FIG. 1-58. Spherically seated one-direction ball thrust bearing.

Question 1-77: How are ball bearings lubricated in centrifugal pumps?

Answer: They are generally grease-lubricated, although in the larger size bearings oil lubrication is frequently applied. When grease-lubricated, the bearings are partly filled with grease, which the rotating balls throw out on the housing walls, creating a slight suction at the inner race. Because the grease is a semisolid, it does not flow into the bearing until a small rise in temperature occurs, when some grease melts and runs into the bearing, lubricating it. After this, continuous circulation of a small amount of grease keeps the bearing lubricated and cool with the minimum of attention.

Do not pack the housing of a grease-lubricated ball bearing full of grease, as this prevents proper grease circulation in the bearing and its housing. A rough rule recommends filling the housing one-third full. An excess of grease causes the bearing to heat and grease to flow out from the seals.

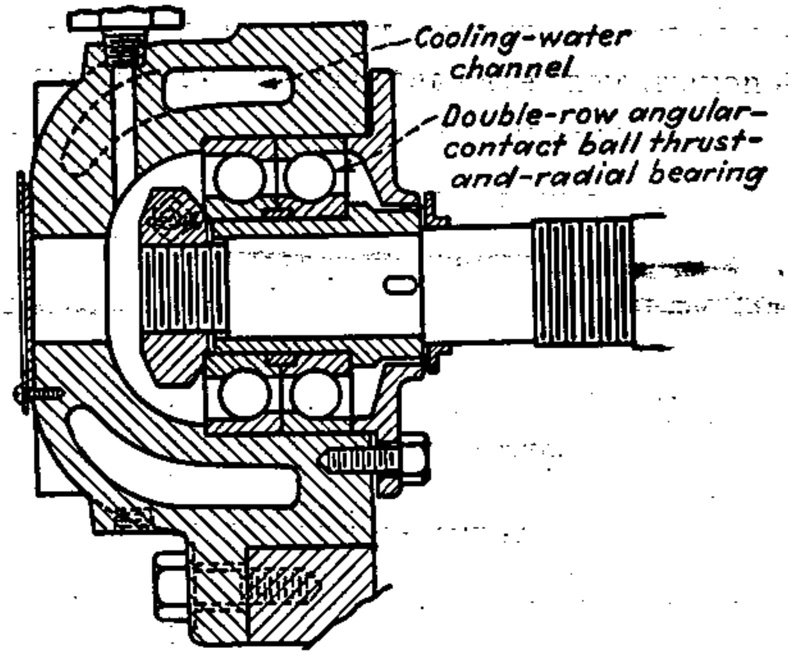


FIG. 1-59. Water-cooled housing for ball bearing.

Question 1-78: When and how should pump bearings be water-cooled?

Answer: If the amount of heat generated in the bearings is too much to be dissipated by air cooling, water cooling becomes necessary. Thrust bearings are more frequently water-cooled than line bearings. This can be done by circulating water or other cooling liquid through a jacket surrounding the bearing (Fig. 1-59) or by cooling the oil in a forced-feed oil-lubricated bearing. Water jackets are most generally used with anti-friction bearings. When pumps handle hot liquids, heat from them may be transmitted to the bearings and their housings by radiation and by conduction along the shaft, thus adding to the cooling problem. There are a few installations where pump bearings are cooled with air forced by fans through ducts in the bearing housings.

Question 1-79: What is an integral-bearing bracket and what are its advantages?

Answer: An integral-bearing bracket is one that is cast onto the pump casing. Since the bracket is bored and machined at the same time as the casing, correct alignment is assured. The bracket is usually designed so that it collects leakage from the stuffing box.

Question 1-80: What are pedestal bearings and when are they used for centrifugal pumps?

Answer: Pedestal bearings have their housings supported on a flat surface, generally a pad on a bedplate or a sole plate in the foundation. They are best suited for radial loads and were used quite extensively on earlier centrifugal pumps. They have been generally superseded by bracket bearings, because of easier alignment and the increased thrust loads resulting from higher head application. Pedestal bearings are still used on centrifugal pumps to support a separate jackshaft or an extension of the impeller shaft on some belt-driven and some large pumps.

Question 1-81: If a centrifugal pump is fitted with ball bearings, should its driver have ball bearings or can sleeve bearings be used?

Answer: It is sometimes argued that both a centrifugal pump and its driver should have the same type of bearings. The basis of this claim is that, if one has anti-friction and the other babbitt bearings, the babbitt bearings wear and throw the two shafts out of line. However, if a pump and its driver both have babbitt bearings, wear in their bearings is unequal, and realignment or rebabbitting is periodically necessary.

Well-designed, properly lubricated babbitt bearings wear very slowly, and the flexible coupling between the pump and driver shafts compensates for the resulting slight misalignment for a considerable period. Regardless of the type of bearings on driver and pump, check the alignment periodically and correct it if necessary. It is immaterial whether bearings are similar.

Question 1-82: Why are thrust bearings used on double-suction pumps?

Answer: A double-suction pump theoretically should be in axial hydraulic balance, but it rarely is. A pump is primarily a product of a foundry, and minor irregularities result that may cause differences in eddy currents around the impeller that produce an axial hydraulic thrust. Conditions in the suction piping may cause hydraulic unbalance of the impeller of the pump. A thrust bearing to take thrust in either direction is therefore necessary to keep the impeller in its proper position.

Question 1-83: What is axial hydraulic thrust?

Answer: It is the summation of unbalanced forces acting on an impeller in an axial direction. Single-suction impellers, if wearing rings are not provided in the back, are subject to an axial thrust because they expose more back-wall surface to the discharge pressure than the opposed front wall, a portion of the front wall being exposed to the suction pressure.

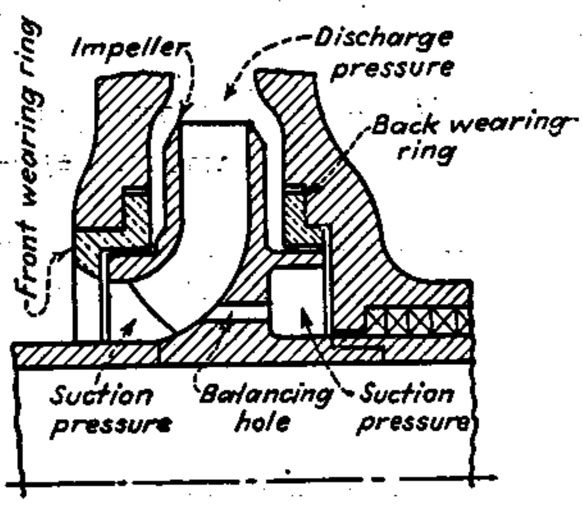


FIG. 1-60. A single-suction impeller with leakage joints of equal diameter on both sides and with balancing holes to obtain hydraulic balance.

Question 1-84: How can axial thrust be balanced in a single-suction pump?

Answer: Balance can be achieved by providing wearing rings of the same inside diameter on the suction and back sides of the

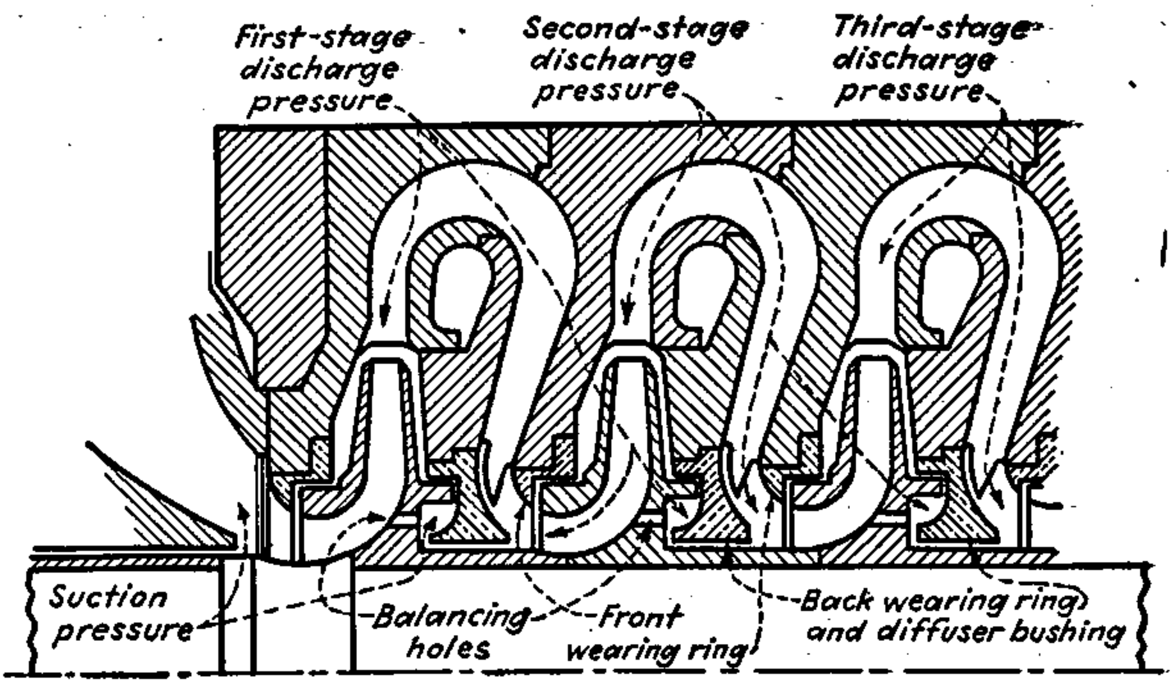


FIG. 1-61. Multistage pump design incorporating impeller design shown in Fig. 1-60.

impeller and by having balancing holes through the latter (Figs. 1-60 and 1-61). In multistage pumps, all impellers may face in the same direction and the pump be provided with a hydraulic balancing device (Fig. 1-62). As an alternative, an even number of impellers may be used and divided into two equal groups. The two groups face in opposite directions (Fig. 1-63), so that the axial thrust of one is practically balanced by that of the other.

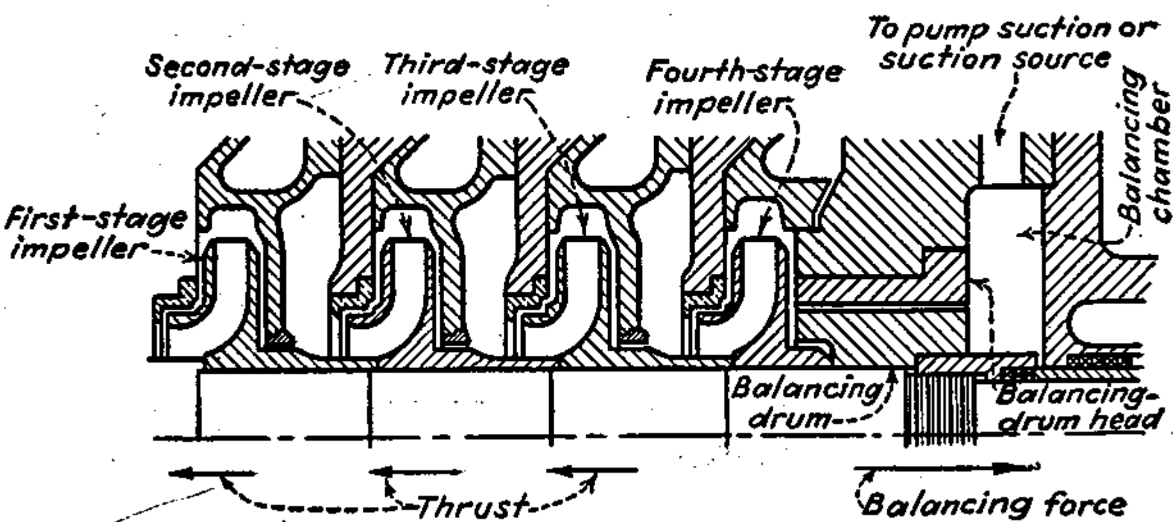


FIG. 1-62. Partial section of multistage pump with balancing drum to compensate for axial thrust resulting from unbalanced impellers.

Question 1-85: What are the most common arrangements of impellers in multistage pumps?

Answer:

1. Several single-suction impellers may be mounted on one shaft, each having its suction facing in the same direction and the stages following in the ascending order (Fig. 1-62). Such a pump requires a hydraulic balancing device.
2. An even number of single-suction impellers are mounted on one shaft, one half facing in one direction and the other half facing in the opposite (Fig. 1-63).

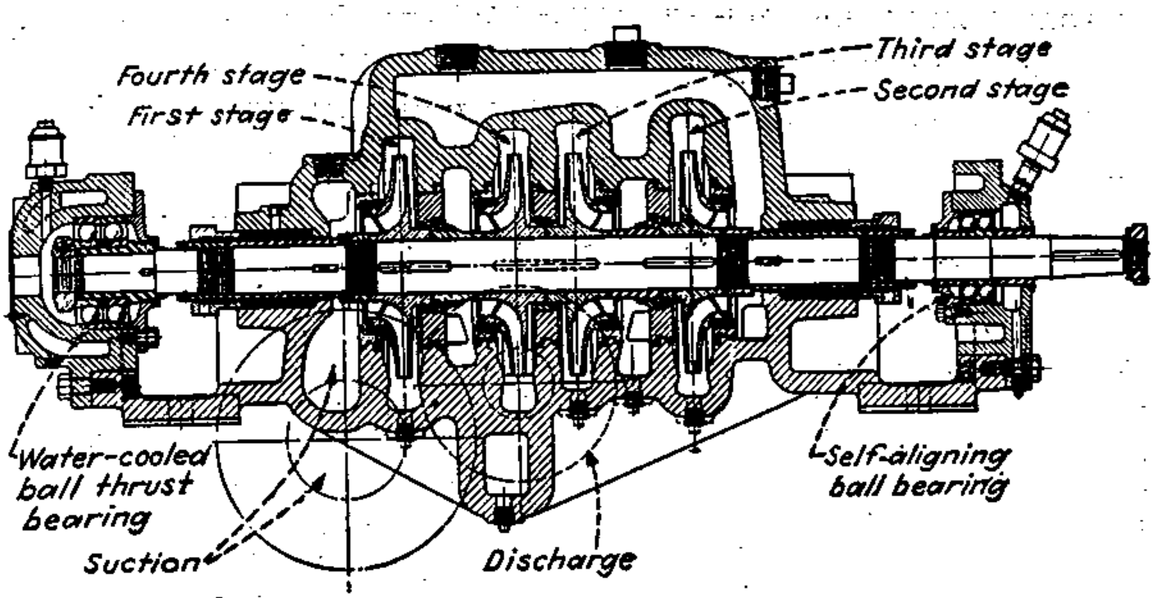


FIG. 1-63. Four-stage pump with two of its single-suction impellers facing opposite the other two to give hydraulic balance.

3. An uneven number of single-suction impellers are used, and shaft and interstage bushing diameters are chosen to give the effect of a hydraulic balancing device to compensate the hydraulic thrust of one stage.
4. Any number of double-suction impellers may be mounted on one shaft.

Question 1-86: What are a balancing drum and a balancing-drum head?

Answer: The rotating part of a radial-clearance hydraulic balancing device (Fig. 1-62) is a balancing drum, and its stationary part is a balancing-drum head.

Question 1-87: How does a balancing drum balance axial thrust?

Answer: The balancing chamber beyond the last-stage impeller (Fig. 1-62) is separated from the pump interior by the balancing drum and drum head. The balancing chamber connects either to the pump suction or to the suction source. Therefore, the balancing-chamber pressure is practically that on the suction, with the discharge pressure acting on the opposite drum face. Then, by proper design a balancing-drum diameter is selected to balance the axial hydraulic thrust completely or to within 90 to 95 per cent, depending on whether the thrust bearing is to carry a portion of the axial thrust.

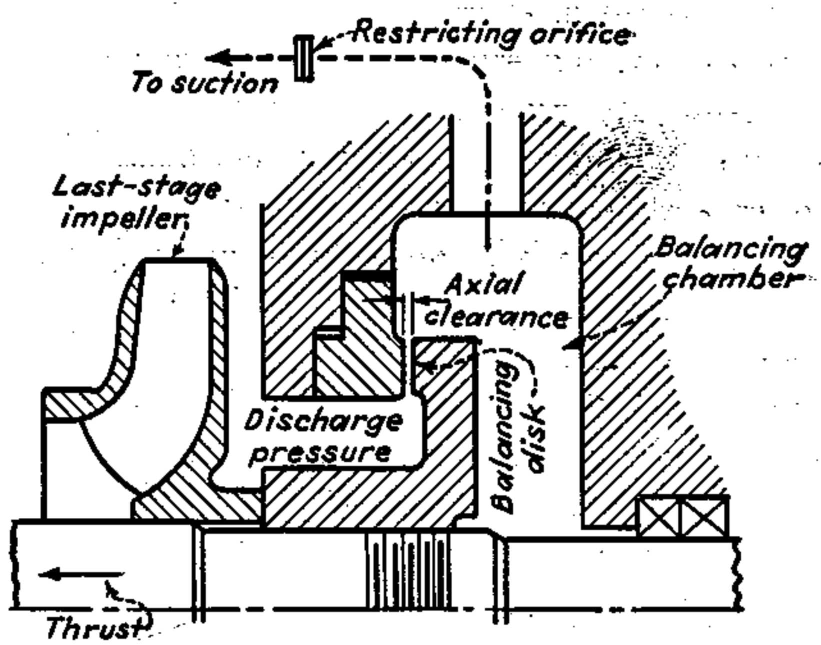


FIG. 1-64. Balancing-disk arrangement.

Question 1-88: What are a balancing disk and a balancing-disk head?

Answer: The rotating part of an axial-clearance hydraulic balancing device (Fig. 1-64) is a balancing disk; the stationary part is a balancing-disk head.

Question 1-89: How does a balancing disk balance axial thrust?

Answer: The balancing disk is fixed to and rotates with the shaft, being separated from the balancing-disk head by a small axial clearance. Leakage through this clearance flows into the balancing chamber and to the pump suction or suction source. Balancing-chamber pressure exists on the back of the balancing disk, while its face is acted upon by an intermediate pressure between the discharge pressure at the smallest diameter of the disk and the balancing-chamber pressure at its periphery. Inner and outer disk diameters are so chosen that the difference between the force acting on the face of the disk and that on its back balances the axial thrust of the impellers.

If the axial thrust of the pump increases, the excess pressure moves the rotor to reduce the clearance between the balancing disk and the disk head. This reduces the flow through the clearance and lowers balancing-chamber pressure until the pump axial thrust is again balanced. Similarly, a reduction in axial thrust causes an increase in clearance between the disk and the disk head. As a result of increased flow, pressure increases in the balancing chamber until the axial hydraulic thrust is again balanced.

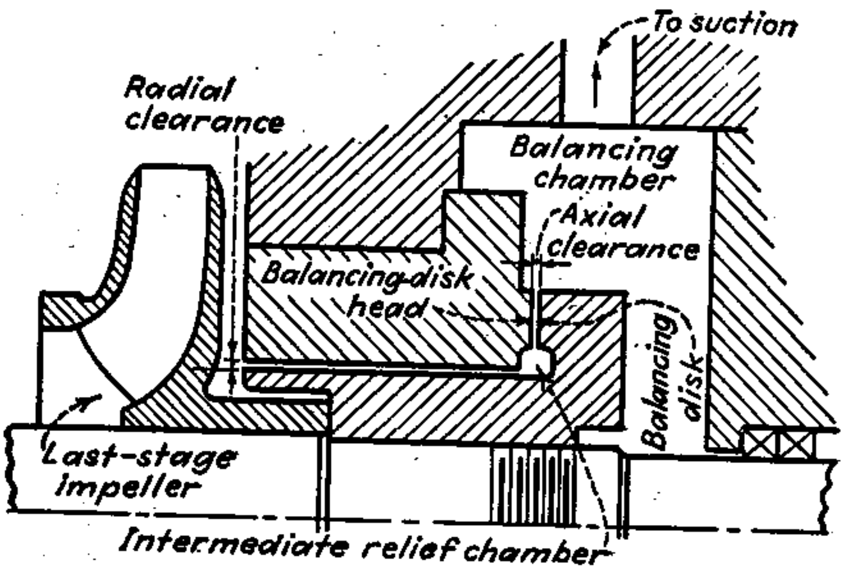


FIG. 1-65. Combination balancing disk and drum arrangement.

Question 1-90: What is a combination balancing disk and drum?

Answer: In modern practice, a simple balancing disk is not commonly used. A combination device with both axial and radial clearances is employed (Fig. 1-65). Such a device is called a "combination balancing disk and drum."

Question 1-91: What types of couplings connect centrifugal pumps to their power units?

Answer: Flexible couplings are most commonly used, but practically all types of couplings and clutches have been applied.

Question 1-92: What is a rigid coupling?

Answer: It is a coupling that solidly connects the driving and the driven shafts, making of them, in effect, a single shaft. These couplings are used mainly on vertical pumps and generally must be capable of transmitting torque and thrust (Fig. 1-66).

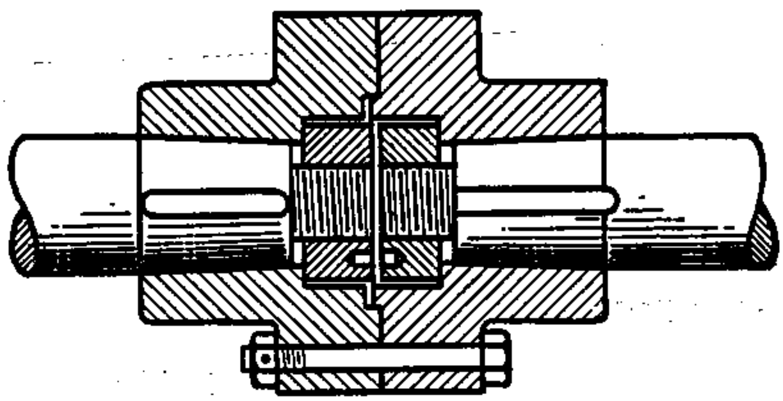


FIG. 1-66. Flanged rigid-type coupling.

Question 1-93: What is a clamp coupling?

Answer: It is a rigid coupling that consists of a split sleeve with bolts so that it can be clamped on the adjoining ends of two shafts to form a solid connection. Axial and circular keys are commonly used, so that transmission of torque and thrust does not depend solely on frictional grip.

Question 1-94: What is a compression coupling?

Answer: Compression couplings are seldom, if ever, used on modern horizontal centrifugal pumps. Their use is normally restricted to vertical sump pumps. A compression coupling is essentially a rigid coupling with its central portion a slotted bushing bored to fit the two shafts and tapered on the outside from the center to both ends. The two coupling halves are bored to suit this taper.

When bolted together, they compress the bushing to the shafts with a frictional grip that transmits the torque without keys.

Question 1-95: What is a flexible coupling?

Answer: A flexible coupling transmits torque from one shaft to another as a rigid coupling does but is of a design that permits

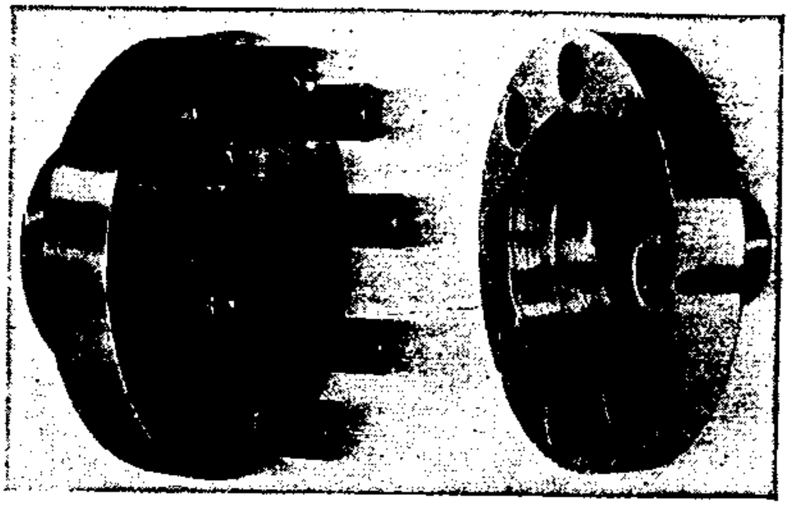


FIG. 1-67. Flexible coupling of pin and rubber-buffer type.

minor misalignment (angular, parallel, or both) of the shafts. Figures 1-67 to 1-70 show four of many designs, and the captions describe them.

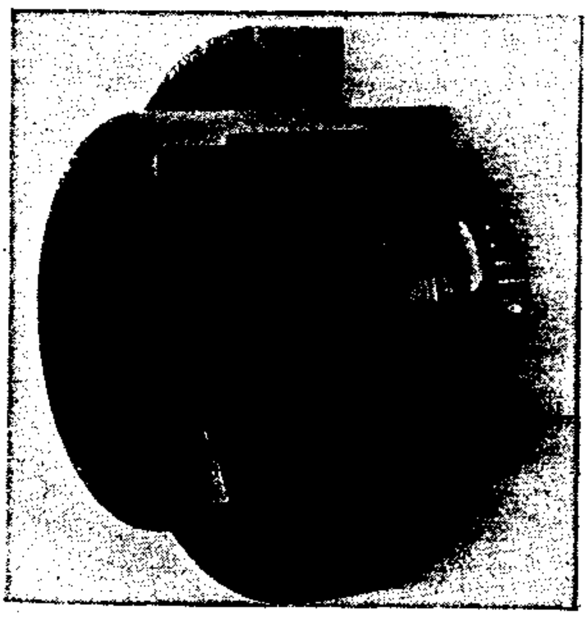


FIG. 1-68.

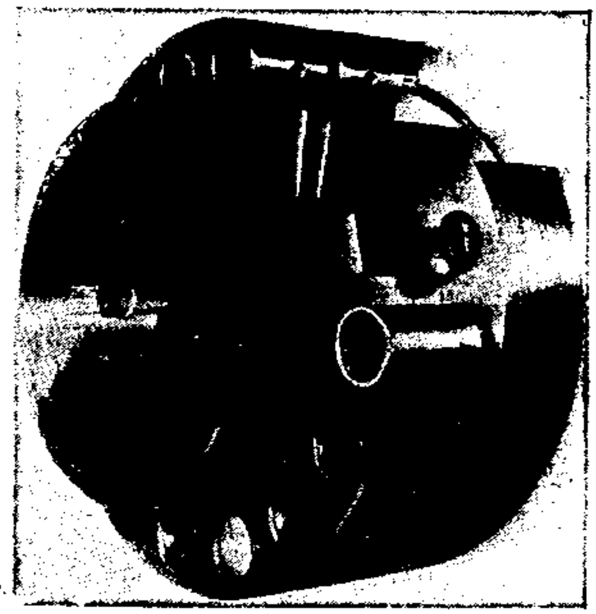


FIG. 1-69.

FIG. 1-68. Flexible coupling of the internal-external gear type.
FIG. 1-69. Flexible coupling of the flexible-metal-pin type.

Question 1-96: What is an extension coupling?

Answer: It is a flexible coupling consisting of two single-engagement elements connected by a sleeve (Fig. 1-71). This gives, in effect, the equivalent of a short floating shaft drive that compensates for greater misalignment than a regular flexible coupling. Extension couplings are commonly used on pumps handling hot liquids when expansion may cause misalignment. They are also used on end-suction pumps in which the impeller and bearing assembly are removed axially toward the driver.

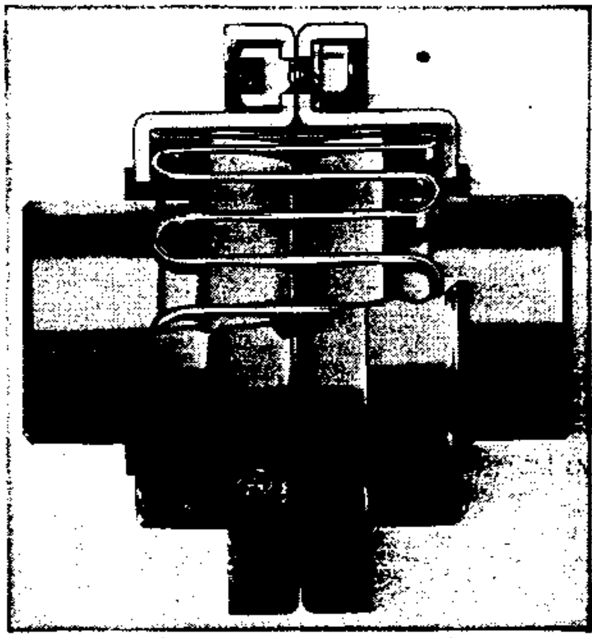


FIG. 1-70. Flexible coupling of the steel-spring-grid type.

Question 1-97: What types of couplings are used in dual-driven pumps?

Answer: On dual-driven pumps, it is generally desirable to have one driver idle when the other is in operation, to save power as well

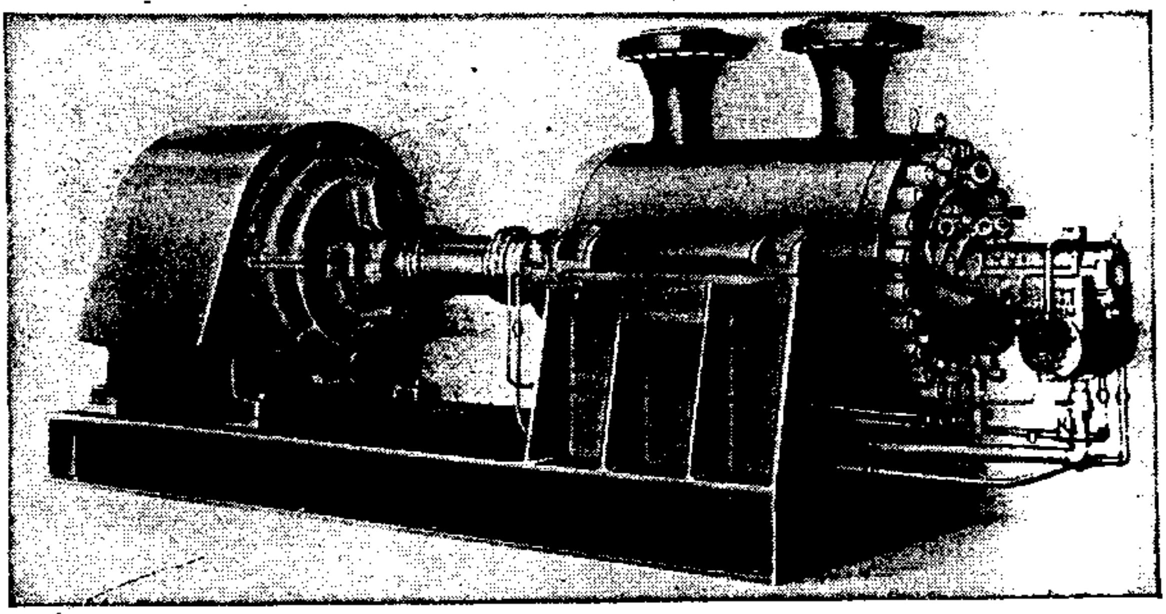


FIG. 1-71. Boiler-feed pump driven by a motor through an extension coupling.

as wear on the idle machine. Some design of quick-disconnect coupling can be used for this purpose. The ideal coupling for dual drives is one that can be readily disengaged and engaged. When the stand-by driver starts automatically, a freewheeling or overrunning clutch is necessary.

Question 1-98: Are clutches used between centrifugal pumps and their drivers?

Answer: Conventional designs of clutches are rarely used, because most of them seriously increase the thrust load on the thrust bearing of the pump, and they require an accurate alignment between clutch parts that may be difficult to maintain. Overrunning clutches are used particularly for dual drives. The most successful designs have a flexible coupling built in as part of the clutch unit. Automatic clutch couplings that have their friction elements applied radially by centrifugal force are also used.

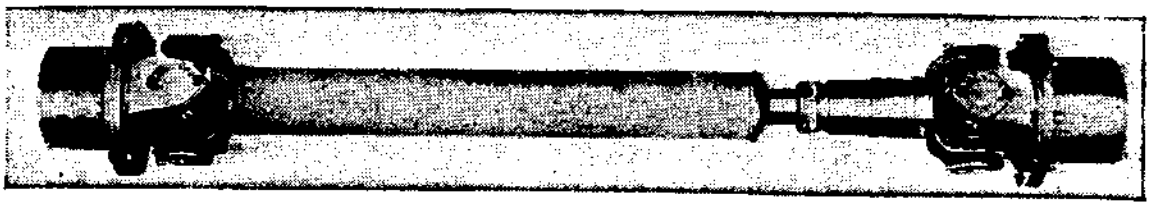


FIG. 1-72. Flexible drive combining two universal joints for angular misalignment and a splined section to compensate for variation in length.

Question 1-99: What is a flexible drive?

Answer: In some installations, to accommodate considerable misalignment between pump and driver shafts, two universal joints or flexible couplings are joined by an unsupported shaft or torque tube. One design using universal joints provides for changes in distance between the shaft ends and is called a "flexible drive" (Fig. 1-72).

Question 1-100: Are magnetic clutches used with centrifugal pumps?

Answer: They are rarely used to connect centrifugal pumps to and disconnect them from their drivers. One application where they have been used is on pumps discharging into accumulator tanks or for similar service where the demand varies over a wide range. It is now general practice to start and stop the entire pumping unit or, if the cycle is too frequent for that, to allow the pump to operate at reduced capacity during the period of small demand. If the demand will at times become too small for proper operation, a by-pass is provided so that the pump discharge capacity never falls below a safe value.

Question 1-101: Are hydraulic couplings and magnetic drives used with centrifugal pumps?

Answer: Both are used with centrifugal pumps where operating conditions warrant changing the output speed to control the pump discharge rate. They have about the same over-all efficiency as wound-rotor motors with speed control and can give any desired output speed, while the control of a wound-rotor motor gives only steps in speed.

Question 1-102: What are sole plates and when are they used?

Answer: Sole plates are cast-iron or steel pads located under the feet of a pump or its driver, or both. They are embedded in the foundation, and the pump and its driver are doweled and bolted to them. Sole plates are generally used for some of the larger units to save the cost of the large bedplates that would be required for such units.

Question 1-103: What is the purpose of a bedplate under a pump and its driver?

Answer: The purpose is to provide a foundation on which they may be aligned with each other and on which they can be assembled as a single unit for moving from one location to another, and to simplify their installation. While bedplates are fairly rigid, they do not provide sufficient support and must be bolted and grouted to a rigid foundation after being properly leveled. Because of handling during transit and erection, bedplates sometimes become distorted, which makes it necessary to check the alignment of the unit after it is installed.

Question 1-104: When is it undesirable to provide a bedplate under the pump and its driver?

Answer: For most large pumping units, the cost of a single bedplate on which to mount and align the different parts exceeds that of the field work necessary to align individual bedplates or sole plates. In such cases, a single bedplate is used only when appearances or the need as a drip collector justifies the additional cost. These bedplates have a raised lip around their top edge and slope toward one end to collect drainage in a screened drain pocket, from which it is piped to a convenient point for disposal. Even for fairly small units, there are installations where a considerable difference in height exists between the level on which the feet of the pump are bolted and the level of those of its driver. In many such installations, a more rigid unit and one of more pleasing appearance can be obtained by using individual bases or sole plates and building up the foundation to various heights as required for the different parts of the equipment.

Question 1-105: When must the pump and the bedplate construction be made to give center-line support?

Answer: When liquid temperatures are high, it is necessary to support the pump casing as nearly as possible to its horizontal center line (Fig. 1-71) to prevent temperature differences from seriously disturbing the alignment of the unit and eventually damaging the pumping equipment. This construction is generally employed for refinery pumps handling high-temperature hydrocarbons and for heater drain and boiler feed pumps operating at temperatures of 300°F and higher. Some pump manufacturers provide center-line support for all boiler feed centrifugal pumps, regardless of operating temperature.

Question 1-106: What is meant by "pump setting"?

Answer: Except for vertical turbine pumps (see page 172), it usually means the distance from the supporting floor to the suction bellmouth of vertical wet-pit pumps. This term is also used in a less

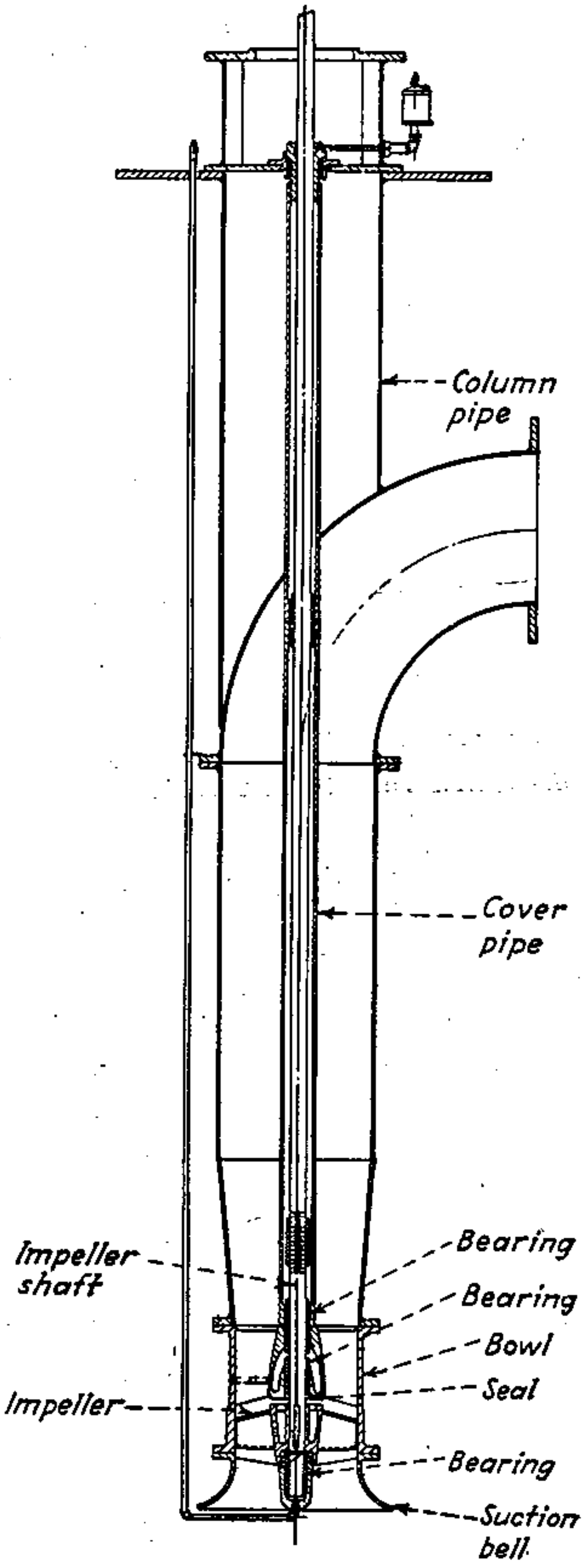


FIG. 1-73. Typical section of vertical wet-pit propeller pump with axial-flow impeller.

definite manner with vertical dry-pit pumps, where it refers to the distance from the floor supporting the driver to either the floor supporting the pump or the suction opening on the pump.

Question 1-107: What is a pump bowl?

Answer: In vertical turbine-pump and propeller-pump practice, the bowl is the stationary part containing the diffusion vanes (Fig. 1-73).

Question 1-108: What is a pump-bowl assembly?

Answer: "Pump-bowl assembly," "working barrel," and "pumping element" are all used interchangeably to mean the assembly of all impellers and pump bowls, together with their rings and bearings and those sections of the shafting on which the impellers are mounted, in vertical turbine or vertical propeller pumps (Fig. 1-73).

Question 1-109: What is a discharge column?

Answer: The discharge column in a vertical turbine or vertical propeller pump is the pipe and complete shafting between the pump-bowl assembly and the pump head or motor stand. It acts both as the supporting column for the pump-bowl assembly and as a pipe to carry the water to where it is discharged (Fig. 1-73).

Question 1-110: What is a column pipe?

Answer: This column suspends a wet-pit pump from its supporting base. With propeller and vertical turbine pumps, the column pipe acts as the discharge pipe and is also called the "discharge column." The column pipe may also be called a "drop pipe."

The terminology used for vertical wet-pit centrifugal pumps has been that used in vertical deep well turbine-pump practice. That type of centrifugal pump, covered in Chap. 10, has been widely applied for open-pit installations and is now generally called "vertical turbine pump" type.

CHAPTER 2

CENTRIFUGAL-PUMP CLASSIFICATIONS

Question 2-1: What is a general-service pump?

Answer: A number of services use centrifugal pumps of the same construction. For this broad application, they are called "general-service pumps."

Question 2-2: What is a booster pump?

Answer: Any pump that operates in series with another to increase the head or to increase pressure in part of a system is generally called a "booster pump." Such a pump has numerous applications, the most common being when water is available under insufficient pressure and a pump is added to boost the pressure to that needed. In another common application, the required head has been increased to where the existing pump or pumps cannot be used. Accordingly, a booster pump or pumps is added in series to provide the additional head. In some high head pumping installations, suction conditions do not permit a pump to operate at the best speed for the head and capacity required. To eliminate suction limitations, a low-speed, low head pump is installed to take the liquid from the suction supply and discharge it to a high-speed efficient pump that develops the major portion of the total head. In such installations, the low-speed, low head pump is a "suction booster pump."

Generally, a booster pump develops considerably less head than the main pump. However, "booster" is often applied to pumps in intermediate stations of pipe lines, even though they develop the same or greater head than those in the first station.

Question 2-3: What location is preferred for a booster pump?

Answer: When a booster pump has to increase the pressure of a high-pressure pump, it is preferably installed as a suction booster.

If it were installed on the high-pressure pump discharge, its casing and stuffing box would be subjected to high pressure. In many installations this would result in high cost and a serious stuffing-box maintenance problem.

Question 2-4: What is a circulating pump?

Answer: In the past the term was understood to cover a pump to supply cooling water to a surface condenser. So many other applications now exist in which water or other liquids are circulated that this name no longer can be properly used for a single service. It must be qualified by giving the service for which the pump is used, like "condenser circulating pump," "hot-water circulating pump," etc.

Question 2-5: What is a boiler feed pump?

Answer: It is used in steam-generating power plants to deliver feed water to the boiler. Depending upon the feed cycle, the boiler feed pump may take its suction from a condensate-pump discharge, a deaerating heater, or, in small plants, directly from the make-up source external to the feed cycle. A boiler feed pump generally handles water at a temperature of 212°F or above.

Question 2-6: What is meant by a hotwell pump?

Answer: A pump to take condensate from the hotwell of a condenser, a stage heater, or an evaporator and deliver it into the boiler feed system may be called a "hotwell pump." In practice, however, the term has been restricted to condenser hotwell service, other applications carrying their own designation.

Question 2-7: What is a condensate pump?

Answer: It handles condensate and, as such, the term could be applied to any pump in a steam power plant cycle. Its use, however, is restricted to a centrifugal pump to remove condensate from a condenser. Sometimes any hotwell pump is called a condensate pump, but this is not exactly correct in the restrictive use of the term.

Question 2-8: What is a tail pump?

Answer: With low-level jet condensers, a pump is required to take water out of the condenser against a head that equals the existing vacuum plus any external head. These pumps are called "tail pumps" and are usually of special design, mechanically and hydraulically.

Question 2-9: What is a heater-drain pump?

Answer: It removes condensate from a closed-heater hotwell. Like all hotwell pumps, it handles liquid at a pressure equivalent to its vapor pressure. Unlike condensate-removal pumps, however, the pumping temperature of a heater-drain pump may be as high as 350 or 400°F, with a corresponding suction pressure, requiring a suitable stuffing-box design.

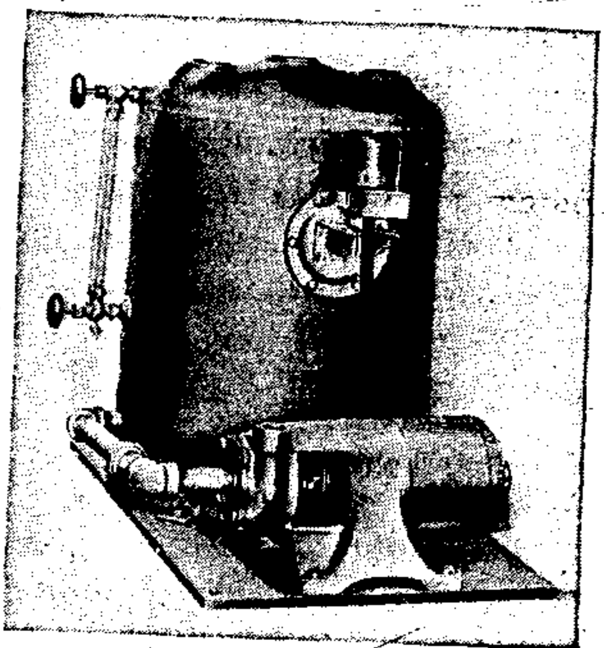


FIG. 2-1. Condensate return unit.

Question 2-10: What is a condensate-return unit?

Answer: The unit is a pump combined with a tank or reservoir and automatic controls (Fig. 2-1). It is installed so that condensate from various sources, such as a building heating or process system, can drain to the storage tank. When sufficient condensate has accumulated, the control automatically starts the pump, which discharges the condensate to a boiler or to some part of the boiler feed system.

Question 2-11: What is a fire pump?

Answer: A pump installed solely to provide water to fight fire, and which is not used for other services, is called a "fire pump." It can be either a regular commercial pump or a specially designed pump approved by the Underwriters.

Question 2-12: What is an Underwriters' fire pump?

Answer: The National Board of Fire Underwriters and the Associated Factory Mutual Fire Insurance Companies have established specifications covering the mechanical design and hydraulic performance features for fire pumps. Pumps built to meet these specifications and approved by the Underwriters are called "Underwriters' fire pumps" (Fig. 2-2).

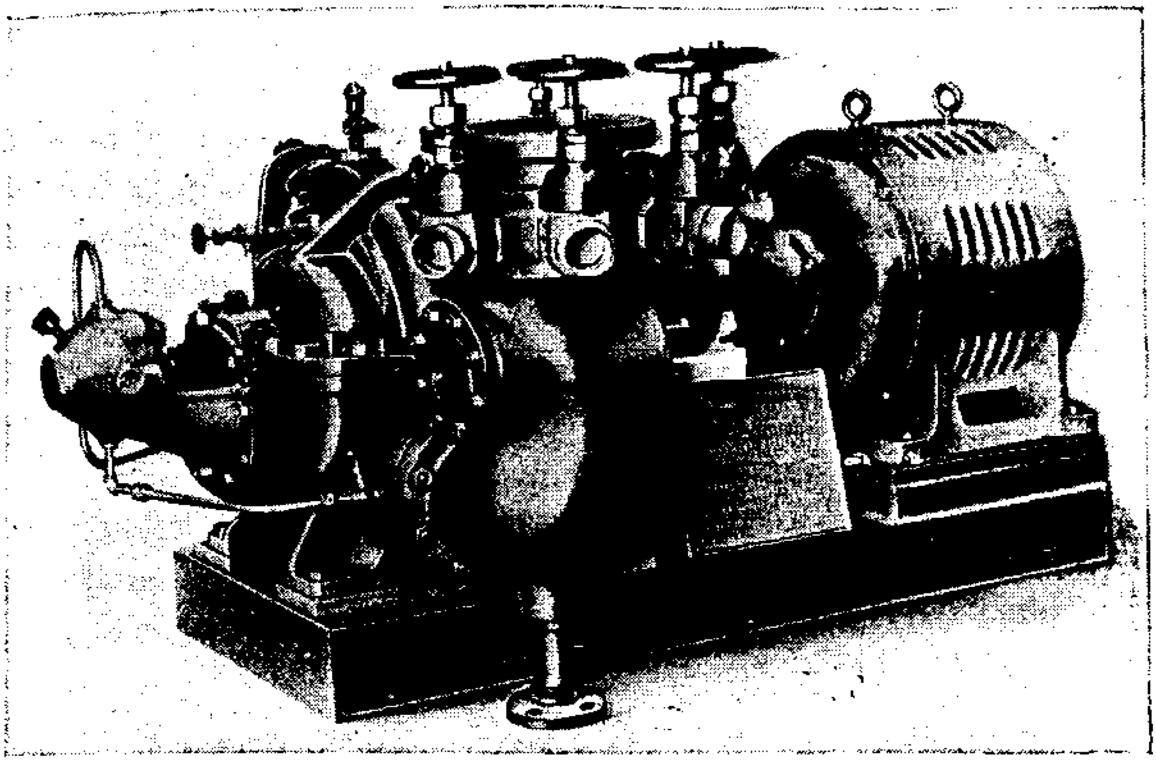


FIG. 2-2. Underwriters' fire pump with hose gates and other Underwriters' fittings.

Question 2-13: What are Underwriters' fittings?

Answer: They are accessories that the Underwriters require for use with fire pumps before the installation can be approved. These include pressure gauges, air and starting valves, manifold with hose gates, relief valve, etc. Local conditions dictate what fittings must be included to have the installation approved.

Question 2-14: What is a trash pump?

Answer: A non-clogging sewage pump was at one time called a "trash pump." The term is rarely used now.

Question 2-15: What is a non-clogging pump?

Answer: It has comparatively large openings between vanes of its impeller, which are well rounded at their entrance ends, to prevent clogging with strings, rags, and solids when handling sewage or other liquids containing matter that tends to clog the impeller. An ordinary centrifugal-pump impeller has comparatively small passages between its vanes, which have sharp entrance ends and are easily clogged when handling such liquids as sewage.

Question 2-16: What is a sludge pump?

Answer: Pumps are required in sewage and some water-purification plants and for some industrial processes to pump sludges of various densities. Most of them require pumps of special design called "sludge pumps." Many sludges are too dense to be handled by centrifugal pumps.

Question 2-17: What is a dry-pit pump?

Answer: Some vertical pumps are mounted in a dry underground chamber, generally called a "dry pit." Vertical pumps installed in such chambers or under similar physical conditions are called "dry-pit pumps." They operate in air and are accessible at all times for servicing.

Question 2-18: What is a wet-pit pump?

Answer: Vertical pumps designed for suspension in or for mounting in their suction supply, like a sump or wet pit, are regularly called "wet-pit pumps." A sump pump is the most common type of wet-pit pump.

Question 2-19: What is a sump pump?

Answer: In centrifugal-pump terminology, "sump pump" generally applies to vertical wet-pit pumps suspended in a sump or wet well where drainage collects (Fig. 2-3). Sump pumps are generally automatically controlled by float switches. Small sump pumps to take seepage out of cellars in homes and small buildings are usually called "cellar drainers."

Question 2-20: What is a propeller pump?

Answer: The term was originally applied to pumps using axial-flow impellers, particularly those with diffusion vanes. The name is now applied to diffusion-vane pumps with both axial-flow and high-speed mixed-flow impellers. Since most installations of pro-

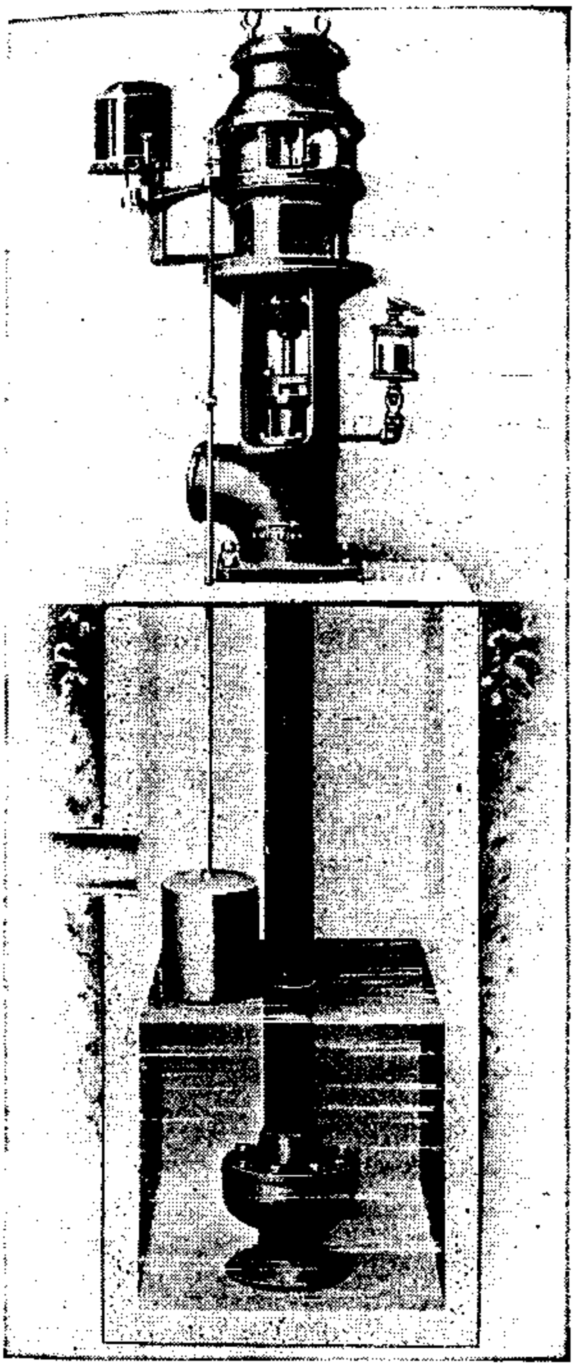


FIG. 2-3.

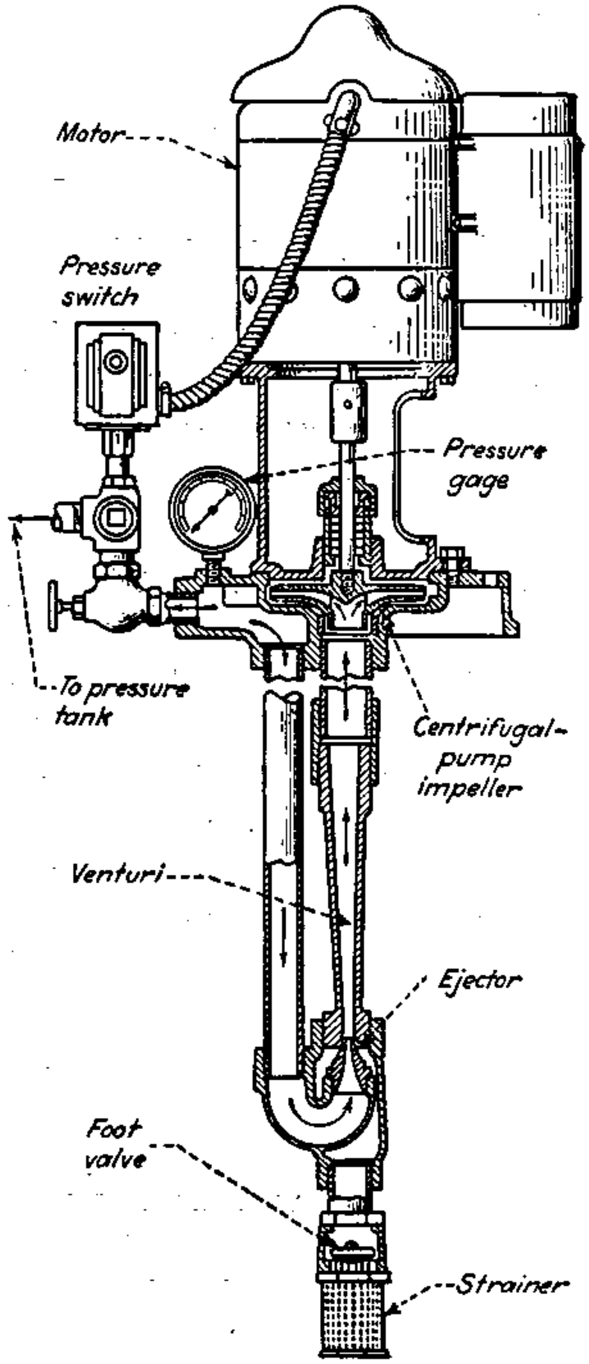


FIG. 2-4.

FIG. 2-3. A typical sump-pump installation.
FIG. 2-4. Ejector-type well pump.

peller pumps are vertical wet-pit applications as shown in Fig. 1-72, the term generally suggests such an installation. However, such pumps are also installed horizontally and are called "horizontal propeller pumps."

Question 2-21: What is a vertical turbine pump?

Answer: It is a vertical-shaft diffuser centrifugal pump of single-stage or multistage design. It is suspended from a driving head by a column pipe that conducts the water to the discharge level and also supports the shaft and bearings. This pump is primarily designed for installation in bored wells. (See Chap. 10.)

Question 2-22: What is a sinker pump?

Answer: It is used to dewater mine shafts and similar excavations that have become flooded. This pump is usually supported on a hoist and is lowered as the water level falls.

Question 2-23: What is an ejector well pump?

Answer: In an ejector type of well pump, such as illustrated in Fig. 2-4, the ejector is suspended below the water level in the well. Power water is supplied to the ejector by a pipe from the surface and is discharged with the water it ejects from the well up another pipe into the suction of the centrifugal pump.

The flow from the centrifugal pump is divided, whatever amount required going to the ejector, with the excess going into the system or the storage tank.

A foot valve in the tail pipe below the ejector is necessary so that the water in the pump and piping will not drain out when the unit is shut down. With pneumatic systems, some device for getting air into the tank must be incorporated.

Question 2-24: What is a close-coupled pump?

Answer: It is built with a common shaft and bearings for the pump and driver to form a single compact unit (Fig. 2-5). Some vertical-shaft pump designs have the driver supported by the pump with their shafts connected by a flexible coupling. There is no established terminology for such a design. Some manufacturers call them "pedestal-mounted" units, whereas others also call them "close-coupled" units.

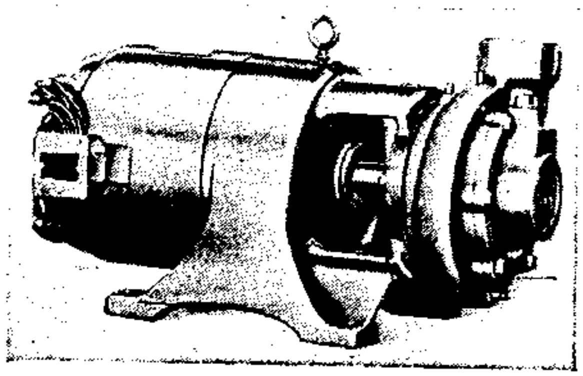


FIG. 2-5. A close-coupled motor-driven pump.

Question 2-25: What is a side-suction pump?

Answer: When the suction nozzle is placed on the side of the pump casing with its axial center line at right angles to the vertical center line of the pump, the pump is classified as a side-suction pump (Fig. 2-6).

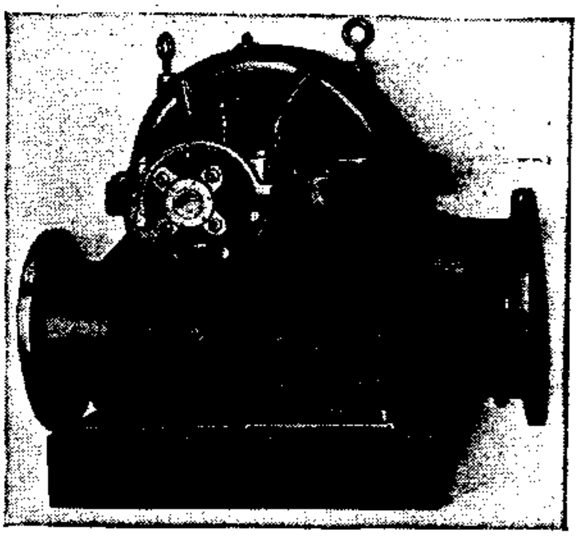


FIG. 2-6. End view of double-suction pump with side suction.

Question 2-26: What is a bottom-suction pump?

Answer: When the suction nozzle of a pump points vertically downward, the pump is called a "bottom-suction pump" (Fig. 2-7). Single-stage bottom-suction pumps are rarely made in sizes below 10-in. discharge size.

Question 2-27: What is an end-suction pump?

Answer: It has a suction-nozzle axis that coincides with that of the shaft or is parallel to it and on the outboard end of the pump

FIG. 2-7. View of double-suction pump with bottom suction.

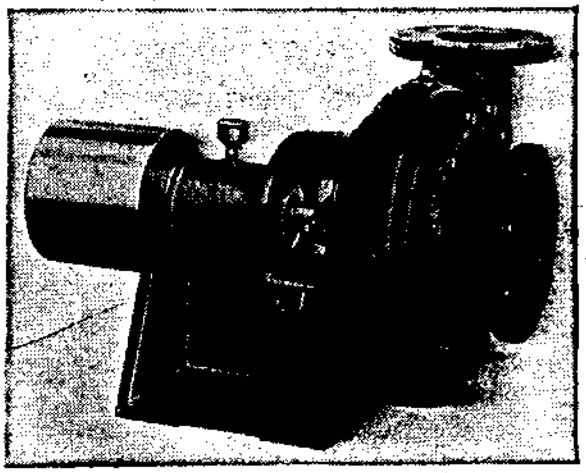


FIG. 2-8. End-suction pump (see Fig. 1-15 for section).

(Fig. 2-8). These pumps have their two bearings on their inboard side.

Question 2-28: How is the rotation of centrifugal pumps designated?

Answer: According to the Hydraulic Institute Standards, rotation is described as clockwise or counterclockwise when looking at the driven end of a horizontal pump, and down on a vertical pump. Some manufacturers, however, still designate the rotation of horizontal pumps as it appears looking at the outboard end. Therefore, to avoid misunderstanding, "clockwise" or "counterclockwise" should always be qualified by the direction from which one looks at the pump.

Question 2-29: What are meant by the inboard and outboard ends of a pump?

Answer: "Inboard end" and "outboard end" are terms applied to horizontal pumps. The inboard end is the one closest to the driver, while the outboard end is the one farthest away. In the case of dual-driven pumps, the terms lose their significance and are not used.

Question 2-30: What is meant by inboard and outboard bearings?

Answer: On horizontal pumps that have a bearing at each end, the bearing at the outboard end is known as the outboard bearing. Overhung impellers have both bearings on the same side of the impeller. On these, the bearing nearest the impeller is the inboard and the one farthest away is the outboard. Generally, on a double-suction pump with bearings on both sides, the thrust bearing is placed at its outboard end and the line bearing at the inboard end.

CHAPTER 3

MATERIALS USED IN CENTRIFUGAL PUMPS

Question 3-1: What determines the kind of materials in centrifugal-pump construction?

Answer: Centrifugal pumps are manufactured of almost all known common metals or metal alloys, as well as of carbon, porcelain, glass, stoneware, hard rubber, and even synthetics. Service conditions and the nature of the fluid pumped determine the most satisfactory materials. Some of the factors that enter into this selection are:

1. Abrasiveness of suspended solids.
2. Corrosion resistance.
3. Electrochemical action.
4. Liquid temperature.
5. Head per stage.
6. Discharge pressure.
7. Suitability of material to the structural features of the particular pump involved.
8. Load factor and expected duration of pumping installation.

Question 3-2: Why is there a preference for bronze pump parts, other than the pump casing?

Answer: Bronze does not rust, and hence replacements of threaded parts are easier. Bronze parts do not seize in pumps held idle. Because bronze is easier to machine than iron and smoother surfaces can be secured at less cost, higher efficiencies are theoretically possible with bronze parts. However, bronze has limitations that frequently make it necessary to forego its advantages.

Question 3-3: What materials are used for pump casings?

Answer: Generally, centrifugal-pump casings are of cast iron, but sometimes alloy irons are used to obtain a more refined crystalline structure. Bronze is frequently used for pump casings when mildly corrosive liquids are handled, as, for instance, sea water. When the pressure or temperature, or both, of the liquids is so high that cast iron is no longer reliable, cast steel or even forged steel may be used for the casing. Stainless-steel casings are used when the pumped liquid is highly active chemically. Under some special conditions it may even be necessary to use a porcelain, stoneware, hard-rubber, carbon, or glass casing.

Question 3-4: What materials are most commonly used for pump impellers?

Answer: Bronze impellers are generally preferred when handling average waters, because bronze does not rust. Casting it in complicated cored sections, machining it, and making its surfaces smooth are easy. However, bronze impellers in cast-iron casings are undesirable if the liquid is a strong electrolyte; therefore, iron, cast steel, or even stainless steel may be used. The latter two, although more costly than cast iron, have several advantages. Because bronze impellers expand considerably more with heat than their steel shafts, cast-steel or stainless-steel impellers are used extensively if water temperatures exceed about 250°F. Sometimes monel impellers are preferred to bronze when handling sea water.

Question 3-5: What materials are used for wearing rings?

Answer: Bronze is the most widely used material for wearing rings as it is for impellers. Cast-iron, cast-steel, and stainless-steel rings are sometimes used, whether the impeller is of bronze or other metal. Cast steel and stainless steels are used when hardness or other properties unobtainable with bronze are required.

Whether single- or double-wearing rings are used, generally the practice is to make the stationary and rotating parts of materials having different hardnesses.

Question 3-6: What materials are used for pump shafts?

Answer: In pumps with shaft sleeves, shafts are generally of open-hearth forged steel. With some corrosive liquids, some seepage may occur either through the pores of the impeller hub or through the joint between the impeller and the sleeves. Such conditions require a noncorrosive stainless-steel, phosphor-bronze, or monel shaft. Pump shafts without sleeves are generally stainless steel, phosphor bronze, or monel, depending upon the liquid handled.

Question 3-7: What materials are used for shaft sleeves?

Answer: Most of them are made of bronze. When bronze is not satisfactory because of its relatively low resistance to abrasion, stainless steel is used. A few years ago, steel sleeves were more prevalent than stainless steel; but today stainless is superseding ordinary steel, and in some services, such as boiler feed, it is replacing even bronze as shaft-sleeve material.

A shaft sleeve that protects a shaft in a stuffing box must be made of a material that can be given a smooth finish for the packing. Thus cast iron is rarely used for a shaft sleeve.

Question 3-8: What materials are commonly used for pump glands?

Answer: Glands are generally made of bronze, although cast iron or steel may be used for all-iron-fitted pumps. Where pumps handle hydrocarbons, iron or steel glands are bushed with bronze to avoid possible sparking, which might ignite inflammable vapors (Fig. 1-47).

Question 3-9: What is meant by pump fittings?

Answer: This expression is used rather loosely with two separate meanings. In the sense generally understood, it refers to the general construction features of the pump, as, for example, "ball-bearing-fitted pump," or to the combination of materials used in the pump, as an "all-iron-fitted pump." In a different category (as when applied in the expression "Underwriter fittings") it may refer to various pieces of auxiliary equipment, such as valves, gauges, or even tools provided with the pump.

Question 3-10: What is a standard-fitted pump?

Answer: The so-called standard-fitted centrifugal pump as defined by the Hydraulic Institute is bronze fitted. (See the accompanying table and Fig. 3-1.) It has a cast-iron casing, steel shaft,

METALS AND ALLOYS FOR CENTRIFUGAL-PUMP CONSTRUCTION*

Part No.†	Parts	Standard-fitted pump	All-iron pump	All-bronze pump
1	Casing	Cast iron	Cast iron	Bronze
35	Suction head	Cast iron	Cast iron	Bronze
4	Impeller	Bronze	Cast iron	Bronze
12	Impeller ring	Bronze	Cast iron or steel	Bronze
3	Casing ring	Bronze	Cast iron	Bronze
	Diffuser	Cast iron or bronze	Cast iron	Bronze
5	Stage piece	Cast iron or bronze	Cast iron	Bronze
2	Shaft (with sleeve)	Steel	Steel	Bronze, monel, or steel
2-A	Shaft (without sleeve)	Stainless steel or steel	Stainless steel or steel	Bronze or monel
10	Shaft sleeve	Bronze	Steel or stainless steel	Bronze
15	Gland	Bronze	Cast iron	Bronze

*Materials for bearing housings, bearings, and other parts are not generally affected by the liquid handled.

†Parts are numbered according to a proposed standard listing suggested to the Hydraulic Institute by Charles J. Tullo, Worthington engineer. This proposed standard gives stationary parts odd numbers, and rotating parts even numbers (Figs. 3-1 and 3-2).

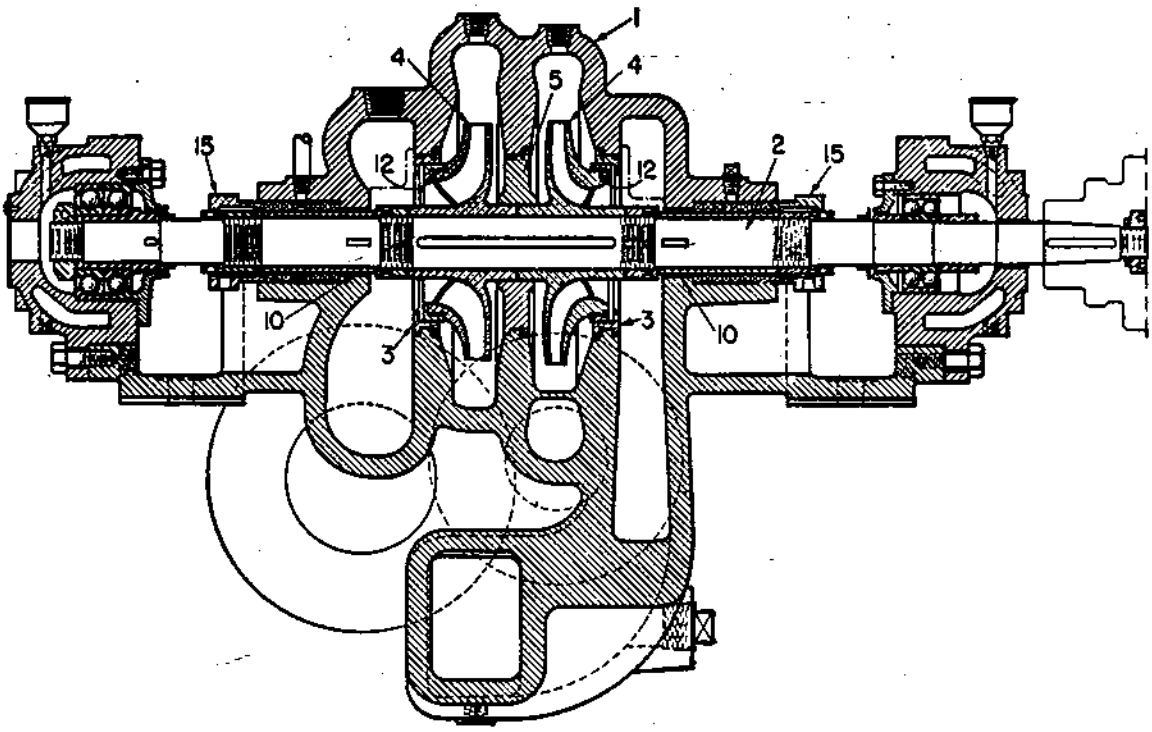


FIG. 3-1. Sectional view of a multistage closed-impeller centrifugal pump with parts identified by numbers (see text and table).

bronze impeller and wearing rings, and bronze shaft sleeves, when used. Some manufacturers regularly furnish stainless-steel shafts on pumps without shaft sleeves as standard-fitted pumps.

Various centrifugal-pump manufacturers have developed specific materials for pumps designed for special services. When a pump is so constructed it is generally termed a "standard pump," although

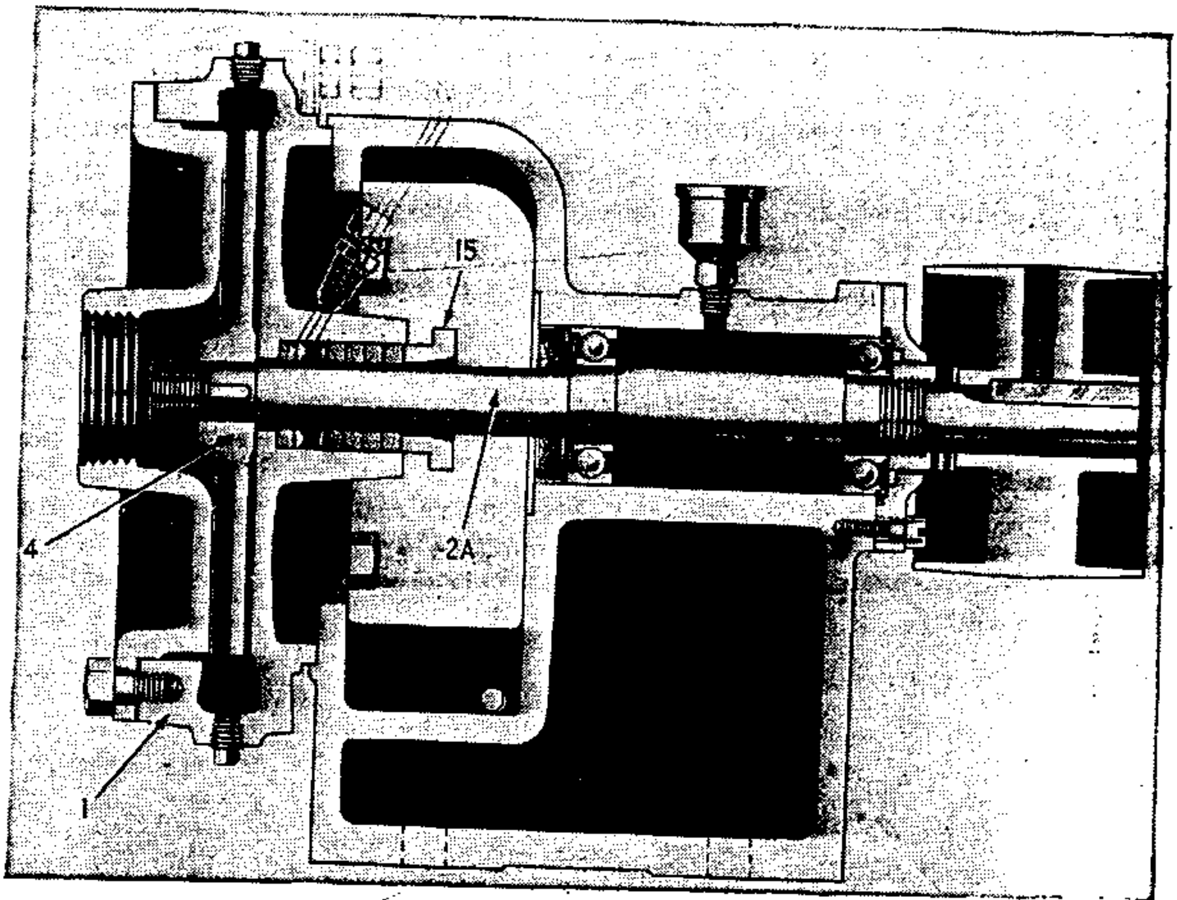


FIG. 3-2. Sectional view of an open-impeller pump with parts identified by numbers (see text and table).

reference is sometimes erroneously made to a "standard-fitted pump," even though the materials do not correspond to the Hydraulic Institute definition.

Question 3-11: What is an all-bronze pump?

Answer: When all parts of a centrifugal pump that come in contact with the liquid pumped are bronze, the pump is considered an all-bronze pump (see table on page 67).

Question 3-12: What is an acid-resisting pump?

Answer: It is a pump in which all parts contacting the pumped liquid are of materials that offer maximum resistance to the corrosive action of the liquid.

Question 3-13: What materials are used in centrifugal pumps handling sea or salt water?

Answer: Such pumps may be built with standard fittings—that is, cast-iron casings and bronze trim—all iron, all bronze, or with iron casings and stainless fittings, depending on a multitude of factors.

Thousands of standard-fitted pumps are used for sea water, but occasionally they give trouble—mostly when the sea water is contaminated, as with harbor waters. Failures are caused by galvanic action between the bronze parts and the cast-iron casing, which graphitizes and is ultimately a total loss. Under high suction-lift conditions, galvanic action is accelerated because oxygen is released from the water.

While an all-iron pump avoids the electrolytic action between bronze parts and cast-iron casings, such action may occur nevertheless. A certain amount of iron dissolution may take place, leaving graphitized areas. They act as cathodes to the cast iron, and the resulting galvanic action is self-accelerated.

To avoid the graphitization and poor resistance to cavitation of cast-iron impellers, these along with other small pump parts may be of stainless steel. However, an all-bronze pump should give the longest life for sea-water conditions. Certain all-bronze pumps have lasted over 20 years, handling sea or harbor water.

Question 3-14: What is meant by graphitization?

Answer: If the liquid pumped chemically attacks the molecules of cast-iron pump parts and removes the products of corrosion, the action leaves a residue high in graphitic carbon. This phenomenon is commonly called "graphitization." The relatively soft surface left exposed is readily washed away by jetting action of the water.

Question 3-15: Are there operating pressure and temperature limitations on the use of cast iron for pump casings?

Answer: Yes. For several reasons, cast-iron casings are seldom used for pressures over 1000 psi and temperatures over 350°F.

Question 3-16: What is semisteel?

Answer: There is no such metal. Ferrous metals must be either steels or irons, and the difference between the two is strictly defined. The name has in the past frequently been applied to high-grade cast irons with a tensile strength of about 45,000 psi, halfway between the 30,000 psi of ordinary cast iron and the 60,000 psi of cast steel.

Question 3-17: Are there any temperature limitations on the use of bronze for pump fittings?

Answer: Because bronze expands about 40 per cent more than steel, and for other reasons, it is generally not used for pump fittings when the temperature of the liquid being pumped exceeds 250°F.

Question 3-18: Does the pH value of the pumped liquid affect the selection of pump materials?

Answer: While pH is not the only factor influencing the selection of materials for pump parts, standard bronze-fitted pumps should not be used for pH values below 6.0 or above 8.5 at pumping temperature. Below 6.0, all-bronze pumps or stainless-steel-fitted pumps should be used, whereas above 8.5, the pumps should be all-iron or all-stainless-steel fitted.

Question 3-19: Why should dissimilar metals not be used in centrifugal pumps handling an electrolyte?

Answer: Severe corrosion may occur because of galvanic action between the metals immersed in the electrolyte.

Question 3-20: Does the choice of materials affect the clearances that must be used for adjacent pump parts?

Answer: In general, stainless steels have a greater tendency to gall than iron, steel, or bronze. As a result, it is the practice to make running clearances between stainless-steel parts larger than for nonstainless metals.

Question 3-21: Why is the central hub of a double-suction impeller cored out?

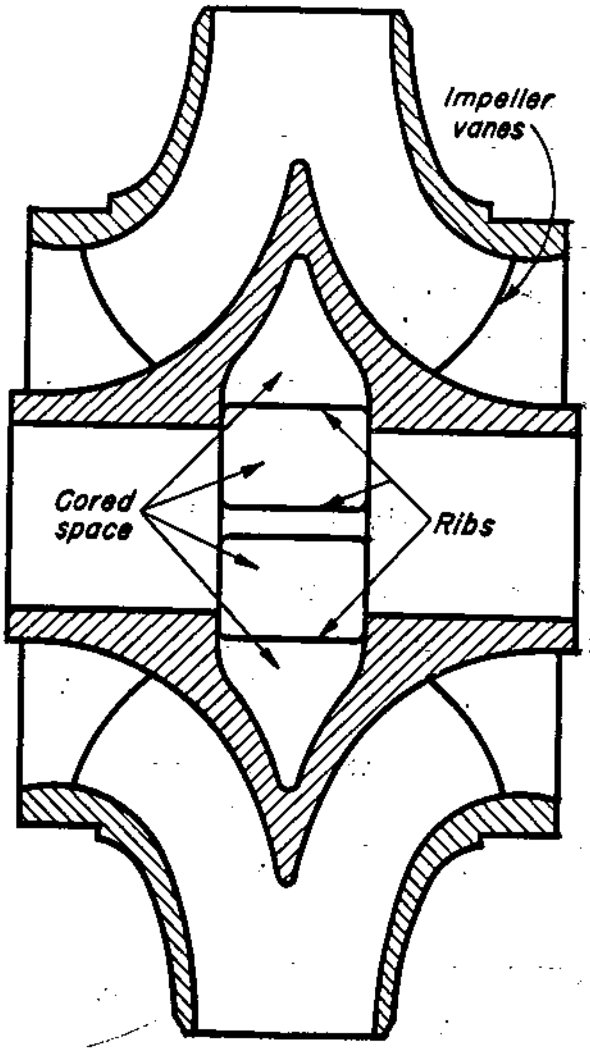


FIG. 3-3. Section of double-suction impeller showing cored-hub construction commonly used.

Answer: It is generally cored out, especially in large impellers (Fig. 3-3), to avoid a heavy cross section that might lead to porous and imperfect castings.

Question 3-22: Does the peripheral speed impose any limitations on the use of bronze impellers?

Answer: The centrifugal stresses exerted on an impeller and the resulting stretch at the impeller hub may become quite appreciable at the high peripheral speeds used in modern high head pumps. As an example, a typical 12-in. impeller mounted on a 3-in. shaft and operating at 3600 rpm has a stretch of about 0.0011 in. for bronze or cast iron. If the pump handles hot water, say at 250°F, there is an additional temperature-expansion difference between the impeller and its shaft of 0.0014, resulting in a total looseness of $0.0011 + 0.0014 = 0.0025$ in. This looseness can easily lead to water-cutting of the shaft under the impeller. To avoid the cumulative effect of excessive thermal and centrifugal expansion, the empirical limit to the use of bronze impellers in pumps handling hot liquids is a peripheral speed of about 160 ft per sec.

Question 3-23: How do the structural features of the parts of a pump affect the choice of materials?

Answer: While the choice of materials can often be made mainly from the viewpoint of corrosion and abrasion resistance, the structural features of the parts of a given pump may play a definite role in the final material selection. Some parts, for instance, may require extremely thin wall sections, and a material such as cast iron is unsuitable, even though it may not corrode. Other parts, such as shaft sleeves, require a high degree of polish, and only materials capable of receiving such a finish should be used for them. Wherever the structural design requires pressed-on or shrunk-on parts, the material for them must be suitable for this method of mounting. For instance, when pump impellers have to be shrunk on the shaft, bronze cannot be used, the choice being restricted to cast or stainless steel.

CHAPTER 4

CENTRIFUGAL-PUMP AND SYSTEM CURVES

Question 4-1: What units express centrifugal-pump capacity?

Answer: The standard unit of capacity varies with the field of application as well as with the country. For instance, centrifugal-pump designers work in terms of U.S. gallons per minute in the United States, imperial gallons per minute in British territories, and cubic meters per hour in countries using the metric system. To indicate the variation within these different fields, the following units of capacity are used regularly in the United States: million gallons per day, cubic feet per second, gallons per minute, barrels per day, barrels per hour, pounds per hour, and acre-feet per day.

Question 4-2: How should the required pump capacity be specified?

Answer: Pump capacity for an installation should be given in gallons per minute at pumping temperature. State clearly any desired or imposed variation in capacity. Centrifugal pumps do not permit as great a capacity range without affecting efficiency as do reciprocating steam pumps. Furthermore, it is generally preferable to have the maximum efficiency of a pump occur at or near the normal capacity.

While any centrifugal pump can occasionally run considerably in excess of its rated capacity, this may not always be practicable or permissible. Sometimes operation at extremely reduced capacities even for very short periods presents definite danger, and means must be provided to avoid this. In other instances, the only disadvantage from reduced-capacity operation is poor efficiency, which is avoided by installing smaller units to operate during light-load periods.

Question 4-3: What are centrifugal-pump characteristics?

Answer: Unlike the positive-displacement pumps, a centrifugal pump at constant speed delivers any capacity from zero to a maximum value, depending on the size, design, and suction conditions of the pump. Total head developed, power to drive the pump, and the resulting efficiency vary with capacity. The interrelation of capacity, head, power, and efficiency are best shown graphically, and these curves are called the "characteristic curves" of the pump

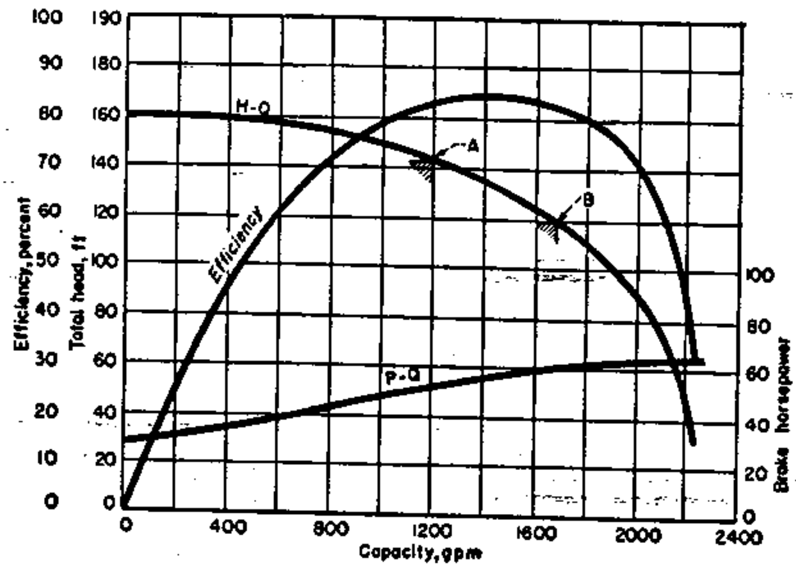


FIG. 4-1. Typical constant-speed characteristic curve showing total head, power, and efficiency plotted against capacity.

(Fig. 4-1). It is usual to plot head, power, and efficiency against capacity at constant speed, as in Fig. 4-1. It is possible, however, in special problems to plot any three components against the fourth. For variable-speed drives, a fifth component, revolutions per minute, is involved. In design studies, many other relations are shown on the same graph.

Question 4-4: What is a head-capacity curve?

Answer: The curve showing the relation between capacity and total head is the head-capacity curve, marked *H-Q* (Fig. 4-1). This pump at 144 ft head has a capacity of 1200 gpm, point *A* on the *H-Q* curve; and at 120 ft head, its capacity increases to 1680 gpm, point *B* on the *H-Q* curve.

Question 4-5: What is a rising characteristic?

Answer: A rising characteristic, or, more correctly, a rising head-capacity characteristic, is a curve in which the head rises continuously as the capacity decreases, as do the *H-Q* curve (Fig. 4-1) and curves *A*, *C*, and *D* (Fig. 4-2).

Question 4-6: What is a drooping characteristic?

Answer: A drooping head-capacity characteristic (to use its full name) covers cases in which the head developed at shutoff is less than that developed at some other capacities (curve *B*, Fig. 4-2). This is also known as a "looping curve."

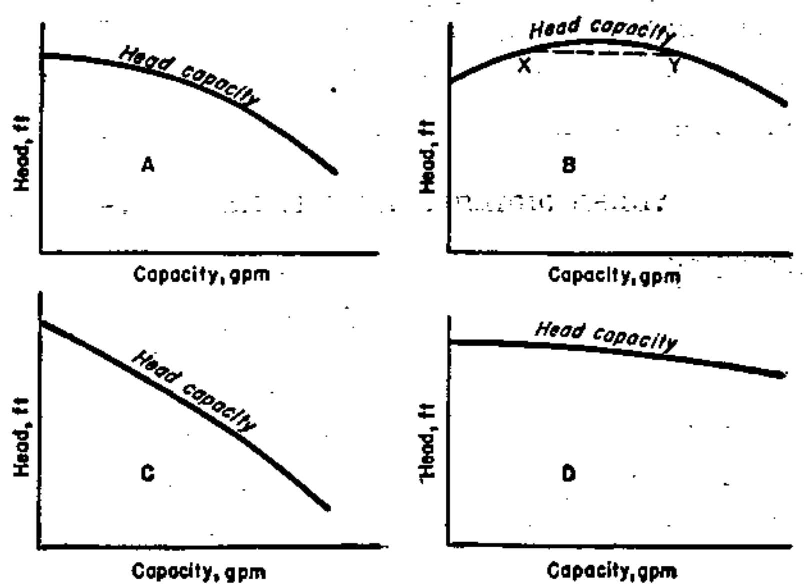


FIG. 4-2. Various impeller types and designs result in different shapes of head-capacity curves; (*A*) rising, (*B*) drooping, (*C*) steep, (*D*) flat.

Question 4-7: What is a steep characteristic?

Answer: This term generally refers to a rising head-capacity characteristic in which there is a large increase in head between that developed at design capacity and that developed at shutoff (curve *C*, Fig. 4-2). The description is sometimes applied to a limited portion of the curve. For example, a pump may be said to have a steep characteristic between 100 and 50 per cent of its design capacity.

Question 4-8: What is a flat characteristic?

Answer: Unless otherwise qualified, it refers to the head-capacity characteristic where the head varies slightly with capacity from shutoff to design capacity (curve *D*, Fig. 4-2). The characteristic might be, in addition, either drooping or rising. All drooping curves have a portion in which the head developed is approximately constant for a range in capacity, as between *X* and *Y*, curve *B* (Fig. 4-2), which is called the "flat" portion of the curve. "Flat" also at times indicates the general shape of other curves, either for their full range or parts of them.

Question 4-9: What is meant by a nonoverloading curve?

Answer: When speaking of a nonoverloading curve, reference is always made to the power-capacity curve. It is one which either remains relatively flat over the entire capacity range of the pump or, if the curve is inclined to the horizontal for most of the capacity range of the pump, has a tendency to level off. To be so classified, the maximum power required at any pump capacity should only slightly exceed the power required for the normal range of application.

Question 4-10: What are stable and unstable characteristics?

Answer: A pump whose head-capacity characteristic is such that only one capacity can be obtained with any one head has a stable characteristic. Basically, this has to be a rising characteristic, as in the *H-Q* curve (Fig. 4-1). With an unstable characteristic, the same head is developed at two or more capacities, as at *X* and *Y*, curve *B* (Fig. 4-2).

Question 4-11: Can pumps with unstable head-capacity characteristics be used?

Answer: Successful application of any pump depends as much on the characteristic of the system on which it operates as on its own head-capacity characteristic. Most pumping systems permit the use of pumps with moderately unstable characteristics.

Question 4-12: What is meant by a type-characteristic curve?

Answer: If the operating conditions of a pump at the design speed—namely, the capacity, head, efficiency, and power input at which the efficiency reaches its maximum—are taken as the standard of comparison, the head-capacity, power-capacity, and efficiency-capacity curves can all be plotted in terms of the percentage of their respective values at the point of maximum efficiency. Such a set of curves represents the type characteristic of the pump, and Fig. 4-3 shows the type characteristic of the pump whose regular characteristic is shown in Fig. 4-1.

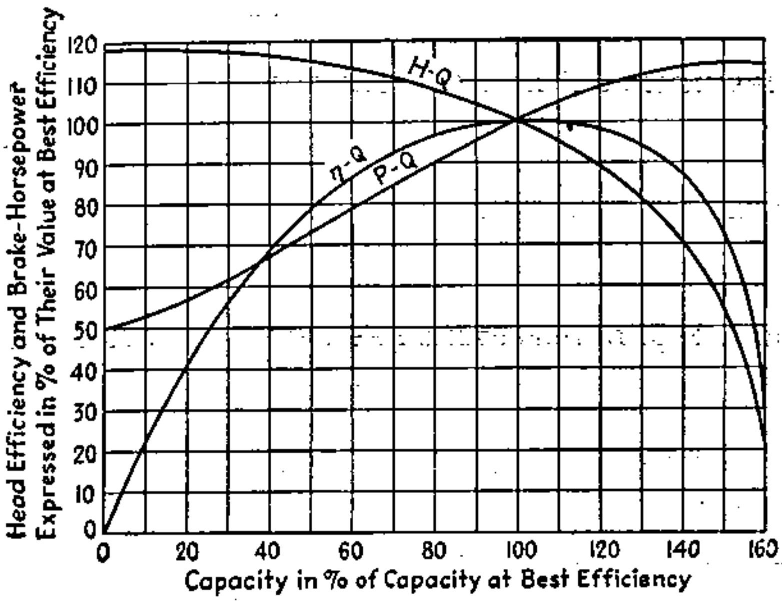


FIG. 4-3. Type characteristic or 100 per cent curve.

Question 4-13: What is specific speed?

Answer: It is an indication of centrifugal-pump type. For units of capacity and head (feet) used in the United States the specific speed of a centrifugal pump is the speed at which an exact model of the pump would have to run if it were designed to deliver 1 gpm against 1 ft head per stage. Thus all pump sizes can be indexed by the rotative speed of their unit capacity-head model. The specific-speed index of a pump is a guide in determining the maximum head against which it can operate, modified by suction conditions. The lower the specific speed is, the higher the head per stage that can be developed. Low specific speeds range from 500 to 1000, medium specific speeds from 1500 to 4000, and high specific speeds from 5000 to 20,000.

Question 4-14: How can the specific speed of a centrifugal pump be calculated?

Answer: The specific speed of a pump is obtained from the formula:

$$n_s = \frac{n\sqrt{Q}}{H^{3/4}}$$

where n_s = specific speed
n = revolutions per minute
Q = gallons per minute
H = feet head per stage

For the correct typing of a pump design, use Q and H values that give maximum efficiency.

Question 4-15: What is meant by "head," as applied to centrifugal pumps?

Answer: Fundamentally, "head" refers to a static difference in level between the free surface of a liquid at rest and an arbitrarily selected elevation—for example, the difference in elevation between the surface of a liquid in the suction well and that in the discharge tank of the pump (Fig. 4-4). This difference may be expressed in pressure (psi) or in feet static head; therefore, it refers to a difference in pressures or levels. In referring to a centrifugal pump, head becomes a measure of energy delivered (work done) by the pump at a given speed and capacity and includes static head, velocity head, and frictional losses in piping and fittings.

Question 4-16: What are the parts of a pumping-system head?

Answer: The total head against which a pump operates includes:

1. Static head.
2. Difference in pressure existing on the liquid, if any.
3. Friction head.
4. Velocity head.
5. Entrance and exit losses.

Question 4-17: What is static head?

Answer: It is a difference in elevation. Thus the total static head of a pumping system is the difference in elevation between the liquid levels of the suction and discharge (Figs. 4-4 and 4-5).

Question 4-18: What is static discharge head?

Answer: It is the difference in elevation between the center line of the pump and the level of the discharge liquid (Figs. 4-4 to 4-6).

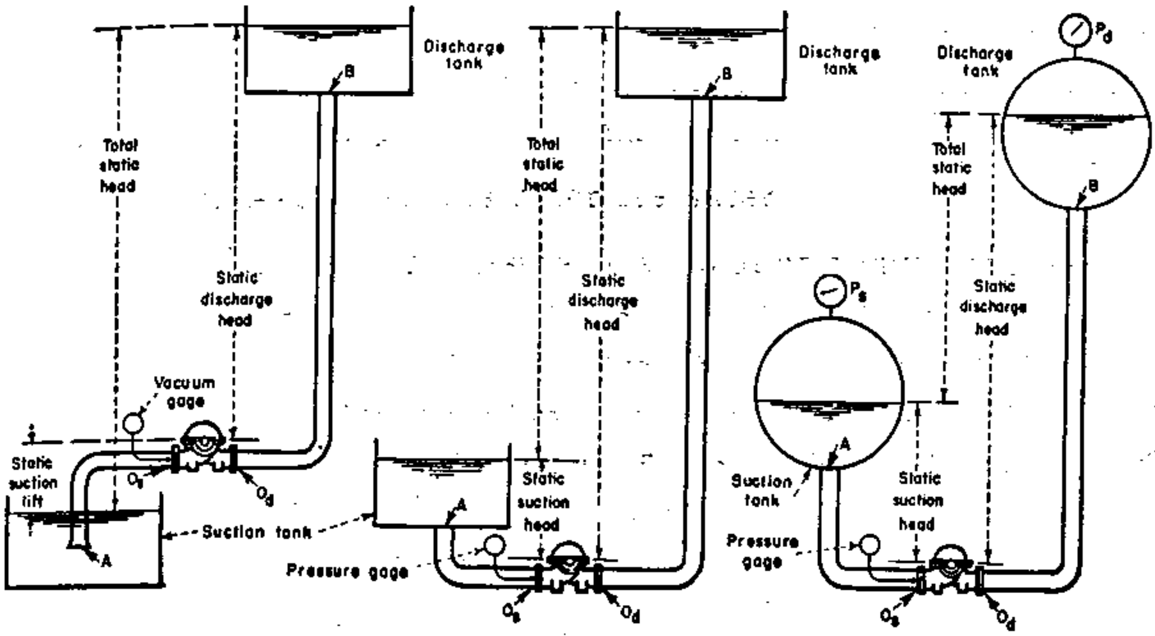


FIG. 4-4. FIG. 4-5. FIG. 4-6.

FIGS. 4-4 to 4-6. Static-head terminology for three typical pumping systems.

Question 4-19: What is static suction lift?

Answer: If the suction liquid level is below the center line of the pump (Fig. 4-4), it is generally called a "static suction lift." This is the difference in elevation between the center line of the pump and the level of the liquid in the suction well.

Question 4-20: What is static suction head?

Answer: When the suction liquid level is above the center line of the pump (Fig. 4-5), it is called a "static suction head" and is the difference in elevation between the suction liquid level and the center line of the pump. When the suction supply is taken from a closed vessel with a liquid level above the center line of the pump, as from the hotwell of a condenser, the difference in elevation between the suction liquid level and the center line of the pump is commonly called "static submergence."

Question 4-21: When either the suction liquid level or the level of the discharge liquid is under pressure other than atmospheric (Fig. 4-6), is this pressure a part of static head?

Answer: Opinion varies as to whether this pressure should be considered part of the static head. If it is considered separately, this generally gives a clearer picture of the pumping system and the head components.

Question 4-22: What is meant by friction head?

Answer: It is the equivalent head, expressed in feet of liquid pumped, necessary to overcome friction losses caused by liquid flow through the piping, including all fittings. Friction head varies with the quantity of flow, the size, type, and condition of piping and fittings, and the characteristics of the liquid pumped.

Question 4-23: What is velocity head?

Answer: Velocity head is related to the velocity of a liquid at any given point in its flow line, expressed in feet head of the liquid. If the liquid is moving at a given speed, velocity head is equivalent to the distance that the liquid would have to fall in a vacuum to attain that velocity:

$$h_v = \frac{V^2}{2g}$$

where h_v = velocity head in feet of liquid
V = velocity in feet per second
g = acceleration of gravity (32.2)

If the velocity in a pipeline is 5 ft per sec:

$$h_v = 5 \times 5 \div 64.4 = 0.39 \text{ ft}$$

Question 4-24: What are entrance and exit losses?

Answer: Except where the suction supply of a pump is from a main under pressure, such as a city water system, it generally comes from a tank or reservoir. The connection of the suction pipe to the tank wall or the end projecting into the liquid is the entrance of the suction pipe. Frictional losses at this point are called "entrance losses." Their magnitude depends on the entrance design, a well-designed bellmouth having the lowest loss. Likewise, where the discharge line terminates, its end is called the "exit." Generally, this exit is the same size as the piping, and the velocity head of the liquid is lost. Sometimes the discharge pipe end has a long taper so that the velocity is reduced and its energy recovered.

Some engineers consider entrance and exit losses part of the friction losses of the suction and discharge pipes. Others consider them separately so as to make sure that they are not overlooked. The latter procedure has the advantage, also, that it is easy to see if either or both losses are excessive.

Question 4-25: What is suction head?

Answer: It is the head existing at the suction nozzle of the pump, and can be expressed in absolute or gauge pressure, or feet head.

Question 4-26: What is suction lift?

Answer: Suction lift exists when the total suction head is below atmospheric pressure.

Question 4-27: How is the suction head or suction lift determined?

Answer: Total suction head, as now generally applied to centrifugal pumps, is the static suction head measured to the center line of the pump, minus friction losses for the capacity pumped, minus entrance losses at the beginning of the suction line, plus any pressure existing on the suction supply. A pressure gauge connected to the suction of the pump indicates total suction head, minus the velocity head at the point where the gauge is connected. Because a suction lift is a negative suction head, a vacuum gauge indicates the total suction lift, plus velocity head at the point where the gauge is connected to the suction line.

Figures 4-4 to 4-6 show the three most common suction conditions. In Fig. 4-4, suction lift equals static suction lift measured to the center line of the pump, plus total friction loss in the suction line and entrance losses at A. In Fig. 4-5, the suction head equals the static suction head, measured to the center line of the pump, minus entrance losses at A and friction losses in the suction line to the pump suction flange. The suction head, under conditions of Fig. 4-6, equals the total static head measured to the center line of the pump, minus entrance losses at A and friction losses from A to the suction flange of the pump, plus pressure indicated by gauge P_s.

Question 4-28: Does a vacuum gauge connected into the suction line at O, (Fig. 4-4) indicate the true suction lift?

Answer: No, it indicates suction lift, plus velocity head or static suction lift, as measured to the center line of the pump, plus entrance losses at A, plus total friction losses in the suction line, plus velocity head. In other words, the vacuum gauge reads too high by an amount equal to the velocity head.

Question 4-29: Does a pressure gauge connected into the suction line at O, (Figs. 4-5 and 4-6) indicate the true suction head?

Answer: No, it indicates suction head minus velocity head. In Fig. 4-5, the gauge reads the static suction head, measured to the center line of the pump, minus entrance losses at A and total friction losses in the suction line, minus the velocity head. For conditions in Fig. 4-6, the gauge also reads the suction head, minus the velocity head. That is, under conditions of Figs. 4-5 and 4-6, the gauge reads low by an amount equal to the velocity head.

Question 4-30: Can a pump with a static suction head, like Fig. 4-5, have a suction lift when operating?

Answer: Yes, particularly if the static suction head is comparatively low and the suction line long with several elbows. Then the sum of the entrance losses to the suction line and the friction losses in the line may be greater than the static head. Here pressure at the suction nozzle of the pump is below atmosphere, not above, and the pump operates with a suction lift. If entrance and friction losses equal the static suction head, we have the condition of zero suction lift or zero suction head.

Question 4-31: What is the discharge head of a pump?

Answer: It is the head measured at the discharge nozzle and can be expressed in absolute or gauge pressure readings, or in feet head. It is the sum of the static discharge head, friction head for the capacity considered, the exit losses at the end of the discharge line, and the terminal head or pressure expressed in feet of liquid being pumped. Discharge head is always measured from the center line of the pump. A pressure gauge connected into the discharge of the pump, when its reading is corrected to the center line of the pump, indicates the discharge head, minus velocity head at the point where the gauge is connected.

Question 4-32: How is discharge head determined?

Answer: This depends on the arrangement of the discharge piping. When the pump discharges into the bottom of a tank (Figs. 4-4 and 4-5), discharge head equals static head, plus friction losses from O_d to B, plus exit losses at B. In Fig. 4-6, discharge head equals static head, plus friction losses from O_d to B, plus exit losses at B, plus pressure-gauge reading P_d.

While Fig. 4-7 represents an overhead tank with discharge above the tank, it applies to all cases of overboard discharge. Here discharge head equals static head D measured from the center line of the pump to the highest point of the discharge pipe, plus friction losses from O_d to B, plus exit losses at B, as in Fig. 4-4. However, the distance from the center line of the pump to the discharge water level is less than the actual static discharge head. It is possible to recover all or part of this difference in static head by installing some form of siphon leg on the discharge (Figs. 4-8 to 4-10). While the arrangement in Fig. 4-9 is the most efficient theoretically, it is frequently desirable to use an open discharge, as in Fig. 4-8.

When the pump is in operation, discharge head equals the static head D (Fig. 4-8), as measured vertically from the center line of the pump to B, plus total friction losses in the discharge piping, plus the exit losses at B. To operate, however, the pump must develop sufficient head to lift the water over the loop against static

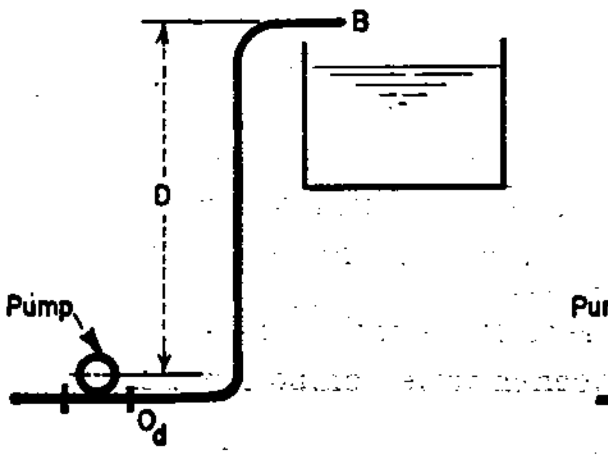


Fig. 4-7.

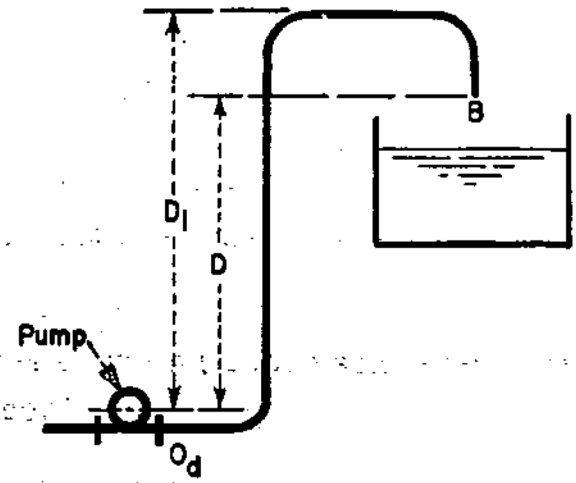


Fig. 4-8.

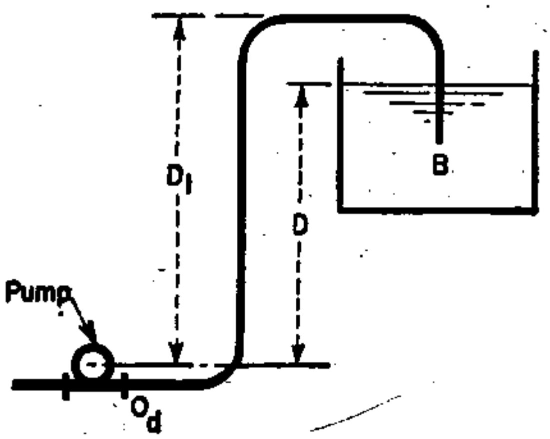


Fig. 4-9.

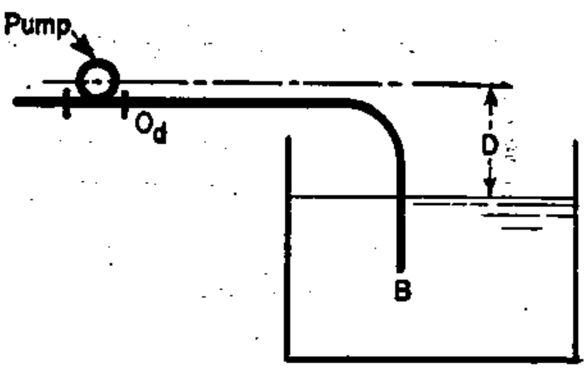


Fig. 4-10.

Figs. 4-7 to 4-10. Four typical discharge arrangements for pumps pumping into open tanks or reservoirs.

head D_1. Normally, a pump selected for the head existing after it is in operation can start the liquid flowing over the loop. If there is any doubt about the pump being able to start the siphon, its head-capacity characteristic should be checked against the maximum static head D_1 of the system in which it will operate. All installations involving a siphon leg should be investigated carefully to determine how much head recovery to expect.

For conditions in Fig. 4-10, the pump is part of a siphon, as in a condenser circulating-water system, and serves to start and keep the liquid flowing. Its discharge head may be negative, that is, less than atmospheric, and is equal to static head D, measured from the surface of the liquid to the center line of the pump, minus friction losses in the discharge pipe from O_d to B, minus exit loss at B. If friction and exit losses are greater than static head D, the discharge head would be above atmosphere, that is, positive.

Question 4-33: What is total head?

Answer: It is discharge head plus suction lift if suction lift exists, or discharge head minus suction head if suction head exists, or discharge head only if zero suction lift or suction head exists. It is a measure of work done by the pump on the liquid, minus all friction losses in the pump from its suction to discharge nozzle.

Question 4-34: What is total dynamic head?

Answer: The four terms, "total dynamic head," "dynamic suction head," "dynamic suction lift," and "dynamic discharge head," are no longer used. While "total dynamic head" referred to the head that is now called "total head," "dynamic suction head," "dynamic suction lift" and "dynamic discharge head" were defined as the heads measured by gauge corrected to the pump center line and thus did not include velocity head. This led to general misunderstanding if the size of the pump suction and discharge openings were not specified, and resulted in different head values for pumps working under identical conditions when openings were not the same. Furthermore, in determining total dynamic head from dynamic discharge head and dynamic suction head, it was necessary to correct for any difference in velocity head. The present method of specifying heads is more satisfactory than the dynamic-head method.

Question 4-35: What is a system friction-head curve?

Answer: In pumping problems, it is often convenient to show graphically the relation between flow and friction head in the piping system. Such a curve (Fig. 4-11) is known as a system friction-head curve. It is obtained by plotting the calculated friction head in the piping, valves, and fittings of the suction and discharge lines against the flow on which the calculation was based. In the figure, at 200 gpm the friction head is 9 ft, point A, and at 400 gpm the friction head is 36 ft, point B. Friction head varies roughly as the square of the flow.

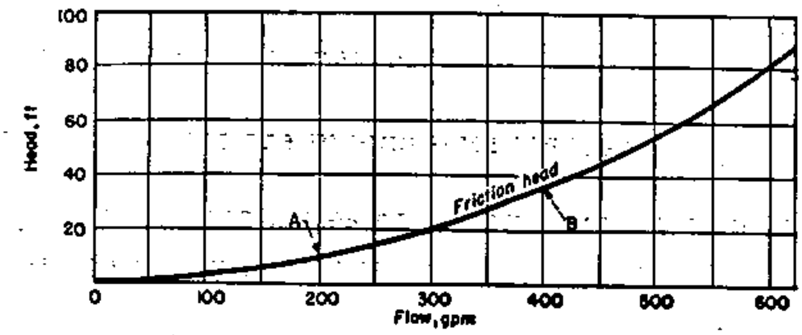


FIG. 4-11. System friction-head curve showing friction head at various capacities.

Question 4-36: What is a system-head curve?

Answer: It is a curve obtained by combining the friction-head curve with the static head and any difference of pressures in a pumping system. In Fig. 4-12, the friction-head curve (Fig. 4-11) is combined with a total static head of 110 ft to obtain the system-

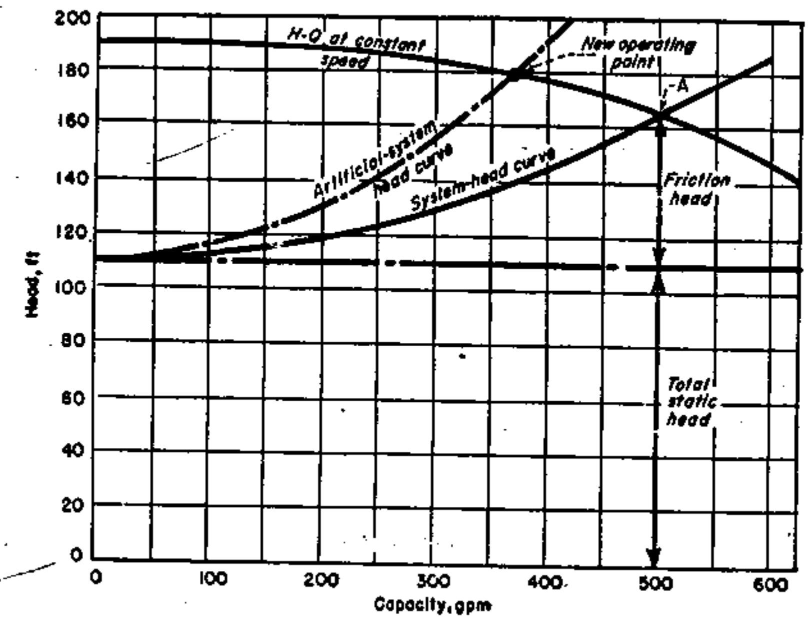


FIG. 4-12. System-head curve is sum of total static head and friction head for the system. (Artificial system-head curve is discussed in Question 5-24.)

head curve. Here the static head is assumed to be constant, as it is where the suction source and discharge levels are constant. Friction head increases with the flow. Superimposing the head-capacity curve of the pump on the system-head curve, as in Fig. 4-12, shows the head and capacity at which the pump will operate, such as point *A* where the two curves cross. In this system, the pump discharges 500 gpm at a 165-ft head.

Where the static head or pressure varies, system-head curves may be plotted for minimum and maximum static heads or pressures. Superimposing a head-capacity curve of a pump on such system-head curves permits predicting the capacity that the pump will deliver at different static heads.

Question 4-37: How is the head on a centrifugal pump generally expressed?

Answer: The head on a centrifugal pump is generally stated in feet of liquid pumped.

Question 4-38: Why is head expressed in feet of liquid pumped?

Answer: Head is then always the same, irrespective of the liquid pumped. For example, in a given system a pump that develops 100 ft total head (43.3 psi) when handling cold water also develops 100 ft total head when pumping 1.2 specific-gravity brine or 0.8 specific-gravity light oil. When pumping brine, the head developed equals $100 \times 1.2 \div 2.31 = 51.9$ psi, whereas for light oil the head equals $100 \times 0.8 \div 2.31 = 34.6$ psi. However, when the heads are measured in feet of the liquid pumped, they are the same. In other words, a pump that has been discharging cold water into a tank against 100 ft head can be expected to discharge the same quantity of brine into the same tank.

Question 4-39: How is pressure expressed in pounds per square inch (psi) converted to feet head of the liquid pumped?

Answer: A column of water about 2.31 ft high produces a pressure of 1 psi at its base. Thus, when cold-water pressure is expressed in psi, multiplying this value by 2.31 gives the head in feet of water. For example, a pressure gauge indicates 45 psi when connected to the discharge nozzle of a centrifugal pump. This pressure is equivalent to $45 \times 2.31 = 104$ ft head. This is not the total head developed by the pump because velocity head, suction, and other conditions must be taken into account. If the liquid pumped is other than cold water, its specific gravity must be considered. In the case of 1.2 specific-gravity brine, the conversion unit becomes $2.31 \div 1.2 = 1.93$.

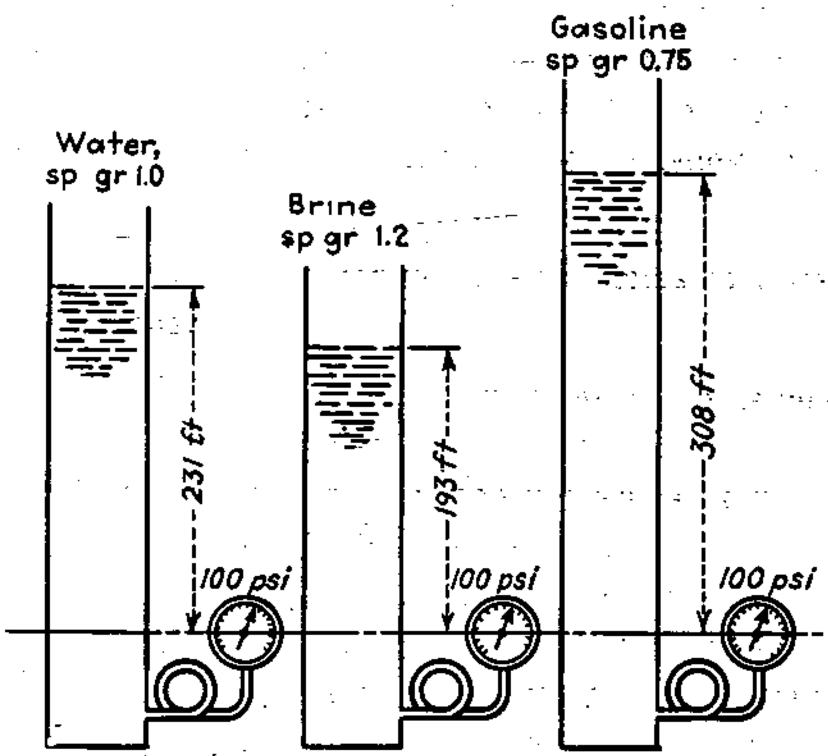


FIG. 4-13. Comparison of heads involved for water, brine, and gasoline for 100 psi pressure.

Figure 4-13 shows the relation between 100 psi and feet head of water, brine, and gasoline. On the left, 100 psi water pressure is equivalent to a column $100 \times 2.31 = 231$ ft high. Having a specific gravity of 1.2, the brine needs a column of only $231 \div 1.2 = 193$ ft high to equal 100 psi. Gasoline of a specific gravity of 0.75 requires a column of $231 \div 0.75 = 308$ ft to equal 100 psi.

Question 4-40: How do suction conditions affect pump design?

Answer: To cause liquid flow into the impeller of a centrifugal pump, an outside source of pressure must be provided. This may be static head when the suction-source level is above the pump center line, atmospheric pressure when the suction source is below the pump center line, or both. When the net absolute pressure above the vapor pressure of the liquid is small, the pump suction water passages must be comparatively large to keep the velocities in them down to a low value. If this is not done, the absolute pressure at the suction eye of the impeller may drop below the vapor pressure and the pumped liquid will flash into vapor, preventing further pumping. Where abnormal suction conditions exist, an oversized or special pump is generally required, which costs more and has a lower speed and efficiency than a pump of equal capacity and head for normal suction conditions.

Question 4-41: What is vapor pressure?

Answer: The vapor pressure of a liquid at a given temperature is that pressure at which it flashes into vapor if heat is added to the liquid, or, conversely, that pressure at which vapor at a given temperature condenses to liquid if heat is subtracted. For instance, if a closed vessel from which all air has been exhausted is partly filled with water, part of it flashes into vapor and occupies the space not filled with liquid. The resulting pressure in the vessel after the water ceases to vaporize is the vapor pressure for water at that temperature and is the pressure at which water at that temperature boils if heat is added.

Question 4-42: What is the vapor pressure of water at 212°F, expressed in feet of water?

Answer: Since the vapor pressure of water at 212°F is 14.7 psia (standard barometric pressure at sea level), the equivalent head in feet of 62°F water is $14.7 \times 2.31 = 33.9$ ft. The specific gravity of water at 212°F is 0.959 (Fig. 4-14). Then water at this temperature has a vapor pressure of $33.9 \div 0.959 = 35.4$ ft of 212°F water.

When figuring pump heads, care must be taken to convert pressures to feet of liquid at the pumping temperature and not to use conversion factors applying to other temperatures. Figure 4-14 gives the vapor pressure and specific gravity for water for a temperature range from 32 to 220°F.

Question 4-43: What is cavitation?

Answer: "Cavitation" describes a cycle of phenomena that occurs in flowing liquid because the pressure falls below the vapor pressure of the liquid. When this occurs, liquid vapors are released in the low pressure area and a bubble or bubbles form. If this happens at the inlet to a centrifugal pump, the bubbles are

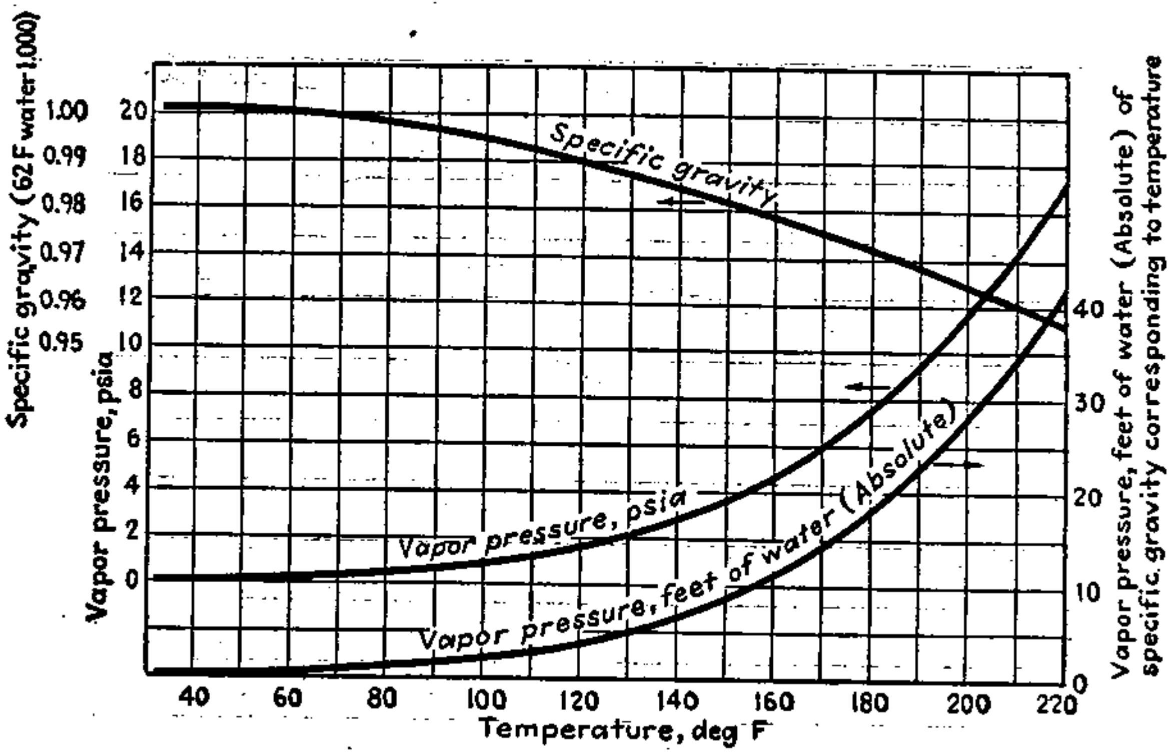


FIG. 4-14. Specific gravity, vapor pressure, and temperature relations of water.

carried into the impeller to a region of high pressure, where they suddenly collapse. The formation of these bubbles in a low pressure area and their sudden collapse later in a high pressure region is called "cavitation." Erroneously, the word is frequently used to designate the effects of cavitation rather than the phenomenon itself.

Question 4-44: In what form does cavitation manifest itself in a centrifugal pump?

Answer: The usual symptoms are noise and vibration in the pump, a drop in head and capacity with a decrease in efficiency, accompanied by pitting and corrosion of the impeller vanes. The pitting is a physical effect that is produced by the tremendous localized compression stresses caused by the collapse of the bubbles. Corrosion follows the liberation of oxygen and other gases originally in solution in the liquid.

Question 4-45: What is NPSH?

Answer: NPSH is an abbreviation of Net Positive Suction Head, which is the suction head (the gauge pressure in feet taken at the suction flange of the pump and corrected to the pump center line, plus velocity head) minus the gauge vapor pressure of the liquid at pumping temperature. Measure both the suction head and vapor pressure in feet of liquid pumped, either in gauge or absolute-pressure units. Vapor pressure of 62°F water at sea level is 0.6 ft of water. Then, a pump handling 62°F water at sea level with zero feet total suction head has a NPSH of $33.9 - 0.6 = 33.3$ ft, while one operating with a 15-ft total suction lift has a NPSH of $33.9 - 0.6 - 15 = 18.3$ ft.

It is necessary to differentiate between available and required NPSH. The available NPSH is a characteristic of the system in which the pump works and is the difference between existing absolute suction head and vapor pressure at prevailing temperature. The required NPSH is a function of pump design and represents the minimum required margin between the suction head and the vapor pressure at a given capacity.

Both available and required NPSH vary with capacity. With a given static pressure or elevation difference on the suction side of a centrifugal pump, the available NPSH is reduced with increasing capacity by the increase in friction losses in the suction piping. On the other hand, the required NPSH, being a function of velocities in the pump suction passages and at the inlet of the impeller, increases approximately as the square of the capacity.

Question 4-46: Is there any relation between the minimum required NPSH and such factors as impeller-eye diameter, velocity at the eye, pickup speed, etc.?

Answer: A great many factors, such as eye diameter, suction area of the impeller, shape and number of impeller vanes, area between these vanes, shaft and impeller-hub diameter, impeller specific speed, the shape of the suction passages, and many others in some form or other enter into the determination of the required NPSH. All these factors, however, cannot be made to yield a simplified method of determining the required NPSH, as different designers use different methods to accomplish the same results; nor can any single one of these factors be used exclusive of the others. Actually, the knowledge of the value of one or several of these factors can serve no useful purpose.

Question 4-47: Why can a centrifugal pump operate with a higher suction lift when its capacity is reduced, as by throttling the discharge?

Answer: The pump can do this because with reduced capacity less NPSH is required to cause flow into the impeller. With atmospheric pressure on the suction supply, reduction in the NPSH required by the pump means that a greater portion of atmospheric pressure can be utilized to lift the liquid to the pump, or, conversely, the pump can operate with a higher suction lift. In addition, if friction losses in the suction piping consume an appreciable portion of the available absolute suction head, the reduction of these losses also allows a greater portion of the atmospheric pressure to be utilized to lift the liquid to the pump.

Question 4-48: Does suction lift affect the efficiency of a centrifugal pump?

Answer: Yes. For a centrifugal pump to develop its normal head-capacity characteristics, the suction lift against which it operates must not exceed a given value. If this value is exceeded, the head developed by the pump and its efficiency begin to decrease to below normal for a given capacity, as indicated in Fig. 4-15. For example, with 146 ft total head the pump has a normal capacity of 1800 gpm, point *A*, and an efficiency of about 87 per cent, point *B*. This can be obtained with suction lifts up to 18 ft. Higher suction lifts cause a reduction in capacity; for example, at 20 ft suction lift and 146 ft head the capacity drops to 1670 gpm, point *C*, at 83.5 per cent efficiency, point *D*.

Normally, the pump should operate under a combination of capacity and suction conditions that permit it to develop normal head and capacity characteristics. Under these conditions the pump also operates at best efficiency for a given capacity.

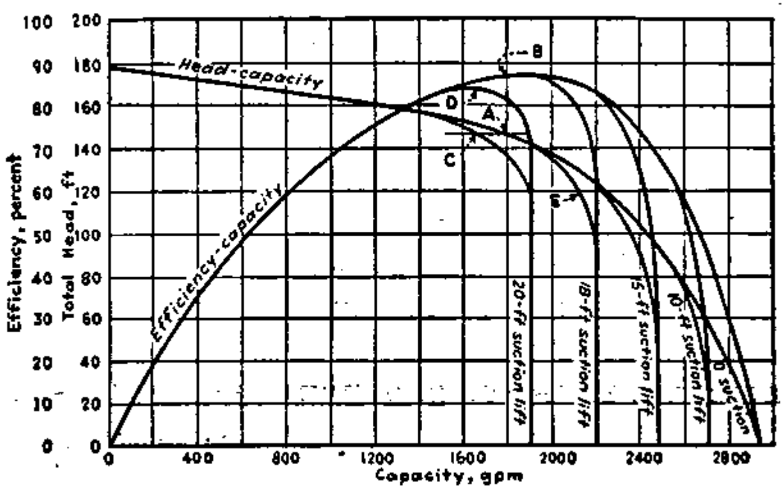


FIG. 4-15. Change in characteristics of a centrifugal pump with change in suction lift.

Question 4-49: What is meant by operating "in the break"?

Answer: If a pump is operated at the maximum capacity that it can deliver with the available absolute suction head and at a total head below that which it can develop with a higher absolute suction head, it is operating "in the break." For example, in Fig. 4-15, when the pump is operating at A on the head-capacity curve with 18 ft suction lift ($33.9 - 18 = 15.9$ ft absolute suction head), it is operating at about its maximum capacity at normal head under this suction condition. If the total head is reduced but the suction lift held at 18 ft, pump performance breaks away from the normal head-capacity curve and follows along curve E. In other words, when the pump is operating at 2200 gpm with 18 ft suction lift, it is operating "in the break." However, if the suction lift is reduced, pump performance continues down along the normal head-capacity curve.

Question 4-50: Does the liquid pumped influence the maximum permissible suction lift?

Answer: Yes, because the maximum permissible suction lift is the difference between the atmospheric pressure and the minimum required NPSH over the vapor pressure of the liquid. On the one hand, the liquid specific gravity determines the value of the atmospheric pressure expressed in terms of feet of liquid, and on the other, the vapor pressure varies with the kind of liquid and its temperature.

Even with cold water, the kind of water has an effect. For example, well water generally contains dissolved gases which, in the vacuum of high suction lifts, dissociate from the water at the pump entrance. If the amount of these gases is sufficient, the pump loses its prime. Some well-supply systems have separating tanks in the pump suction lines so that the gases can separate out and be removed by an air pump.

Question 4-51: What effect does elevation above sea level have upon centrifugal-pump performance?

Answer: Elevation above sea level decreases atmospheric pressure about 1 in. of mercury per 1000 ft of elevation. At an elevation of 4000 ft, atmospheric pressure is therefore 4 in. of mercury or about 4.5 ft of 62°F water less than at sea level. This means that a given centrifugal pump can handle 4.5 ft less suction lift than at sea level. This effect should not, however, lead to the confused notion that the NPSH required for a pump changes with elevation above sea level. It does not, but the atmospheric pressure that is available is reduced.

CHAPTER 5

APPLICATION OF CENTRIFUGAL PUMPS

Question 5-1: What is water horsepower?

Answer: Incidentally, it should be called "liquid horsepower" and is the hydraulic power generated by a centrifugal pump. It equals pump capacity in gallons per minute, times the total head in feet, times the specific gravity of the liquid, divided by 3960. If a pump discharges 1800 gpm of cold water against 110 ft total head, water horsepower (whp) $= 1800 \times 110 \times 1 \div 3960 = 50$.

Question 5-2: What is brake horsepower?

Answer: It is the power required to drive a pump and equals its water horsepower divided by its efficiency. It is frequently called "horsepower." If the efficiency of a pump is 0.80 and its water horsepower 50, its brake horsepower (bhp) $= 50 \div 0.80 = 62.5$.

Question 5-3: What are the losses in a centrifugal pump?

Answer: They are classified as:

1. Hydraulic.
2. Disk-friction.
3. Leakage or short-circuit.
4. Mechanical.

Question 5-4: What are hydraulic losses?

Answer: They are losses caused by shocks, eddy currents, and friction of the fluid in its path through the impeller and casing waterways.

Question 5-5: What are disk-friction losses?

Answer: Disk-friction losses (commonly called "disk horsepower") represent the power required to rotate the impeller in the liquid surrounding it in the pump casing. With other factors constant, the disk horsepower increases rapidly with the viscosity of the pumped liquid.

Question 5-6: What are short-circuit losses?

Answer: They are losses caused by leakage from the discharge to the suction side of a pump through the clearances between the casing and impeller wearing rings. This leakage reduces the effective capacity of the pump because the leakage has to be repumped from suction to discharge.

Question 5-7: What are mechanical losses?

Answer: They are the losses resulting from friction in the stuffing boxes and the pump bearings.

Question 5-8: How does the viscosity of a liquid affect the disk-friction losses of a pump?

Answer: They increase with increase in viscosity, because the thicker a liquid, the more power is needed to rotate the impeller through it. If viscosity is sufficiently high to reduce the head generated by the pump, the diameter of the impeller must be increased to recover this head loss, thus further increasing disk-friction losses, which vary approximately as the fifth power of the diameter of the impeller.

Question 5-9: How does the viscosity of a liquid affect the total head of a pump?

Answer: The head-capacity curve of a centrifugal pump handling a viscous fluid does not duplicate the head-capacity curve of the same pump when handling water. The effect of viscosity on the performance of a typical pump is indicated in Fig. 5-1. For example, when handling water at 120 gpm, the pump develops 44 ft head, point *A*, but when handling a liquid of 1000-SSU viscosity at 120 gpm, it develops a head of only 29 ft, point *B*. This loss of head with increasing viscosities is caused by increased friction head as well as by the over-all effect of the viscosity on the flow configuration in the impeller and casing.

Question 5-10: How does the viscosity of a liquid affect pump efficiency?

Answer: Since two of the losses in a centrifugal pump, disk horsepower and friction head, increase with increased viscosity, the

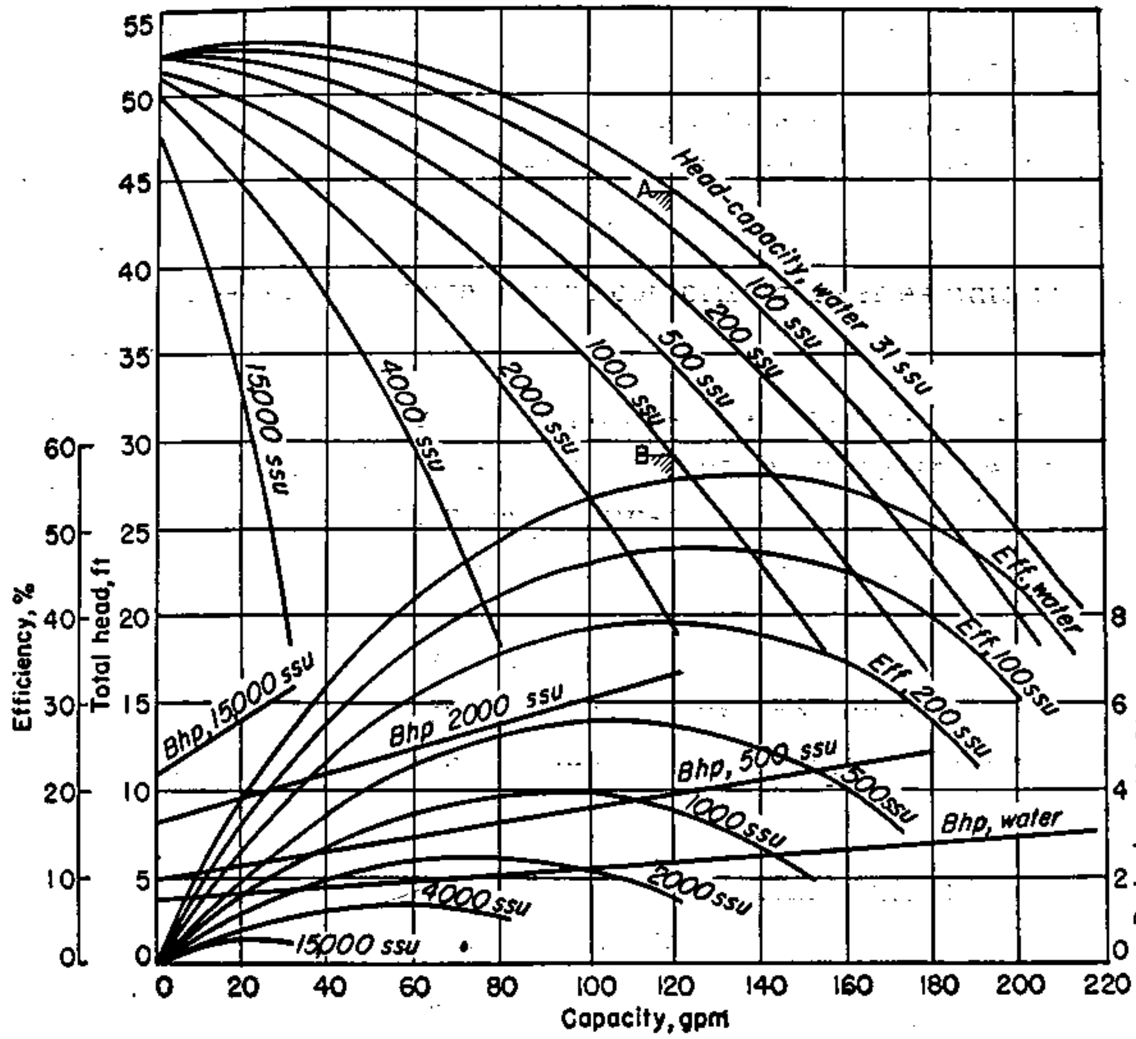


FIG. 5-1. Effect of viscosity of liquid pumped on characteristics produced by a centrifugal pump.

efficiency of a pump handling viscous liquids will be lower than one handling water, as indicated in Fig. 5-1. How much it will decrease depends on the type of pump, its hydraulic design, and the relation of its operation to the conditions at best efficiency as well as to the viscosity of the liquid.

Question 5-11: How does the viscosity of a liquid affect pump selection?

Answer: Because disk friction varies as the fifth power of the diameter of the impeller but only as the speed cubed, it is desirable to use the highest speed (and the resulting smaller impeller diameter) consistent with design conditions. Head loss due to viscosity increases as operating conditions approach the best-efficiency point of the pump because of increased velocity through the impeller. Therefore, there is an advantage in selecting a relatively oversized pump for a highly viscous liquid so as to place the operating conditions well to the left of capacity at best efficiency on low-viscosity service.

Question 5-12: What range of efficiencies is obtainable with centrifugal pumps?

Answer: The efficiency of a centrifugal pump depends on many factors, such as the specific speed of the pump, its relative size, the service for which it is intended, the materials of which it is constructed, and physical characteristics such as the viscosity of the liquid. Large centrifugal pumps have developed over 92 per cent efficiency. Small pumps handling viscous liquids, under severe mechanical conditions calling for relatively large clearances between the wearing rings, may have efficiencies as low as 10 or even 5 per cent.

Question 5-13: What is the design speed of a pump?

Answer: It is the speed at which the pump was designed to operate and deliver its rated capacity (gpm) against its rated head in feet. Thus the design speed is the normal operating speed of the pump. At a speed higher than normal, the head and capacity of the pump increase, and at speeds below normal, they decrease.

Question 5-14: How does a speed change affect pump performance?

Answer: Its capacity varies directly as the speed, its head as the square of the speed, and its power input as the speed cubed.

Example: A pump that delivers 1000 gpm against 100 ft total head at 1000 rpm, point *A* (Fig. 5-2), and requires 40 hp to drive it, point *B*, will deliver 900 gpm against 81 ft total head when run at 900 rpm, point *C*, and will require 29.2 bhp to drive it under those conditions, point *D*. All points on the characteristics are similarly affected so that if the ones for one speed are known, those for another speed can be predicted.

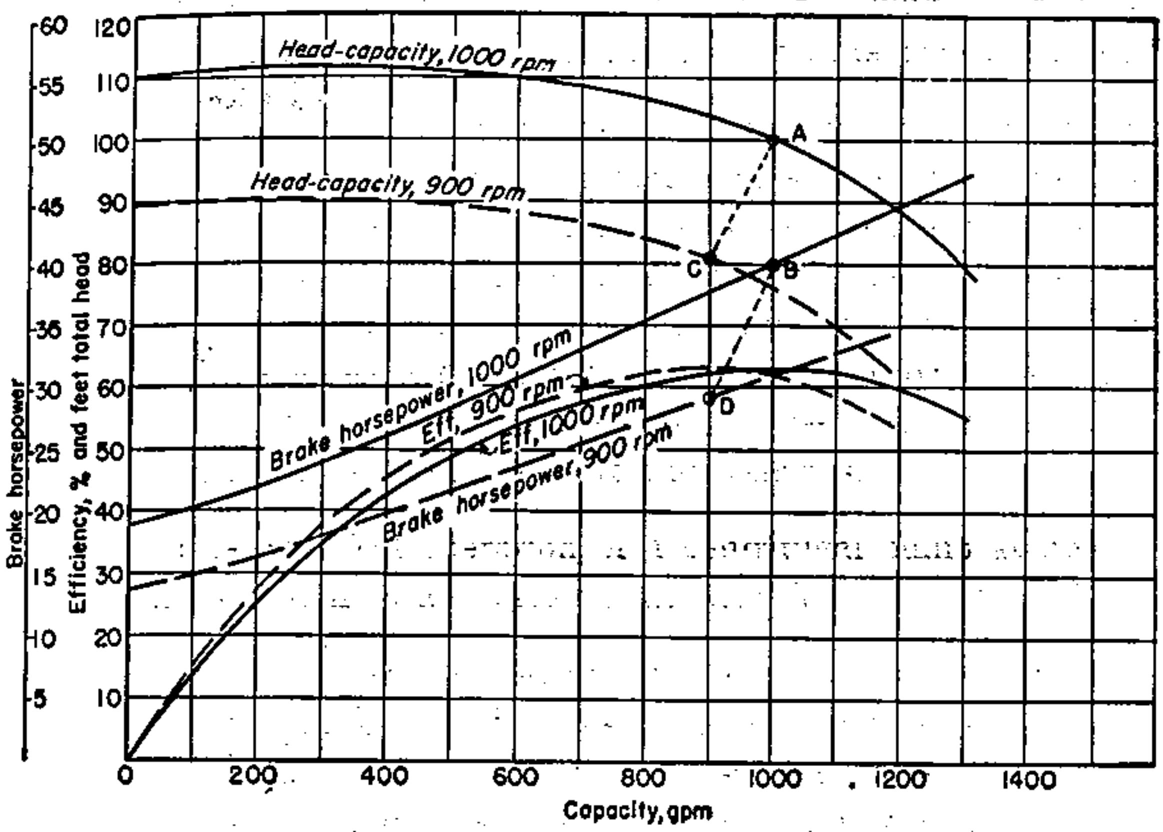


FIG. 5-2. Change in pump characteristics with change in speed.

If pump speed is increased sufficiently, the limit of operation at the prevailing suction conditions can be exceeded, and its head and capacity range will be reduced. Likewise, if the speed is reduced excessively, the efficiency of the pump will be somewhat reduced so that its power input will no longer follow the speed-cubed relationship.

Question 5-15: How does a change in impeller diameter affect pump performance?

Answer: Since the peripheral speed of the impeller changes directly as its diameter, the pump capacity varies directly as the impeller diameter, whereas the head varies as the diameter squared and the power input varies as the diameter cubed. For example, in Fig. 5-3 with a 14.75-in. impeller operating at 1800 rpm, the pump has a capacity of 2250 gpm against a 218 ft head, point *A*, and requires 151 hp to drive it, point *B*. If the impeller diameter is reduced to 14 in., its capacity at 196 ft head will be 2135 gpm, point *C*, and will require 129 hp, point *D*.

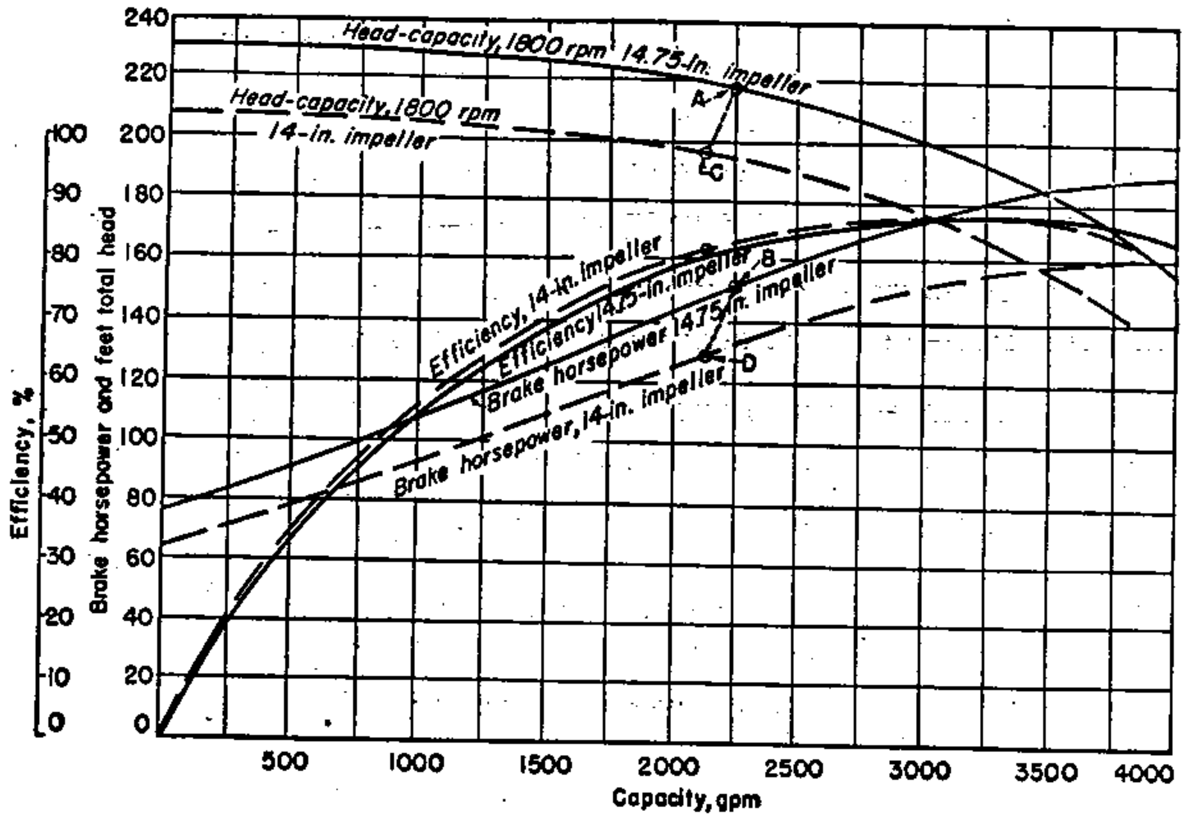


FIG. 5-3. Change in pump characteristics with change in diameter of impeller.

A reduction in diameter is the equivalent of a basic design change of the impeller and thus affects the characteristics of the pump. For that reason, the basic rule can be applied only over a limited range, depending on the design of the impeller.

When a pump is to be adapted for smaller capacity or head, or both, at the same speed, or for the same capacity and head at a higher speed, consult the manufacturer to see whether the existing impeller can be cut down to produce the desired results.

Question 5-16: Knowing the diameter of an impeller and its operating speed, how can one determine approximately the head that a pump can develop?

Answer: The head in feet developed by an impeller at the capacity at which maximum efficiency is obtained can be determined from the formula:

$$H = \left(\frac{Dn}{1840K_u}\right)^2$$

where H = head, feet of liquid
D = impeller diameter, in.
n = speed, rpm
K_u = pump-design coefficient

The value of K_u may vary between 0.95 and 1.15 for normal impeller designs. Generally, using 1.03 is accurate enough, in which case:

$$H = \left(\frac{Dn}{1900}\right)^2$$

Question 5-17: How does the nature of a liquid affect pump selection?

Answer: It determines to a certain extent the range of capacities and heads as well as the pump types most frequently used for a given service. The widest range of selections is available for water, which is divided into fresh and salt water. Selection of materials for the latter varies greatly when considered from the viewpoint of first cost, length of life, and the customer's preference and experience. Apparently insignificant impurities in the liquid may be important in selecting the materials.

Similarly, the nature of a liquid will affect not only the pump material, but possibly the mechanical construction best suited to the purpose, depending on whether the liquid is an acid, alkali, or oil. For example, if acid is handled, not only must the pump be built of corrosion-resisting material, but it must avoid leakage of the acid to the atmosphere through the stuffing box.

Question 5-18: How does the temperature of a liquid affect pump selection?

Answer: A standard line of general-service pumps has definite temperature limitations. Higher temperatures may require special materials, water-cooled stuffing boxes, or special mechanical features such as center-line support of the casing. Likewise, for extremely low temperatures, such as in brine service, nickel cast iron or nickel cast steel may be used in preference to standard gray cast iron to obtain a more refined crystalline structure and prevent fractures. Any wide variations in operating temperatures should also be known, as they affect the specific gravity and viscosity of the liquid.

Question 5-19: How does the specific gravity of a liquid affect pump selection?

Answer: The specific gravity must be known to determine the proper size of driver at design conditions. If head is given in pounds per square inch, the specific gravity of the liquid must be known to convert the head into feet of liquid handled. If a customer specifies a net pressure of 100 psi for a fire pump, the total head required is 231 ft for fresh water, or 225 ft for salt or sea water with a specific gravity of 1.03.

Question 5-20: Will a change in the specific gravity of a liquid affect the power required to drive the pump?

Answer: Yes, power to drive the pump varies as the specific gravity. If the power input to a pump is 10 hp when handling cold water, it increases to $10 \times 1.2 = 12$ hp when pumping 1.2 specific-gravity brine, other conditions remaining the same.

Question 5-21: How do the nature and size of solids in a liquid affect pump selection?

Answer: They determine the type of impeller best suited for the purpose and the construction materials of the pump. If the solids are very abrasive, an open impeller is generally used, and, where the pumping job warrants, special and more expensive wear-resisting materials may be justified. If the solids are relatively soft, as is generally true of sewage, a closed impeller can be used. When solids reach a certain size, special non-clogging impellers are required.

Question 5-22: How does load factor affect pump selection?

Answer: If a pump is used in intermittent service, then it is not necessary to use the most efficient unit available, and the selection generally can be made on the basis of lowest first cost. On the other hand, a pump intended for continuous service should have high efficiency, reliability, and long life.

Question 5-23: Does variation in operating conditions affect the selection of pump type?

Answer: The types of systems on which constant-speed centrifugal pumps are generally applied can be classified as:

1. Constant capacity and head.
2. Variable capacity with nearly constant pressure.
3. Variable head with some capacity variation permitted.

Type 1 operation is only at one point on the pump head-capacity curve so that its shape is not important. Type 2 is common when a pump discharges into a system like the water supply main of a community or industrial plant, in which nearly constant pressure is maintained irrespective of demand. Here a flat head-capacity curve is desirable. Type 3 occurs when there is a variation in static head, such as when filling a tank from the bottom. In some installations, considerable capacity variation can be tolerated because of static head changes, but in others the capacity variation must be as small as possible. The latter requires pumps with steep head-capacity curves, and sometimes for the desired results it is advantageous to sacrifice efficiency.

There are many other cases where variations in operating conditions are such that pumps of special characteristics are required. In some of them, the desired results are obtained only by variable speed or throttling.

Question 5-24: How can reduced pump capacity be obtained with constant-speed operation?

Answer: This is done by throttling the pump discharge to increase the friction head in the system. The capacity at which a centrifugal pump operates is determined by the intersection of its head-capacity characteristic curve with the system-head curve, as explained in Question 4-34. Throttling the discharge of the pump increases the friction head in the system. Consequently, a new system-head curve results, such as that indicated by the artificial system-head curve (Fig. 4-11). The point of intersection of the head-capacity curve of the pump with the new system-head curve determines the new operating capacity of the pump, as indicated

Question 5-25: Will throttling a centrifugal-pump discharge build up excessive pressure?

Answer: Unlike reciprocating pumps, throttling the discharge of a centrifugal pump cannot cause excessive pressure. For example, in Fig. 4-11, if a valve is slowly closed, the head developed by the pump increases until at shutoff it reaches 190 ft, or 16.5 per cent above that at design capacity. Head at shutoff varies with the pump type, or specific speed, but under no conditions will a centrifugal pump develop an excessive head at shutoff. However, do not operate any centrifugal pump with the discharge valve completely closed, or it will overheat.

Question 5-26: Will throttling a centrifugal-pump discharge overload its driver?

Answer: This depends on the shape of the power-capacity curve and the selection of the driver size. In pumps of low and medium specific speeds, power consumption is lowest at zero discharge and rises with capacity increase. Thus throttling the discharge of such a pump cannot overload the pump driver.

With pumps of extremely high specific speed, the shape of the power-capacity curve is such that maximum power consumption occurs at shutoff, decreasing with an increasing flow rate. If the driver is so selected as to correspond to maximum power consumption, throttling the pump discharge cannot overload the driver. If, however, as is generally the case, the driver is selected on a basis of power required at normal operating head, throttling the pump discharge below a certain minimum capacity causes overloading.

Question 5-27: When are two or more pumps used in parallel?

Answer: Demand in pumping installations varies over a wide range. If a pump were large enough for the maximum flow, it would be inefficient at reduced capacities and would develop excess head. Also, where high reliability is required, as in a waterworks, there would have to be a duplicate unit of the same size as a spare.

For such installations, it is best to have two or more smaller pumps so that capacity can be adjusted to demand. Sometimes such units have equal capacity; at other times they are of different sizes. Waterworks and similar services have a sufficient number of units so that one or more of them is in reserve.

Even when constant capacity is needed, two or more pumps operating in parallel are often desirable or necessary. Some installations are so large that a unit to handle full capacity is not feasible. Other setups use units of one-half, one-third, one-fourth, or smaller capacity, with one or more spares that together have a lower installed cost than two full-sized units. Still other plants use two or more pumps in parallel so that the most efficient speed of existing drivers can be utilized.

Question 5-28: Must two pumps be identical before they can operate in parallel?

Answer: Satisfactory operation of pumps in parallel depends as much upon the system-head curve on which they are to operate as on the pumps themselves. If the system head permits, pumps can have dissimilar characteristics. The same combination might not be satisfactory for other system characteristics. When pumps operate in parallel on a system where flow is regulated by throttling (most boiler feed-water systems are of this type), they should have a similar and stable characteristic with practically the same shutoff head. For such applications, units may safely have unequal capacities, provided the shutoff heads and general shape of the head-capacity curves are similar.

Question 5-29: When are two or more pumps used in series?

Answer: When the total head of an installation exceeds that which can be developed in commercially available single-stage pumps, multistage pumps are normally used. Multistage pumps

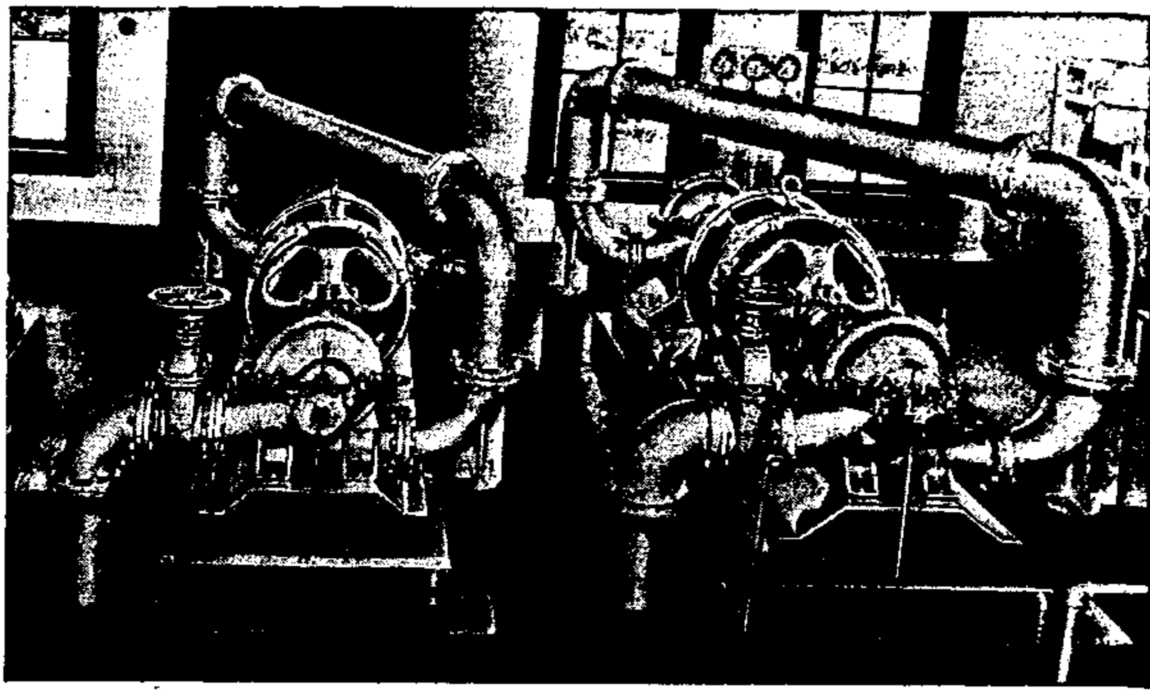


FIG. 5-4. Two series pumping units, each consisting of two single-stage double-suction centrifugal pumps with interconnecting series piping. (Motors are between stages.)

are commercially available only in the smaller sizes, with some coverage in intermediate sizes and with practically none for pumps of over 12-in. discharge size. Practically the same results accomplished with a two- or three-stage pump are obtained by connecting two or three single-stage pumps in series by external piping (Fig. 5-4) instead of the integral water passageways generally used in multistage pumps.

Series units are most generally applied for heads of 250 to 500 ft and are generally two single-stage double-suction pumps in series. Generally, series units are arranged to be powered by a common driver, but separate drivers for each stage can be used. Sometimes waterworks use two pumps in series for small capacities rather than a multistage pump, often thus obtaining a better pumping efficiency.

Question 5-30: Must two pumps be identical before they can operate in series?

Answer: Characteristics of pumps in series do not have to be similar for satisfactory operation because the head developed by one pump adds to that of the other. For an efficient unit, capacities at best efficiency for the individual pumps should be about the same. If one pump has a larger capacity than the other, it should preferably be used for the first stage.

Question 5-31: Why are pumps comprising a series unit usually identical?

Answer: While other combinations are possible, most series pumping installations consist of two identical single-stage double-suction pumps. With identical pumps:

1. Most or all parts of the individual pumps are interchangeable, and a smaller number of spare parts is necessary to guard against outages.
2. Unit cost is generally, but not always, the lowest.
3. A common driver can be used, which permits selection of the best operating speed and obtains highest over-all efficiency.

Question 5-32: What is the normal arrangement of pumps operating in series with a single driver?

Answer: One arrangement commonly employed with small units has a double-extended-shaft motor between the two pumps in series (Fig. 5-4). For larger units and, when necessary, for small units, single-end drivers are used, and one pump is built with a double-extended shaft. Then the shaft of the pump next to the driver must transmit the total horsepower required by the unit. This arrangement is used with steam turbines and most internal combustion engines because they are not suitable for double-shaft-extension drive applications.

Question 5-33: What is a tandem-driven pumping unit?

Answer: In some industries, two or more pumps are necessary for different steps in a process. If the service conditions permit the same rotative speed and if the pumps can be placed together, it is often advantageous to use a common driver on one end. Such units are called "tandem driven." Strictly speaking, two pumps in series, with the driver at one end, form a tandem-driven unit.

Question 5-34: What prime movers can be used to drive centrifugal pumps?

Answer: Probably every form of prime mover and source of power has been used, with some form of intermediate transmission if necessary, for centrifugal-pump drives. Most centrifugal pumps, however, are driven by electric motors. Steam turbines, gasoline engines, diesel engines, gas engines, and water turbines are also used. Steam engines are now rarely used.

Question 5-35: Is the type of pump used affected by the type of driver?

Answer: The choice of rotative speed is limited with certain types of drivers, for example, gasoline engines. Thus the speed for the ideal pump for the service conditions may be much higher or much lower than the driver speed. For units in constant use, a speed-increasing or -decreasing device is warranted. For units to be used occasionally, like a stand-by unit, a compromise pump design can often be used, thus saving the cost and complication of a speed changer, at some sacrifice in pump performance and, in some cases, with some increase in pump cost.

Question 5-36: What factors influence the selection of driver size for a centrifugal pump?

Answer: A number of local conditions affect permissible loading of various drivers. Safe loading of an electric motor depends on its design as well as on the temperature and density of the surrounding air and variations in line voltage. Other types of drivers are also affected by local conditions. The driver should be capable of driving the pump over its entire operating range under the most adverse conditions.

Where the driver is suitable for variable-speed operation and a decrease in head is acceptable, the pump is sometimes operated at reduced speed to keep the maximum power within the rating of the driver.

Question 5-37: When selecting a motor for driving a centrifugal pump, should a normal- or high-starting-torque type be used?

Answer: Normal-starting-torque motors develop more than sufficient torque for starting centrifugal pumps, even with the discharge valve open. All centrifugal pumps require low torques to start. The torque required to start from rest generally does not exceed 15 to 20 per cent of normal operating value. It is generally highest in a sleeve-bearing pump if it has been idle for some time, causing the packing to stick to the shaft sleeves and the oil to be squeezed out from between the shaft and bearings.

Question 5-38: Will operation of a centrifugal pump at heads other than its design value overload the driver?

Answer: The power required to drive a centrifugal pump from shutoff to maximum capacity varies with its specific speed and its individual design. For the lower specific-speed types, the power requirement is minimum at shutoff and maximum at or near maximum capacity. Operation of the pumps at heads below normal increases the load on the driver and sometimes causes an overload.

On the other hand, a reduction in head on high-specific-speed pumps causes a reduction in power. An increase in operating head on these pumps causes an increase in power consumption and sometimes may overload the driver.

Question 5-39: Can a centrifugal pump overload an electric motor?

Answer: Yes, if the pump horsepower increases with a change in capacity. It is thus dependent upon the shape of the pump power-characteristic curve and system-head characteristics.

Question 5-40: Can a centrifugal pump overload a steam turbine?

Answer: No, it cannot. With a fixed throttle position, a steam turbine develops a constant torque, and the speed of the unit is always at a value where pump torque equals turbine torque. If the turbine is equipped with a constant-speed governor, a continued increase in torque causes the governor to open the throttle valve more and more until it is wide open. If the pump torque exceeds the turbine torque at the set governor speed, the unit slows down until the pump torque equals that developed by the turbine. This is also true when the turbine is equipped with a constant- or excess-pressure regulator, the only difference being in the impulse that tends to maintain the required torque balance.

Question 5-41: What factors must be considered in selecting the size of internal combustion engines to drive centrifugal pumps?

Answer: Some internal combustion engines are rated for the load they can carry continuously at rated speed, while others are rated for the load they can develop on test with full-open throttles. In the latter, power ratings may or may not include power required by the normally attached auxiliaries, such as circulating pump, fan, etc. Engines of the automotive type are generally rated for their maximum developed power on dynamometer test while stripped of all auxiliaries. Such engines usually operate safely with a continuous loading of only 75 to 80 per cent of their developed power.

Most automotive engines are high-speed and have a short life when run continuously. There are available medium-speed engines that have a longer life and low-speed ones that have a long life with minimum maintenance.

Most internal combustion engines that are rated on the basis of load they will carry constantly will also carry an overload. Engines rated on maximum developed power can be operated loaded to more than 75 to 80 per cent of their developed power. But if other than momentary, this will generally result in mechanical failure. The engine best suited for any particular application must therefore be selected with consideration of service requirements.

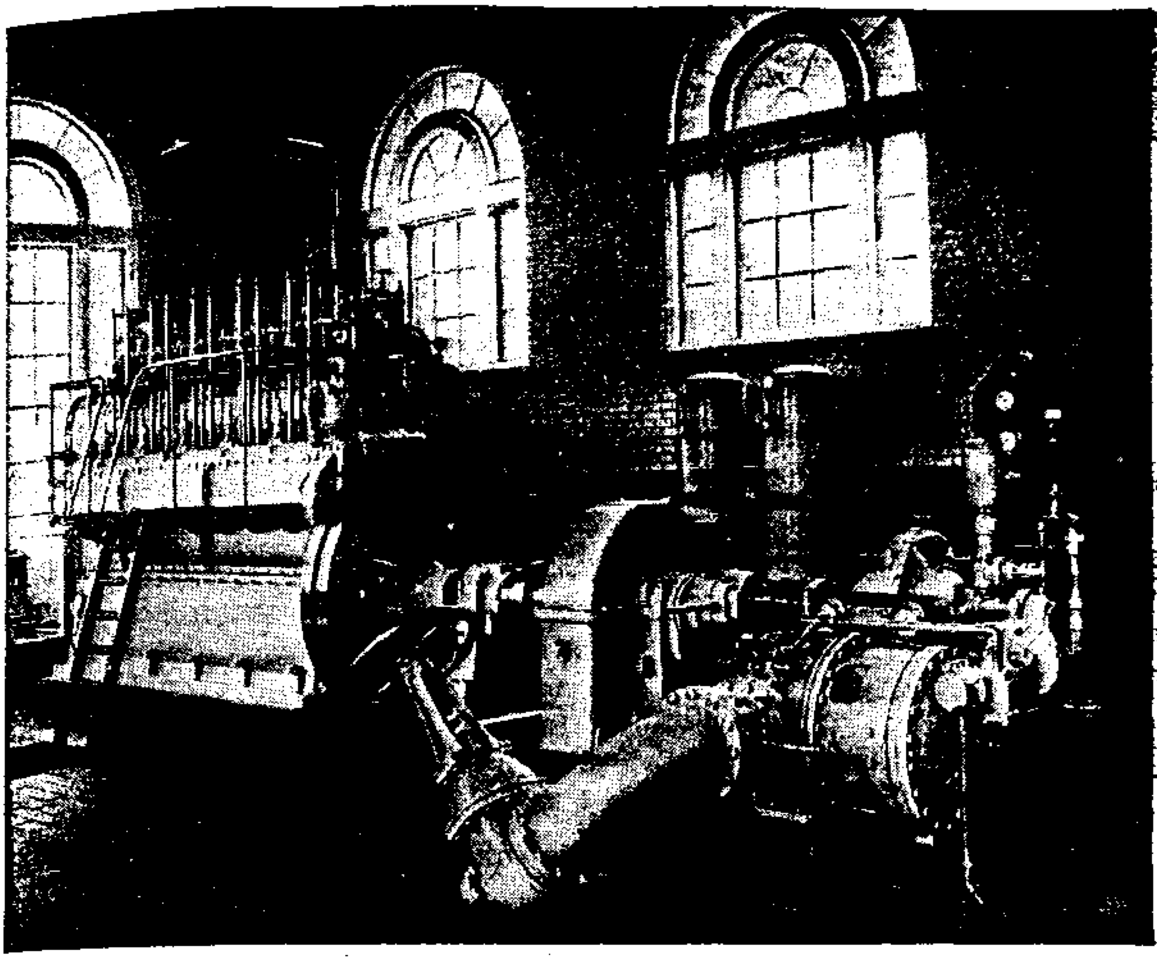


FIG. 5-5. A centrifugal pump in a water works connected through a step-up gear to a low-speed high-duty diesel engine.

Question 5-42: Does the variation in operating conditions affect the driver selection?

Answer: When centrifugal pumps operate under variable head or discharge, or both, drivers that permit economical speed variation are preferred. Choice of drivers suitable to variable-speed operation is quite wide: steam turbines, internal combustion engines, wound-rotor a-c motors, and sometimes d-c motors. Synchronous or squirrel-cage motors are also used with a variable-speed transmission, such as a hydraulic or magnetic drive.

Question 5-43: What advantages can be derived from variable-speed drive?

Answer: There are a great number of centrifugal-pump applications where operation at varying capacity and/or total head may be required. Since the operating conditions of a centrifugal pump are determined by the intersection of its head-capacity curve with the system curve, the only means available to vary its operating conditions, if the pump is operating at constant speed, is the alteration of the system head by throttling. In many cases it is more practical and more economical to change the point of intersection of the pump and system curves by varying the operating speed and, hence, the head-capacity curve of the pump.

Question 5-44: Is variable-speed drive always more economical than constant-speed drive when operating conditions are variable?

Answer: No, it is not. Many factors enter into the evaluation of the most economical operating method under such conditions. While considerable horsepower may be saved in varying the speed of a steam turbine or of an engine, the efficiencies of wound-rotor motors and of hydraulic and magnetic transmissions are practically proportional to their operating speed. Power saving, if any, must be considered along with the increased cost of the drive over a constant-speed drive. In some applications, such as boiler feed service, power losses due to throttling the pump discharge are converted into heat that is returned to the boiler and must be taken into account in any evaluation of variable- and constant-speed drives.

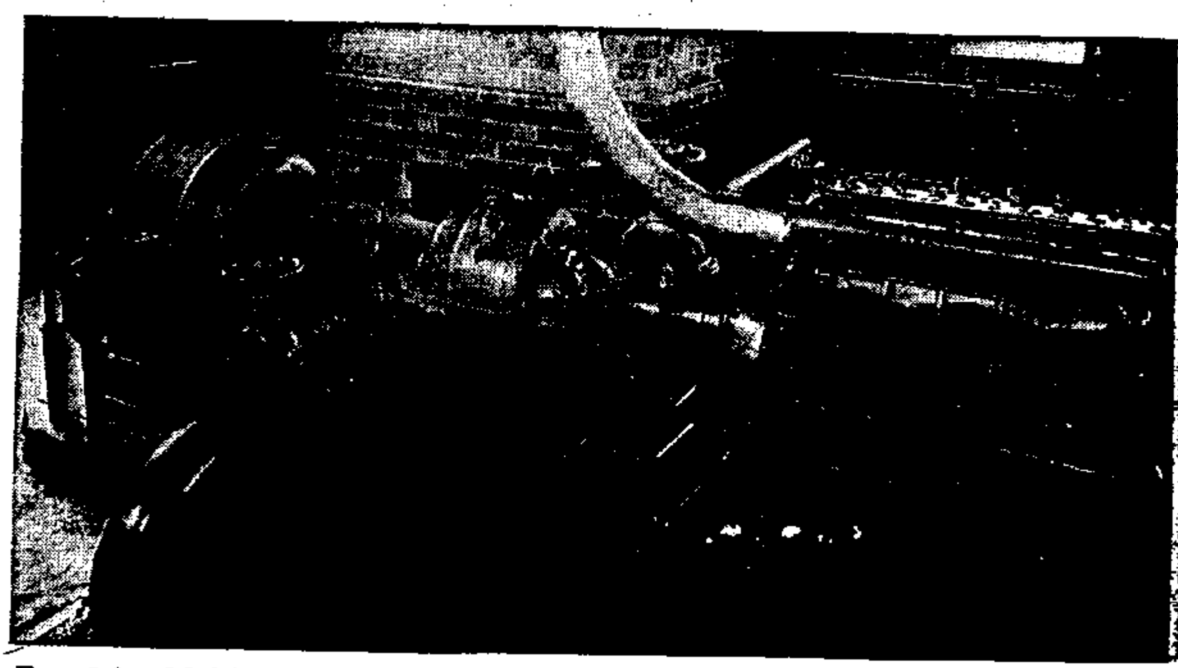


FIG. 5-6. Multistage pump, dual-driven by squirrel-cage motor or gasoline engine.

Question 5-45: What is a dual-driven pump?

Answer: It is a pump arranged so that it can be driven by either of two drivers (Fig. 5-6). Generally the two are different, one, such as an electric motor, for regular service, and the other, say, a gasoline engine, for use when power to the motor fails.

Question 5-46: How are dual drives usually arranged?

Answer: One driver is generally connected to each end of the pump, so that the one not being used can be disconnected. This arrangement requires a double-extended pump shaft, and end-suction pumps cannot be used. Also, pump selection is often compromised to suit the available drive speed.

Question 5-47: Can two drivers be used on one centrifugal pump, each taking part of the load?

Answer: Yes, in a few installations the pump is driven by two power units, each carrying part of the load.

The most common installations of this kind combine an electric motor and a water or steam turbine. For such an application, a water turbine without a governor is used. Where the other power unit is an induction motor, pump speed varies slightly with changes in operating conditions to maintain a balance in which the power developed by the two drivers equals that needed by the pump.

CHAPTER 6

INSTALLATION OF CENTRIFUGAL PUMPS

Question 6-1: Are there general rules for the preferred location of a pump?

Answer: Yes, here are five:

1. Install the pump in a light, dry, clean room. If the location is cramped, dirty, wet, or poorly lighted, a unit is difficult to dismantle and repair and is likely to be neglected by the operators.
2. If a unit must be placed where it might be submerged, either install a wet-pit pump or make provision to remove incoming water, using auxiliary wet-pit drainage pumps, so that neither the main pump nor its driver will be flooded.
3. If a motor-driven unit is to operate in a damp or moist location, a motor construction, suitable for such conditions, must be obtained. The various motor manufacturers make special designs for such adverse operating conditions, as well as for installation in the open.
4. Locate the pump as close as possible to the supply source. If possible, place it below the liquid level of the suction reservoir. Always be guided by the manufacturer's recommendations for suction conditions.
5. Install it where a substantial foundation can be provided.

Question 6-2: Can suction-pipe design affect pump performance?

Answer: Yes, in two ways. If the suction pipe is too small, friction losses combined with static lift may cause a total suction lift greater than that on which a pump operates successfully. This reduces pump capacity by an amount depending upon the system characteristics as the pump is "operating in the break."

Elbows and other fittings in the pump suction line may disturb liquid flow at its suction nozzle and to the impeller. This commonly occurs with double-suction pumps when an elbow turns toward one side of the pump (Figs. 6-1 to 6-3). On a horizontal-shaft double-suction pump, an elbow in the vertical plane pointing down or up can be used. If the elbow must be turned toward one side, use a special design with center baffle to guide the flow around it, or install a straight run of pipe, three pipe diameters or longer, between the elbow and suction nozzle (Fig. 6-1).

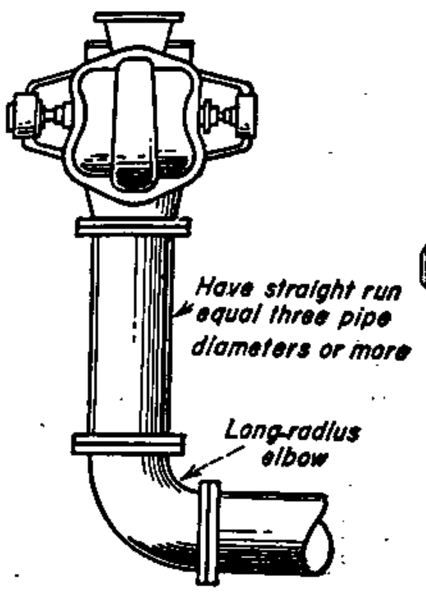


FIG. 6-1.

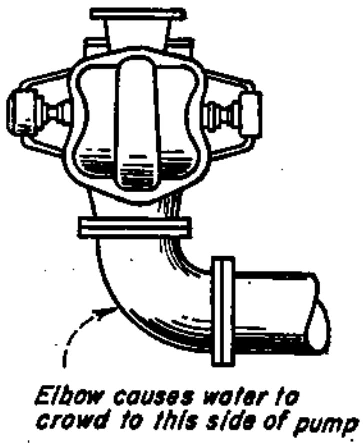


FIG. 6-2.

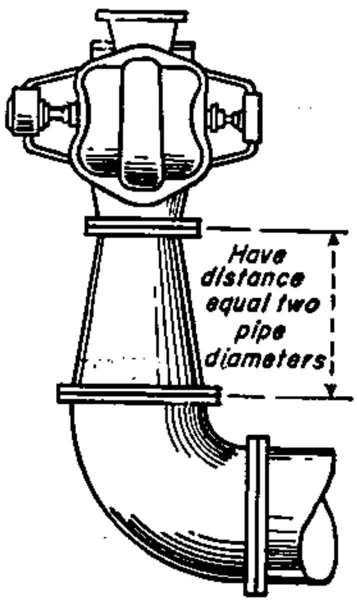


FIG. 6-3.

FIG. 6-1. Correct suction connection for a double-suction pump with side-turned suction elbow of same size as pump suction.

FIG. 6-2. Incorrect suction connection for a double-suction pump with side-turned suction elbow of same size as pump suction.

FIG. 6-3. An eccentric reducer of adequate length can be used to connect a side-turned suction elbow of larger size to pump suction.

Question 6-3: What precautions are necessary in the design and location of suction-line inlets?

Answer: They should be of a design and proportions that have low friction losses and must be sufficiently submerged so that vortexes will not form and suck air into the pump. Use a bellmouth on the inlet of the suction line to reduce inlet loss. On vertical lines, locate the bellmouth far enough above the bottom and away from all walls of the suction well so that flow to the bellmouth is not restricted. Place it far enough below low-water level to prevent whirlpools. Horizontal suction lines coming into a pit or tank horizontally do not generally involve flow restrictions but must be submerged enough to prevent sucking air.

Simple rules cannot be given that ensure satisfactory suction inlets under all conditions. Even in apparently excellent layouts, trouble may occur because of some unforeseen reaction in liquid flow. Then baffles must be installed or some design change made to correct the trouble. Vortexes drawing air into the suction line can frequently be stopped by boards or planks floated in the water surface around the suction pipe.

Question 6-4: How should a suction line be constructed?

Answer: The liquid should be delivered to the suction nozzle without disturbance and with sufficient absolute suction pressure to permit proper pump operation. If it operates with a suction lift, the line must be airtight for the existing vacuum.

Question 6-5: How can the size of a suction line be determined?

Answer: Suction-line size depends primarily on the friction-head loss that can be permitted. Generally, on a pump operating with a suction lift, suction-pipe size must be one or two pipe sizes larger than the suction nozzle. When pumps operate with a positive suction head, higher friction losses can be permitted, and pipe may be of smaller size than when a suction lift exists.

Question 6-6: Can the suction pipe be smaller than the pump suction nozzle?

Answer: Generally this pipe should be larger or at least equal in size to the suction nozzle. In certain rare instances, however, when circumstances require a pump to operate always at part of its normal design capacity, it is permissible to use suction piping smaller than the pump's suction nozzle. Here the resultant increase in friction losses would have no appreciable effect on pump performance. However, provide a suitable increaser in the suction piping, located well away from the pump to avoid disturbances at its entrance.

Question 6-7: Why are long-radius elbows used in suction-line piping assemblies?

Answer: Because they have less friction loss than standard elbows. Flow through a long-radius elbow is more uniform and there is less chance of disturbed flow, which would affect pump performance.

Question 6-8: What precautions should be taken in the design of a suction header supplying a number of pumps?

Answer: When a common suction header serves a number of pumps, taper it so that velocities at all points are the same. Since

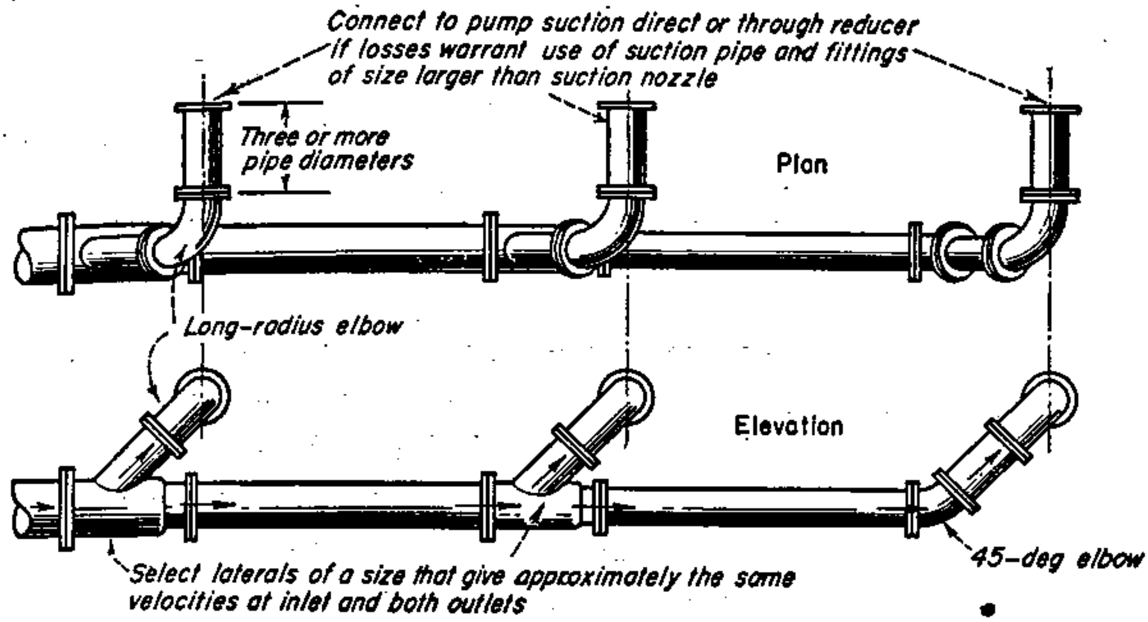


FIG. 6-4. Recommended suction-header design for three pumps served by a common suction line.

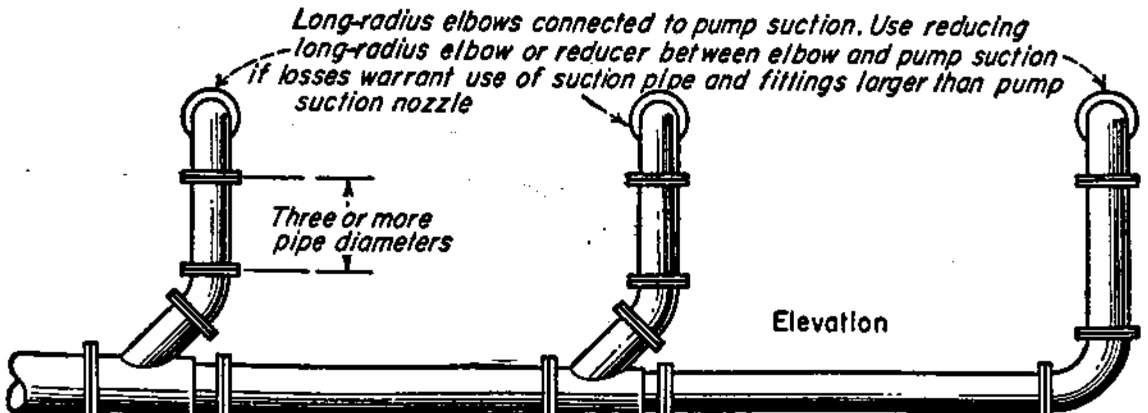


FIG. 6-5. Alternate layout to Fig. 6-4 if suction header is to be located at lower elevation.

this cannot always be done with commercial pipe sizes, some compromise may be necessary. Reducing laterals are generally most desirable for such a header (Figs. 6-4 and 6-5). A header of constant size with smaller outlets is not as good as one with reducing laterals, but is often used in waterworks where the header is a loop with valves so that any section can be isolated in an emergency. Even with the low velocities used for such installations, water distribution to the pumps is sometimes unequal.

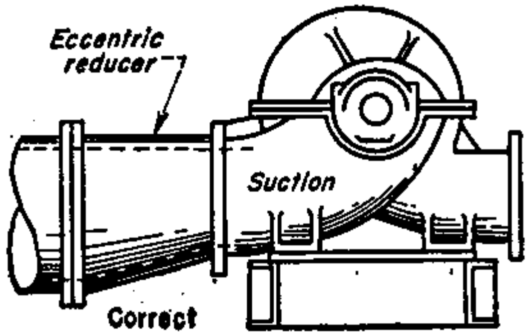


FIG. 6-6.

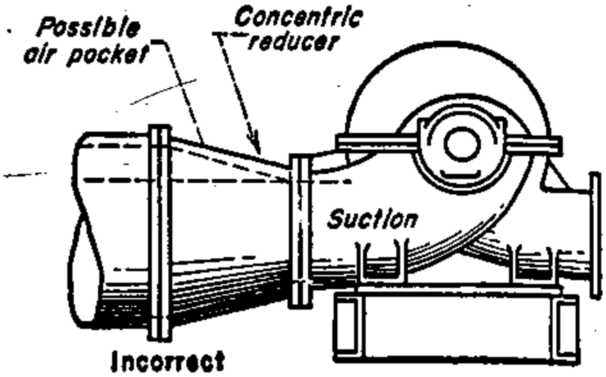


FIG. 6-7.

FIGS. 6-6 and 6-7. Eccentric reducers, Fig. 6-6, should be used for suction piping to avoid air pockets that can result with concentric reducers, Fig. 6-7.

Question 6-9: Why are eccentric reducers used on suction lines?

Answer: Eccentric reducers (Fig. 6-6) are used so that the tops of adjacent sections of different sizes may be at the same elevation. Thus air pockets that may form when concentric reducers are used, are avoided (Fig. 6-7).

Question 6-10: Why do pump manufacturers caution against loops in suction lines?

Answer: Because air pockets may form in them and affect pump performance adversely. It is possible to operate a pump with an upward loop in the suction line if the high point is vented through a vent valve to a vacuum system in which a lower absolute pressure is maintained than in the suction line at the loop.

Question 6-11: Why are siphon suction lines sometimes used in the chemical industry?

Answer: In the chemical industry there are many cases where the corrosive properties of liquids or solutions make it undesirable to have tanks with outlets in the side or bottom. Thus, the suction line of a pump drawing from such a tank must enter the tank from the top. If there is no floor at the level of the top of the tank, the

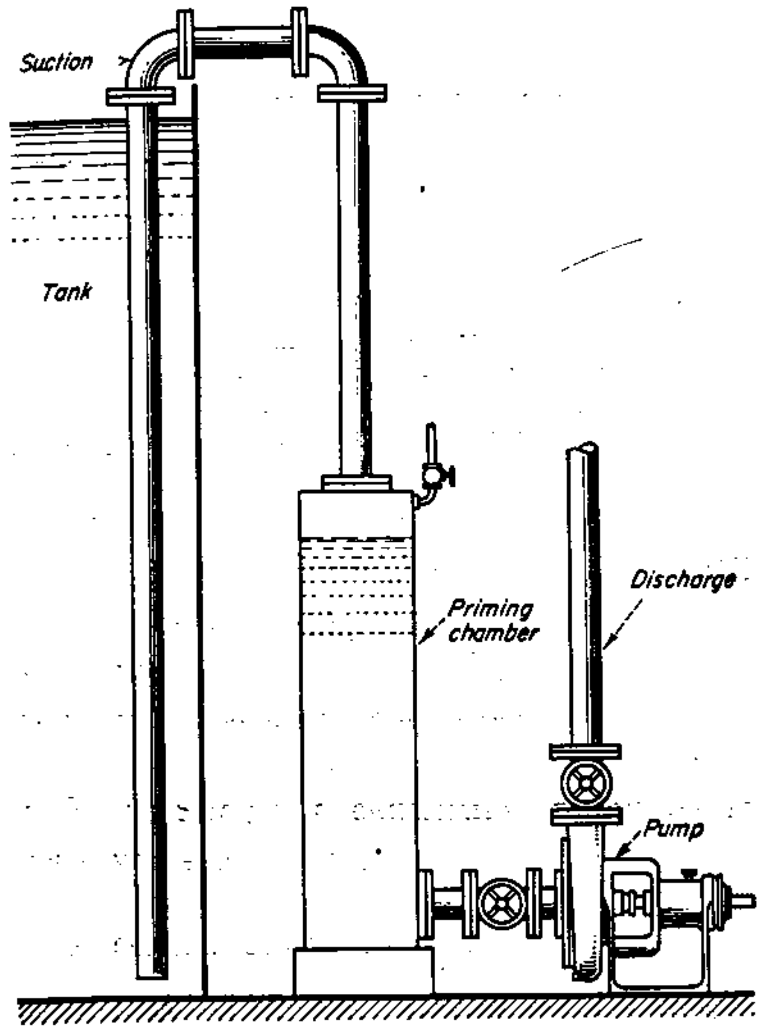


FIG. 6-8. Siphon-type suction line on installation using priming chamber.

pump is generally placed on the floor adjacent to the bottom of the tank, and a siphon suction line is used (Fig. 6-8), with or without a priming tank. Care must be taken to exhaust all the air from the siphon before starting. With a simple siphon without a priming chamber, a careful watch has to be made that the siphon does not become broken. The maximum height of the siphon and permissible size of the pipe must keep the absolute pressure at all points above the vapor pressure of the liquid.

Question 6-12: When should screens be used in a suction line?

Answer: They should be provided on any suction supply in which there may be foreign material larger than that which passes through the pump easily, or where this material is objectionable in the liquid. Screens vary with the kind, size, and amount of material to remove. Sometimes, in place of screens, strainers are used. While more accessible, they usually require frequent clearing and generally cause higher friction losses.

Even when screens are not necessary in normal operation, a temporary strainer is recommended in the suction line of a newly installed centrifugal pump. This strainer catches foreign material inadvertently placed in the piping, such as bolts, nuts, and even a wrench or two that might damage the pump.

Question 6-13: Why do some centrifugal pumps have vents on the suction side of the impeller?

Answer: Hotwell pumps and some in process work handle liquids under absolute pressures corresponding to or near their vapor pressure. Under adverse operating conditions, such pumps can become vaporbound. To prevent this, vents are provided on the suction side of the impeller to vent vapors either back to the suction source above liquid levels or to some suitable vessel.

Question 6-14: Under what conditions does air trapped in a volute-casing top affect pump operation?

Answer: In most medium- and high-head centrifugal pumps, air trapped in their volutes is washed out by the flow of water. With lower head pumps and when air leaks into the suction line or through the stuffing boxes, this air is not washed out of the volute and remains to reduce pump capacity.

In certain applications, washing the air out into the discharge is objectionable. For instance, a boiler feed pump should not feed air into the feed system, and it must be vented out of the pump casing.

Question 6-15: What is the purpose of a check valve in the discharge line?

Answer: In most installations, a gate and a check valve are used between pump and discharge piping. The check valve prevents reverse liquid flow when the pump stops for any reason. The gate valve should be closed if the pump is shut down for an extended period, to prevent leakage back through the check valve.

Question 6-16: Are flap valves used with centrifugal pumps?

Answer: The usual form of flap valve is a swing disk attached to the end of the discharge pipe. It serves as a check valve and is used the same as a regular check valve, particularly in low- and moderate-head pumping installations discharging into a body of water such as a canal, near or adjacent to the pumping station.

Question 6-17: When are expansion joints used in suction or discharge piping?

Answer: They prevent transmitting piping strains to the pump. Frequently on pumps handling hot liquids, these joints compensate for pump and piping expansion.

When expansion joints are used, take precaution in the design of the foundation and in mounting the pump on it so that reactions resulting from flow and pressure conditions are properly absorbed. Advise the pump manufacturer of these conditions, especially when high head pumps are involved. Thus proper provision can be made to transmit the thrust to the bedplate and foundation (Fig. 6-9).

Question 6-18: Why are vent valves used on the discharge of a pump?

Answer: They permit escape of air, gas, or vapor trapped in the casing when liquid is admitted to the pump. While sometimes these gases would be dislodged by the liquid's flow, air or gas in the discharge is objectionable in most systems and should be vented from the pump.

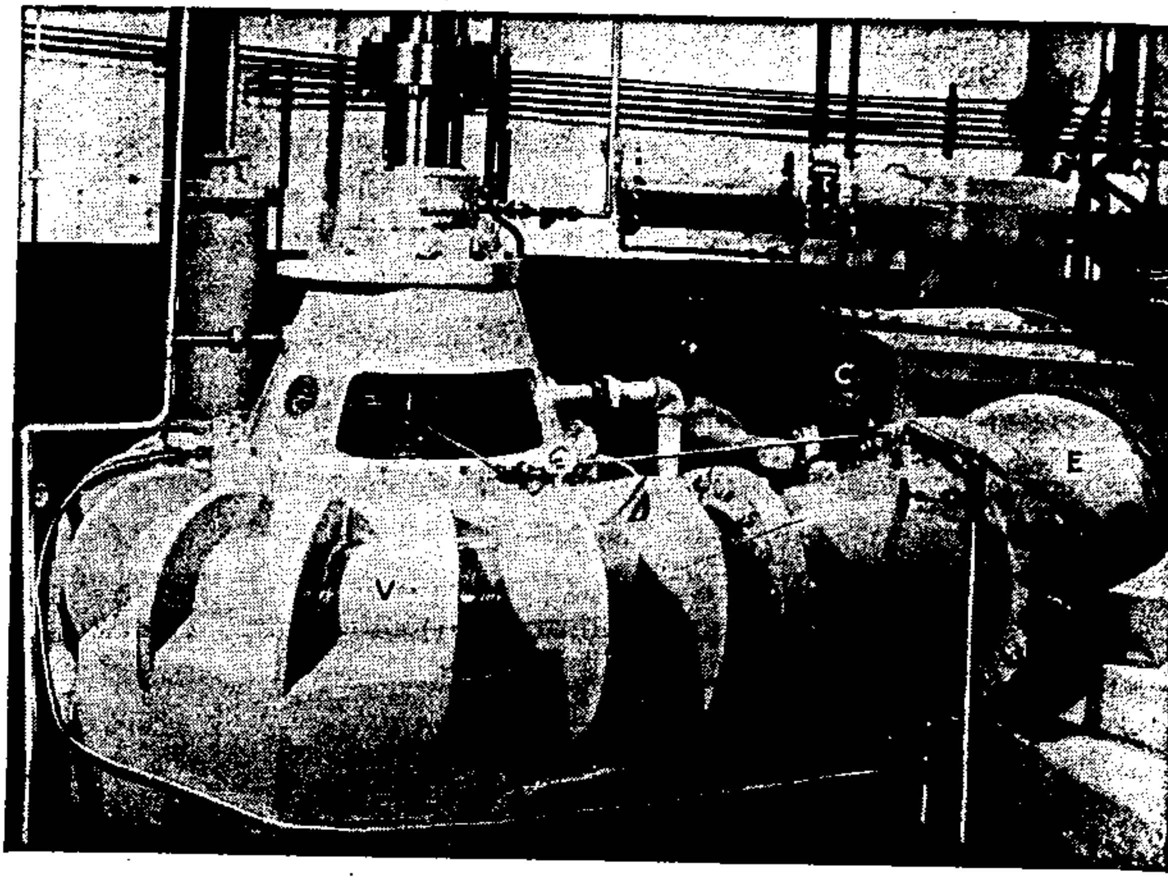


FIG. 6-9. In installation shown, inadequate provision had been made originally to take the hydraulic thrust resulting from a flexible pipe connection in the discharge line beyond valve *C*. It was corrected by building the foundation up to anchor the 45-deg elbow *E* and the pump volute *V*.

Question 6-19: Where should the suction vent be led?

Answer: A suction vent on a centrifugal pump removes vapor, air, or gas from the suction waterways at the suction eye of the impeller. It must therefore be piped to some space of slightly lower absolute pressure than at the suction eye—for example, above the liquid level of the source of supply, where the vapor can be condensed or from which vapor, gas, or air can be removed. On a condensate pump, the suction vent is piped to the condenser shell or hotwell above the condensate level. Here the vapor is condensed and any gas freed is removed by the air ejector. On a process pump or for petroleum refinery service, a suction vent may have to be connected to a separate vessel in which a lower pressure is maintained.

Sometimes where the suction pressure is above atmospheric, the pump can be vented, with proper precautions, to atmosphere. A suction vent on a pump for priming is connected to the vacuum-producing device.

Question 6-20: What arrangement is necessary to dispose of stuffing-box leakage and other drains?

Answer: Pipe all drain connections to a sump or sewer. Often the disposal of pump-drain liquid requires a separate sump pump or a water-jet eductor to pump this waste to a sewer.

Question 6-21: What is standard equipment for a horizontal centrifugal pump?

Answer: Standard equipment generally includes the pump with necessary packing, water-seal piping, seal cages, dowels, plugs, grease cups or oil gauges, cooling-water piping if required, vent cocks, and special wrenches—that is, all parts needed for a complete pump on usual applications. Often many of these parts are not assembled but are separately boxed or bagged and shipped with the pump.

Question 6-22: What is a firewall extension?

Answer: When a pump handles an inflammable volatile liquid like gasoline, there is a hazard unless the driver is explosion-proof. To avoid this risk, pump and driver are placed in separate rooms with a firewall between them. This arrangement requires a shaft extension with a stuffing box so that the driver can be connected to the pump through the wall. Various types of firewall extensions have been used, some of which are shown in Figs. 6-10, 6-11, and 6-12. These extensions have a seal at each end made of a close-fitting felt gasket or of one or more rings of packing with a gland. The space between the two seals is usually filled with a light grease to form a vaportight seal.

Question 6-23: Are a centrifugal pump and its driver checked for alignment at the factory before shipment?

Answer: Yes. When a unit consisting of pump, base, coupling, and driver is assembled at the factory, the base plate is placed on a flat, even surface. Pump and driver are mounted on it and the coupling halves accurately aligned. Shims are used under the mounting surfaces of pump and driver where necessary to obtain

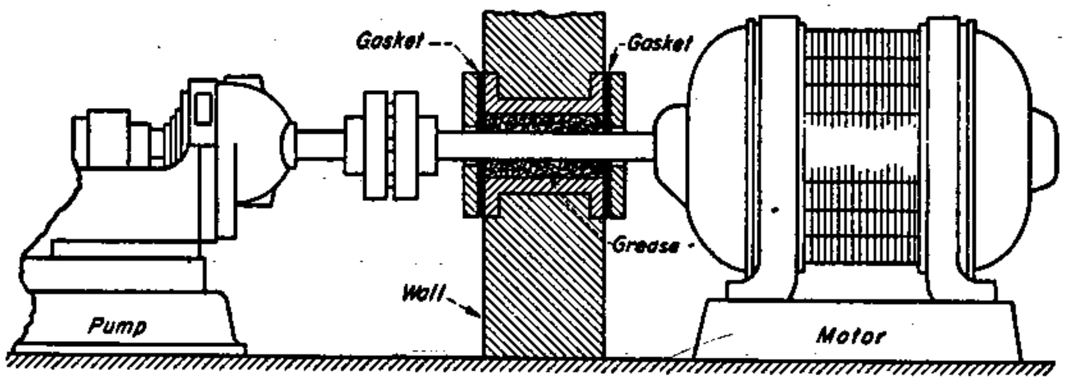


FIG. 6-10. Firewall extension using extra-long motor shaft.

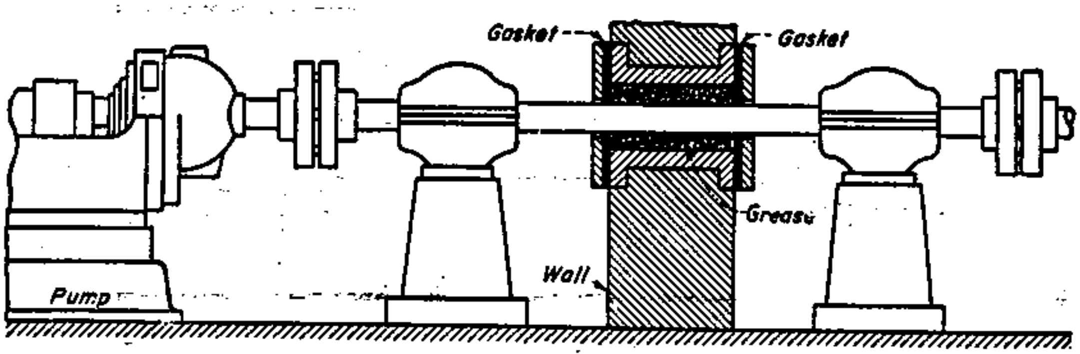


FIG. 6-11. Firewall extension provided by separate shaft supported by two pedestal bearings, one on either side of wall.

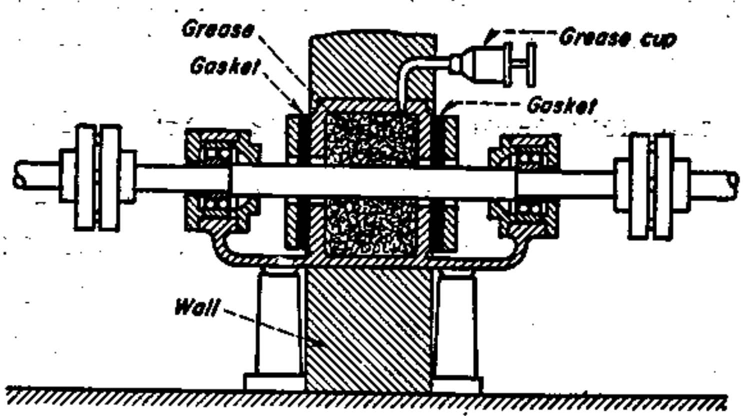


FIG. 6-12. Firewall extension similar to Fig. 6-11 in principle but with ball bearings supported by brackets.

alignment. The pump, but not the driver, is generally doweled to the baseplate at the factory. The driver is not doweled because field duplication of factory alignment is thus simplified.

Question 6-24: Why should a pump be realigned in the field?

Answer: All base plates are flexible and subject to shipping strains. When a unit is installed, a careful check is necessary to determine whether factory alignment has been disturbed.

Question 6-25: How is a pump leveled on its foundation?

Answer: Pumps are generally shipped mounted, and it is usually unnecessary with units of moderate size to remove pump or driver from its base plate when leveling. Place the unit over its foundation, and support it on short strips of steel or shim stock close to the foundation bolts. Allow from 0.75 to 2 in. space between base plate and foundation top for grouting. Have the shim stock extend across the supporting edges of the base plate. *Remove coupling bolts before proceeding with leveling of unit and alignment of coupling halves.*

A small spirit level can be used on the projecting edges of pads supporting pump and motor feet, when scraped clean, for leveling the base plate. Where possible, it is preferable to place the level on some exposed part of the pump shaft, sleeve, or planed surface of casing. Adjust the steel supporting strips or shim stock under the base plate until the pump shaft is level, the flanges of suction and discharge nozzles vertical or horizontal as required, and observe that the pump is at the specified height and location. When the base plate has been leveled, make the nuts of the foundation bolts hand tight.

While leveling the pump and base, maintain accurate alignment of unbolted coupling halves on pump and driver shafts.

Question 6-26: How are a centrifugal pump and its driver aligned?

Answer: Make this alignment with the pump and driver coupling disconnected. Revolve the pump and driver rotors by hand to ensure that they run freely. Place a straightedge across the top and sides of the coupling (Fig. 6-13). Check the faces of the coupling halves for parallelism by a tapered thickness gauge or feeler gauges (Fig. 6-14). Alignment can also be checked with a dial indicator clamped to one half of the coupling and resting on the other half.

When the coupling peripheries are true circles of equal diameter and the faces flat, exact alignment exists when a straightedge lies squarely across the rims at any point and the distance between the faces is equal at all points. If one coupling half is higher than the other, the difference may be determined by straightedge and feeler gauges. If the faces are not parallel, the thickness gauge or feelers show a variation in the distance apart at different points.

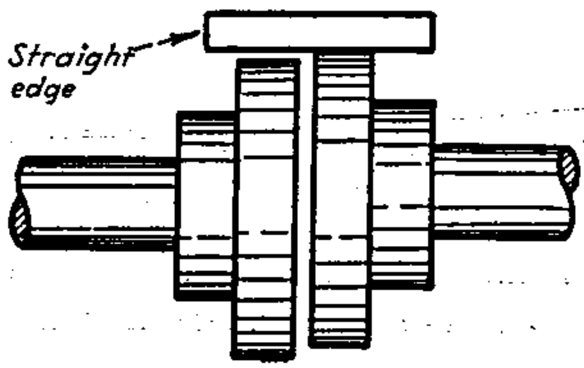


Fig. 6-13.

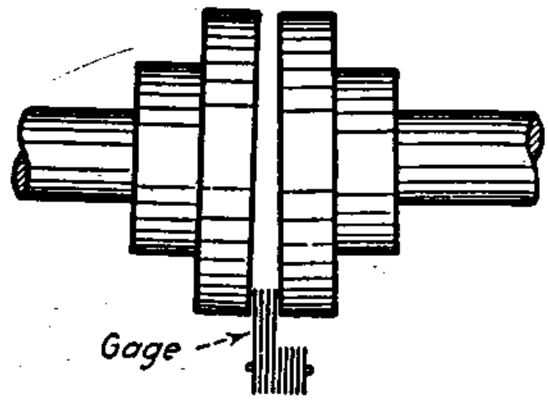


Fig. 6-14.

Fig. 6-13. Parallel alignment of a pump and its driver connected by coupling having cylindrically turned peripheries can be checked by using a straight edge.
Fig. 6-14. If coupling faces are perpendicular to the shaft, angular alignment can be checked by feeler gauge at each 90 deg.

Alignment must be rechecked after suction and discharge piping have been bolted to the pump, to test the effect of piping strains. When handling hot liquids, disconnect the nozzle flanges after the unit has been in service, to check the direction in which the piping expansion is acting. Correct for strain effect as required to obtain true flange alignment.

Question 6-27: What precautions must be observed when a pump and its driver operate at appreciably different temperatures?

Answer: Where a steam turbine drives a pump, make final alignment with the driver heated to its operating temperature. If this is not possible at the time of alignment, make allowance for the temperature effect by setting the turbine shaft when cold a proper amount lower than the pump shaft. Similarly, if pump its entire area, and hence is termed an "unbalanced" valve. The flat or "D" valve is satisfactory for steam pressure up to 175 or 200 psi and should have reasonable service life, particularly where steam-end lubrication is permissible. On large pumps, the force required to move an unbalanced valve is considerable, and a balanced piston valve (Fig. 13-3) has advantages. For high-pressure and highly superheated steam, the balanced valve is preferable for small pumps as well as large.

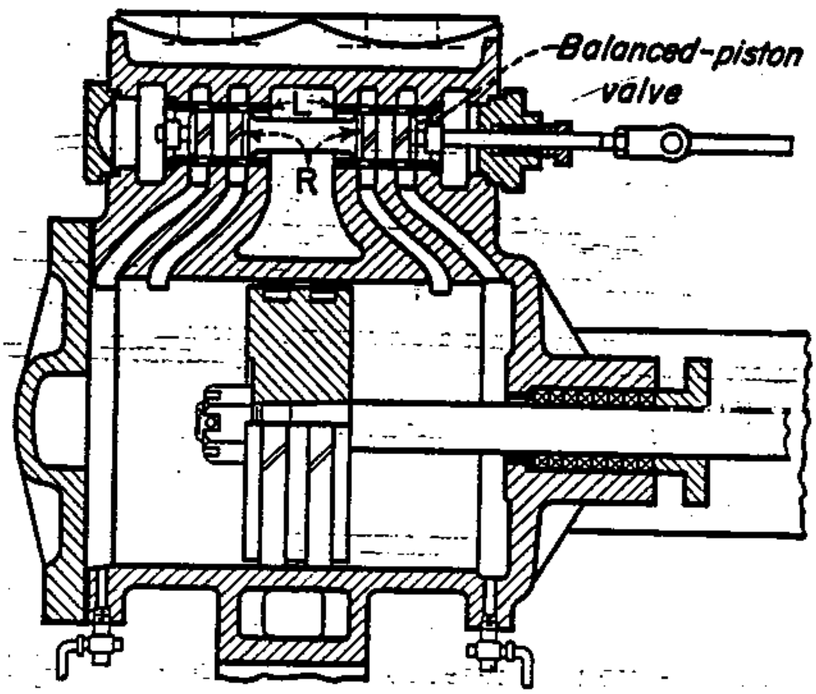


FIG. 13-3. Duplex steam cylinder with integral-piston-valve steam chest.

Question 13-7: Why are piston rings R and steam-chest liners L usually used with piston valves on duplex steam pumps (Fig. 13-3)?

Answer: The lever connection to the piston-rod crosshead moves the steam valves of a duplex pump relatively slowly across the center position of its travel. Because of this slow motion, piston rings are necessary to keep wire-drawing or steam-cutting of the valve at a minimum, particularly as it crosses center position. Whenever piston rings are used on a piston steam valve, liners are required to provide the bridges on which the rings ride in crossing the ports to the main cylinder.

Question 13-8: Do the steam valves of a direct-acting pump have steam and exhaust lap similar to the lap used in steam-engine valves?

Answer: In a direct-acting pump, steam piston pressure is transmitted directly to the liquid piston. Since there is no flywheel or its equivalent, constant steam admission throughout the stroke is necessary to prevent stalling, particularly at low speed. The steam valve does not control cutoff or compression, and therefore does not require lap or lead.

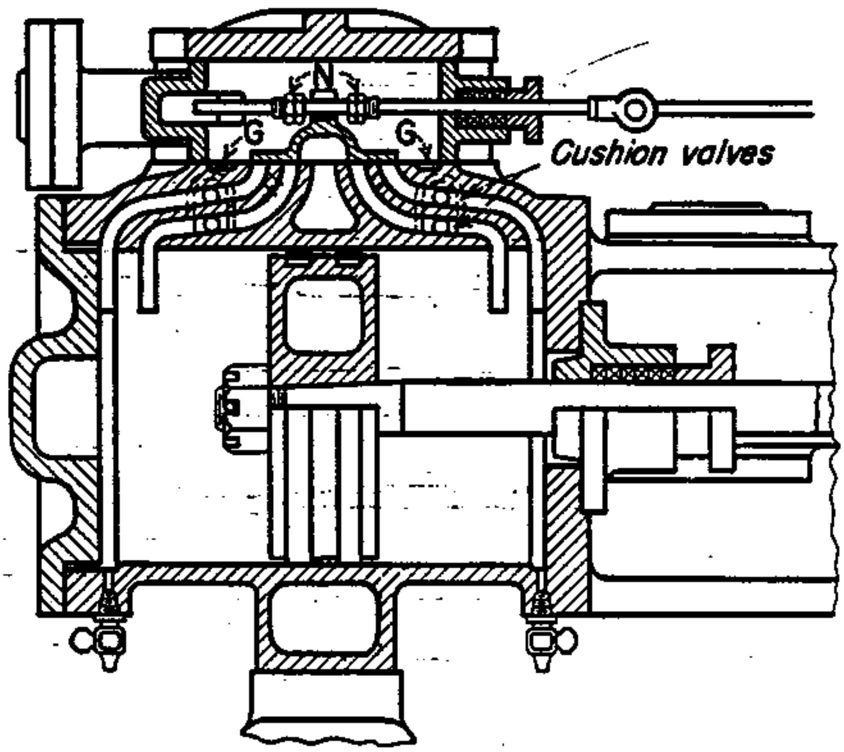


FIG. 13-4. Large duplex steam cylinder with cushion valves and inside adjustable lost motion.

Question 13-9: What is lost motion?

Answer: Lost motion is the play provided in the steam-valve operating mechanism of all direct-acting steam pumps. On small duplex pumps, it is customary to use a single square valve-rod nut *N* (Fig. 13-1), of a width somewhat less than the space between the two steam-valve abutments. This lost motion is not adjustable except by changing the thickness of the nut. On larger duplex pumps, pairs of nuts *N* locked together may be used on both sides of a single abutment on the valve (Fig. 13-4), or an equivalent arrangement provided outside the valve chest. This construction provides for adjusting the lost motion and varying the valve travel in either direction to obtain best operating results.

Question 13-10: How much lost motion is provided in a duplex steam pump?

Answer: It equals one-quarter to three-eighths of the valve-rod travel. With the valve-rod linkage designed to give full port opening at rated pump stroke, the total lost motion is less than the steam-port width.

Question 13-11: What controls the stroke length of a duplex pump, and how is the piston stopped without undue shock?

Answer: Exhaust ports *F* (Fig. 13-1) in the steam cylinders of a duplex pump terminate in the cylinder bore some distance from the end of piston travel. Separate steam ports *S*, nearly always of the same size as the exhaust port, run to the counterbores

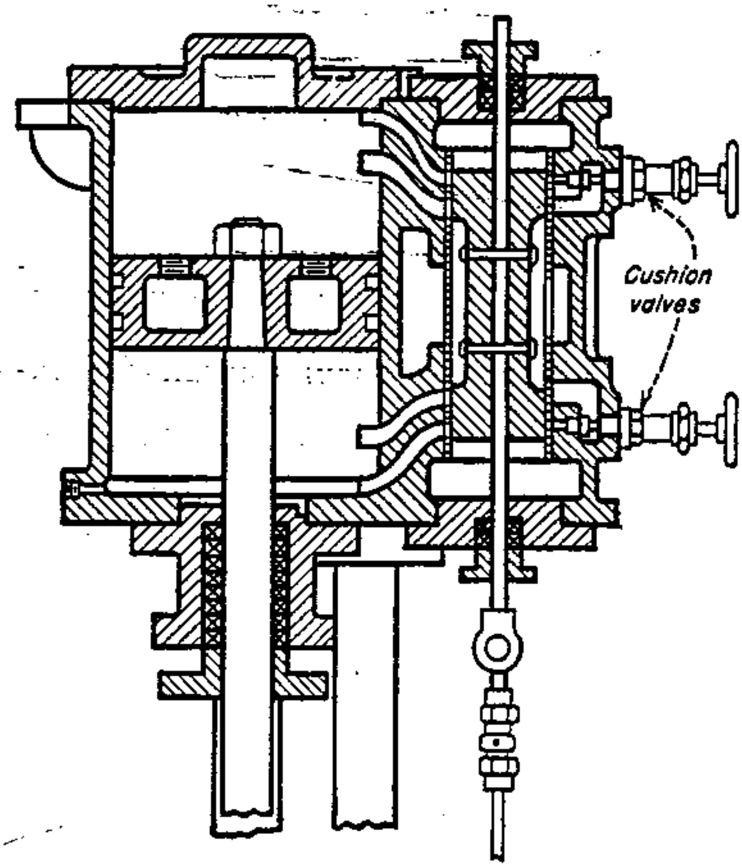


FIG. 13-5. Vertical duplex steam cylinder with cushion valves and outside adjustable lost motion.

at the cylinder ends. As it approaches the end of its stroke, the steam piston cuts off the exhaust port, trapping steam, which is compressed by the motion of the piston. This cushion brings the piston to a stop with a minimum of shock in the liquid end, even though steam is not cut off from the other side of the piston until the piston in the opposite side of the pump has passed midstroke.

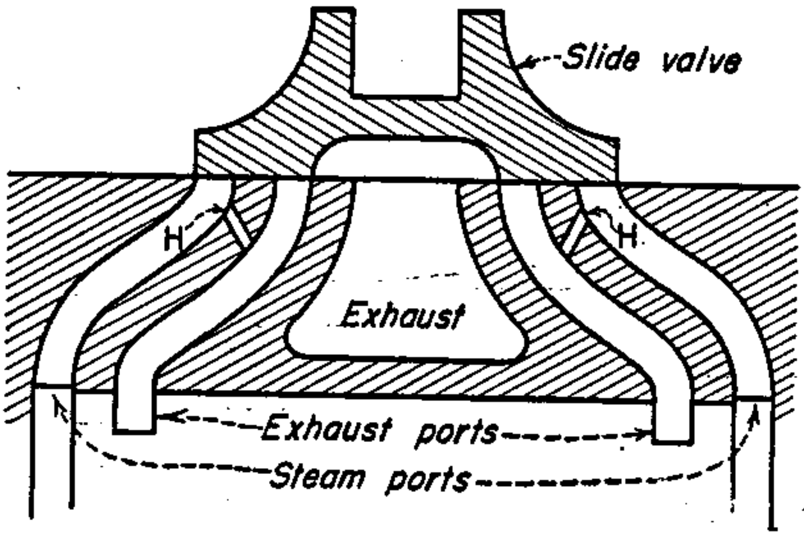


FIG. 13-6. Small duplex steam cylinder with drilled holes (H) for cushion release.

Question 13-12: What are cushion valves?

Answer: On large steam pumps, valves are provided in a passage leading from the steam to the exhaust port (Figs. 13-4 and 13-5) to control the effect of steam trapped between the piston and cylinder head. Opening these valves permits the trapped cushion steam to leak through to the exhaust port, to reduce the effect of the cushion. Closing them stops the leakoff and causes maximum cushioning effect. On small pumps for service where relatively little steam cushion is needed, small holes, H, (Fig. 13-6) are sometimes drilled through the bridge between the steam and exhaust ports. These provide a fixed leakoff like that obtained through an open cushion valve.

Question 13-13: What is a simplex direct-acting steam pump?

Answer: It is a pump with a single liquid cylinder and piston, in line with and operated by a single steam piston and cylinder

(Fig. 13-7). The main piston is controlled by a steam-actuated main valve (Fig. 13-8), which is controlled by a pilot valve. This valve is operated from a crosshead on the main piston rod, through

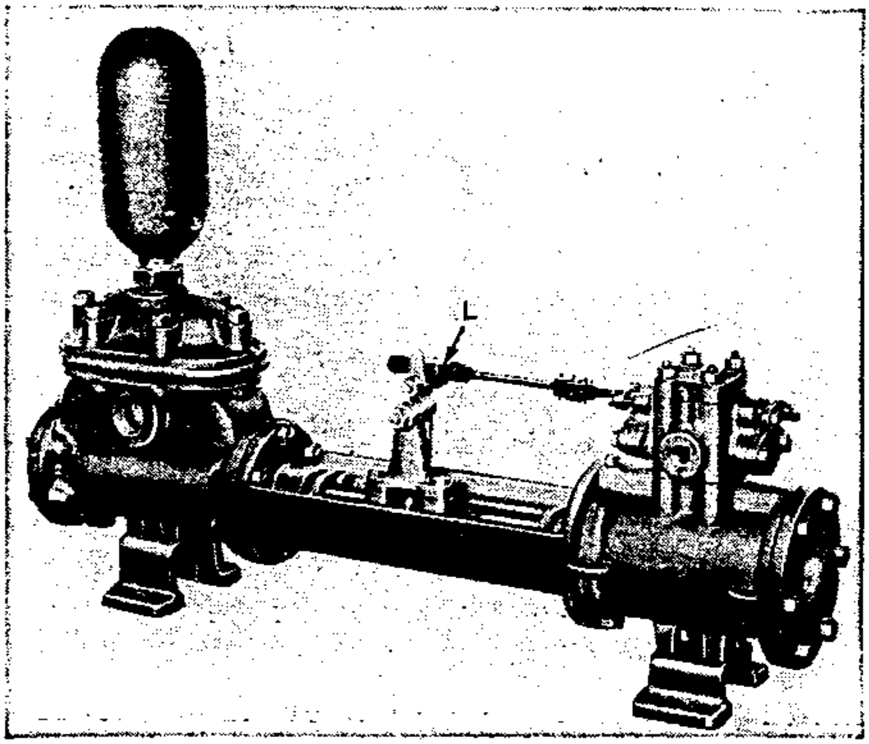


FIG. 13-7. A simplex piston-pattern direct-acting pump of the cap and valve-plate type.

a system of levers and valve rods. Many arrangements of main valve, pilot valve, and operating levers have been used since the original direct-acting simplex steam pump was invented.

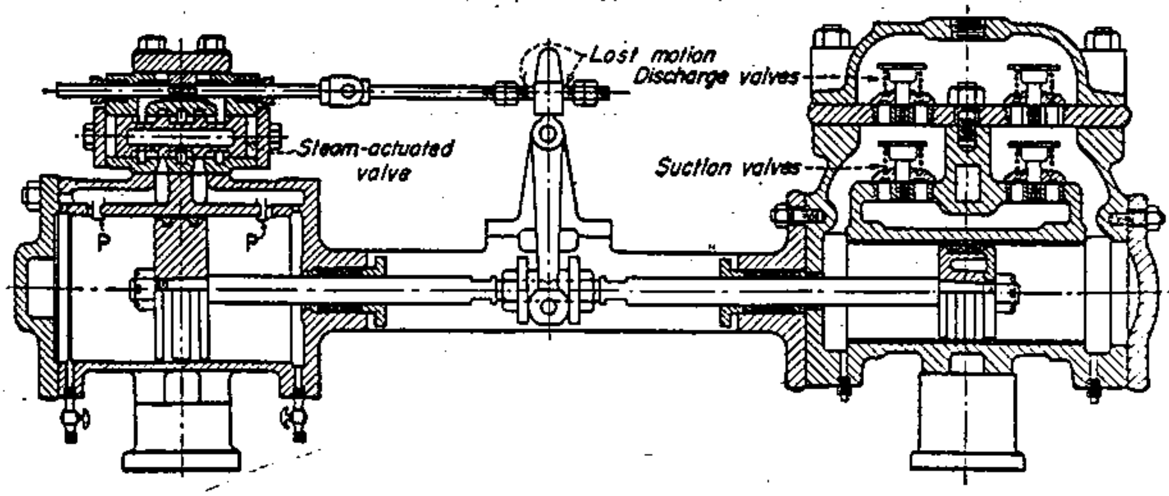


FIG. 13-8. Cross section through simplex direct-acting pump showing "D"-type pilot valve seating on steam-chest cover and piston main valve.

Question 13-14: How much lost motion is provided in a simplex steam pump?

Answer: On a simplex steam pump, the pilot valve is much smaller than the steam valve of a duplex pump, hence requires less movement to obtain full port opening. In spite of a somewhat lower actuating-lever ratio than that used on a duplex pump, practically all its movement is absorbed in lost motion. The valve-gear lost motion of a simplex pump is set with the pump in operation to get full stroke and a smooth reversal rather than as a definite proportion of the actuating-tappet travel. This makes it necessary to use an external lost motion as, at *L* (Fig. 13-7), on all sizes of pumps, instead of only on the larger sizes as in duplex pumps.

Question 13-15: How is a simplex steam cylinder arranged for cushioning main-piston travel?

Answer: Most simplex steam cylinders are made with a main port *P* (Fig. 13-8) serving as both a steam and an exhaust port, some distance from the cylinder end. A much smaller port extends from the main port, or from the steam chest, to the counterbores at the cylinder ends. After the steam piston cuts off the main port, the trapped steam forming the cushion leaks off through this small port as the piston advances to the point where the steam-valve reversal takes place. It is not necessary for the cushion to stop the main piston completely, as it does in a duplex pump. After the valve reverses, the small port leading to the counterbore serves as a starting port to move the piston to the point where the main port is uncovered.

Question 13-16: Why are plug piston valves frequently used in simplex steam-pump valve gears?

Answer: In a simplex pump the steam-actuated piston valve travels quickly from one end of the steam chest to the other, even at minimum pump speed. Under this condition, wire drawing of the cutoff edges of valve and steam chest cannot be serious. Therefore a simple plug valve in a steam chest without a liner gives good results and possibly is subject to less wear than a valve equipped with piston rings.

Question 13-17: Why cannot a simplex-pump steam valve connect directly to the main piston rod, as on a duplex pump?

Answer: If this were done, the pump would stall instead of reversing, particularly at low speed, because as soon as the steam valve reaches midposition, it cuts off both steam and exhaust ports. To avoid stalling requires some means independent of main-piston motion to throw the valve all the way when the control is actuated at the end of each stroke. In the earliest designs of direct-acting steam pumps, the steam valve was actuated by a spring deflected during the stroke against a latch released at the end of the stroke. For many years practically all simplex pumps have had steam-thrown valves, which have had an almost infinite number of different arrangements.

Question 13-18: Why are counterbores (Fig. 13-1) provided at each end of a steam cylinder?

Answer: Counterbores are provided at each end of a steam cylinder so the leading piston ring can override, for part of its width, the end of the cylinder bore, thus preventing wearing a shoulder on the bore. A simplex pump should be adjusted for full rated stroke to accomplish this. It is impossible to prevent short-stroking in a duplex pump, but it should be reduced to a minimum by adjusting the steam cushion and lost motion.

Question 13-19: What is the normal or rated stroke of a direct-acting pump?

Answer: The normal or rated stroke is that length indicated on the nameplate of the pump. A pump should maintain approximately this stroke over a considerable range of speed without the piston contacting the steam cylinder heads at maximum operating speed. At normal stroke, the leading steam-piston ring should slightly override the edge of the counterbore at each end of the cylinder, thus preventing shoulders forming with wear.

Question 13-20: What is the contact stroke of a direct-acting pump?

Answer: It is the maximum stroke. During such operation the steam piston touches or strikes the cylinder head at the end of each stroke.

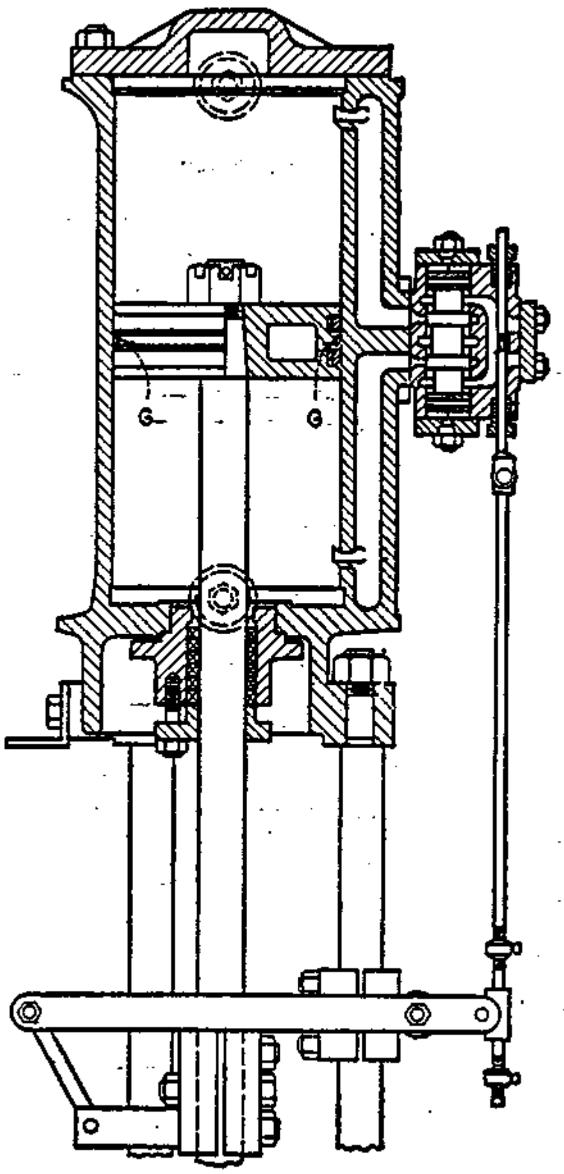


FIG. 13-9. Vertical simplex steam cylinder with grooved piston.

Question 13-21: How many steam-piston rings are used on a direct-acting pump?

Answer: Since these rings must be relatively wide to meet all operating requirements, usually only two are used.

Question 13-22: Why do steam pistons of many direct-acting pumps have a groove *G* between the two piston rings (Fig. 13-9)?

Answer: With a plain piston in large-sized pumps, a dull knock frequently occurs at each end of the stroke, even though the piston stops some distance from the cylinder head. This knock is particularly noticeable when the piston is at the top cylinder end of a vertical simplex pump. Many operators have attributed this knock to the quick rise in pressure in the steam port following reversal of the valve, in the belief that the piston was driven against the side of the cylinder opposite the steam port.

The same kind of knock occurs in large horizontal duplex pumps in which the steam port leads into the cylinder counterbore and the exhaust ports in the cylinder bore are the only ones covered by the piston body. Under these conditions, the knock can be caused only by suddenly releasing steam pressure from between the two rings on the side of the piston covering the exhaust port. A groove between the piston rings equalizes the pressure around the piston sufficiently to prevent this annoying but not serious condition.

Question 13-23: Why is a groove *G* provided in the valve seat at each end of steam-slide-valve travel, as in Fig. 13-4?

Answer: If a flat valve is not permitted to overtravel the length of the seating surface, wear forms a shoulder. This rounds the cutoff edge of the valve and tends to lift it off its seat at the end of each stroke, causing steam leakage and cutting of the seating surfaces. If a raised valve-seat face is not provided, a groove with its inner edge inside the edge of the valve with minimum travel should be cast or cut in the valve-seat face.

Question 13-24: What materials are used in the steam end of a direct-acting pump?

Answer: For most services, cast iron is an excellent material for cylinders, cylinder heads, steam chests, valves, and steam pistons and rings. It is readily cast in the complicated shapes required and, largely because of its free graphite content, possesses good wearing qualities, particularly where lubrication is not permissible. Steel piston and valve rods are usually used with bronze-bushed stuffing boxes and glands. Piston rods are frequently made of monel.

For higher pressures and superheated steam and for naval combat and auxiliary vessels, cast-steel cylinders with iron linings are used. In marine service, steam-cylinder lubrication is seldom permissible, and a combination two-piece piston ring of iron and bronze is used to obtain a reasonable service life under the most severe conditions.

Question 13-25: What is the difference between a piston and a plunger pump?

Answer: In a piston pump, packing is on the piston inside the fluid cylinder (Fig. 13-10). Because of packing location the operator cannot see the leakage past it or make adjustments that might mean the difference between good operation and packing failure. This low-cost construction is satisfactory for low and

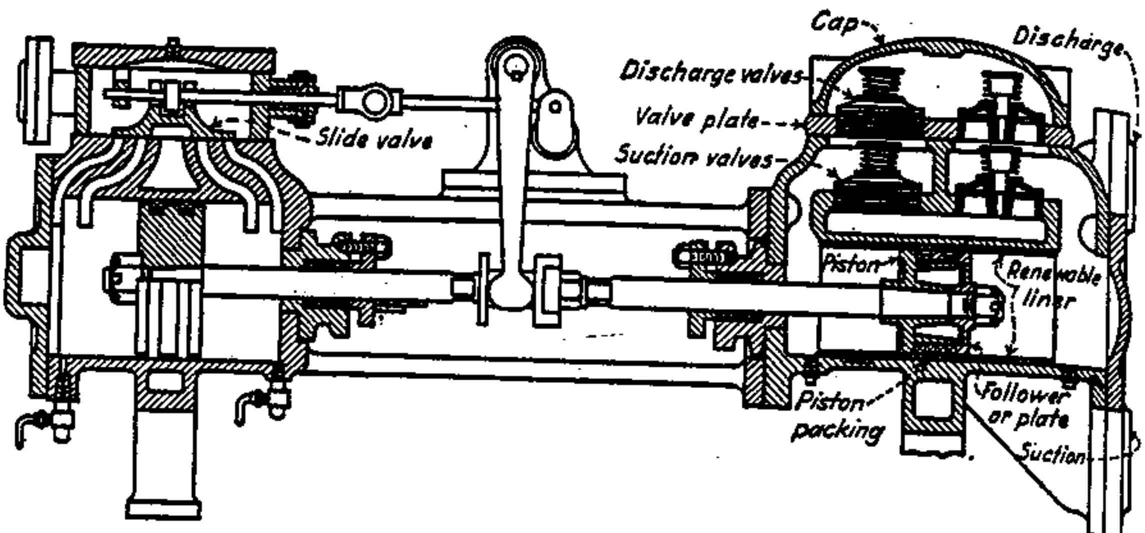


FIG. 13-10. Cross section through duplex pump showing fibrous liquid piston packing in lower and iron snap-ring and bull-ring packing in upper half of illustration.

moderate pressures, but for higher pressures and heavy-duty service the packed-plunger pump is usually favored.

Plunger pumps (Figs. 13-11 and 13-12) have stuffing box, packing, and gland of the same type as those on the piston rods of

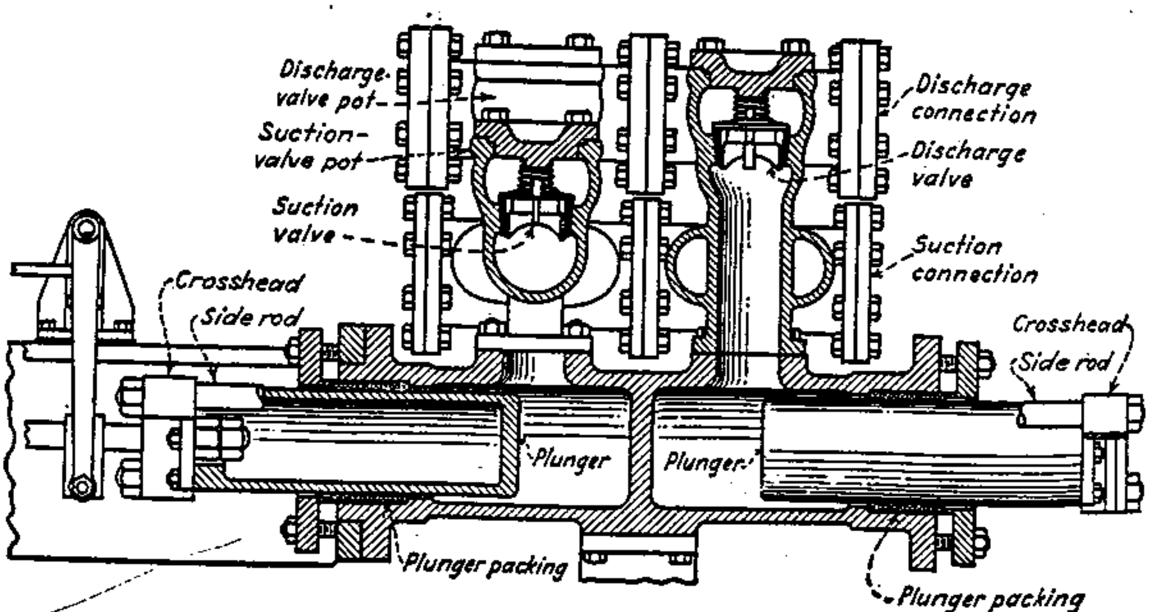


FIG. 13-11. An outside-packed plunger pump of the valve-pot type.

piston pumps. All packing leakage is external, where it is a guide in adjustments for minimum leakage compatible with reasonable friction losses and long life of packing and plunger. During operation, additional lubrication can be supplied to external packing, either through a connection to a stuffing-box lantern ring or by direct application to plunger.

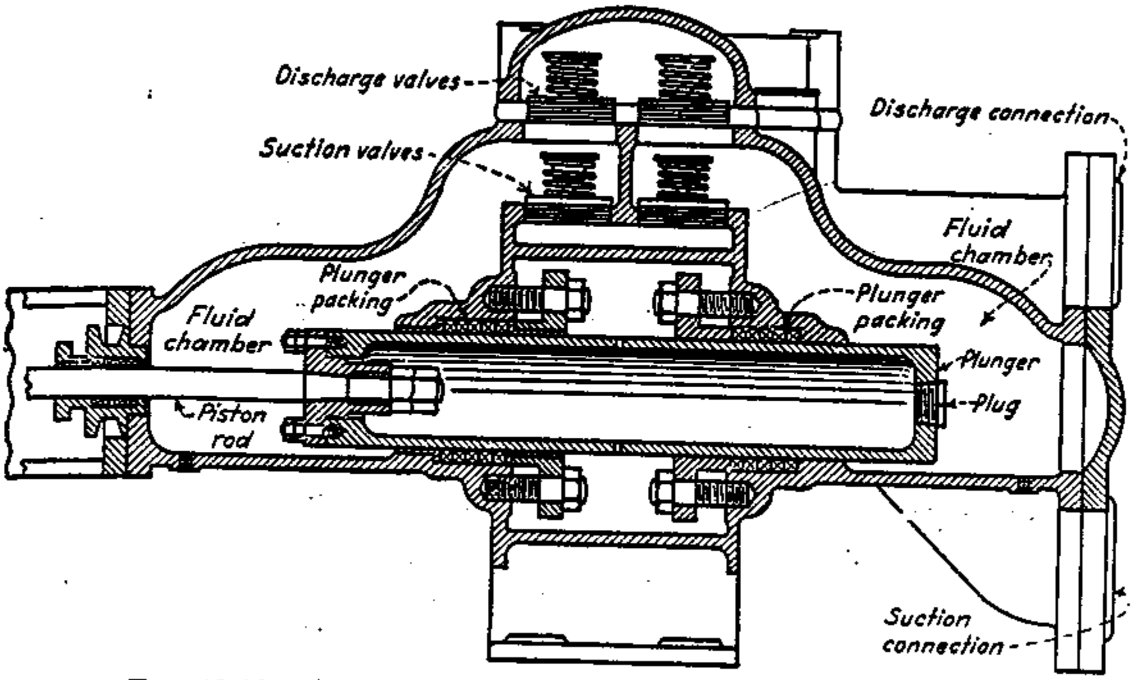


FIG. 13-12. A center-packed cap and valve-plate plunger pump.

Question 13-26: What is the difference between an outside-packed and an inside-packed plunger pump?

Answer: An outside-packed plunger pump (Fig. 13-11) has a separate plunger for each cylinder, with each pair of plungers connected by crossheads and side rods external to the pump cylinder. An inside- or center-packed plunger pump (Fig. 13-12) has a single elongated plunger for each pair of fluid chambers, with the plunger connected to the driving cylinder by a piston rod like that in a piston pump. Stuffing boxes of an outside-packed pump (Fig. 13-11) are more accessible. In larger sizes where plunger weight becomes a problem, external crosshead guides may be used to relieve packing and stuffing-box bushings of this weight. A center-packed plunger (Fig. 13-12) has advantages in a vertical installation where plunger weight does not affect the stuffing boxes and where a stuffing box at the bottom of the cylinder end may be inaccessible.

Question 13-27: What is a cap and valve-plate direct-acting pump?

Answer: This valve arrangement on the liquid cylinder (Fig. 13-10) may be used on either a direct-acting steam pump or a

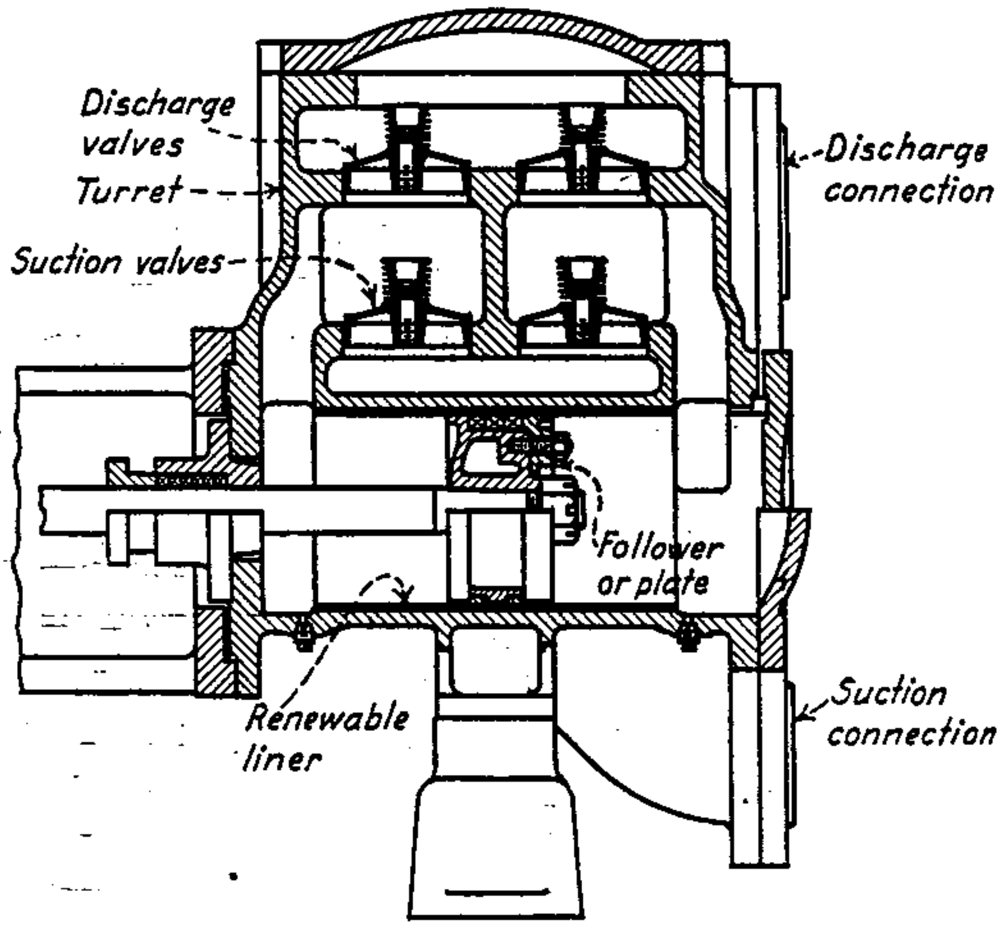


FIG. 13-13. Turret-type liquid cylinder.

power pump. It is the least expensive and most used cylinder and valve arrangement for reciprocating units. The discharge-valve seats, together with valves, valve stems, and springs, are mounted in a separate plate, easily removed for servicing or to obtain access to suction valves. In this plate, an opening communicates with a cored passage in the cylinder casting leading to the discharge connection. A dome-shaped cap, subject to discharge pressure, covers the discharge-valve plate.

Suction-valve seats are in the cylinder casting directly below their respective discharge valves. A cored passage leads from below the suction-valve seats down between the cylinders to the suction connection.

Cap and valve-plate pumps are usually built for moderate pressures and temperatures, although sometimes they are rated to 350 psi discharge pressure and to 350°F.

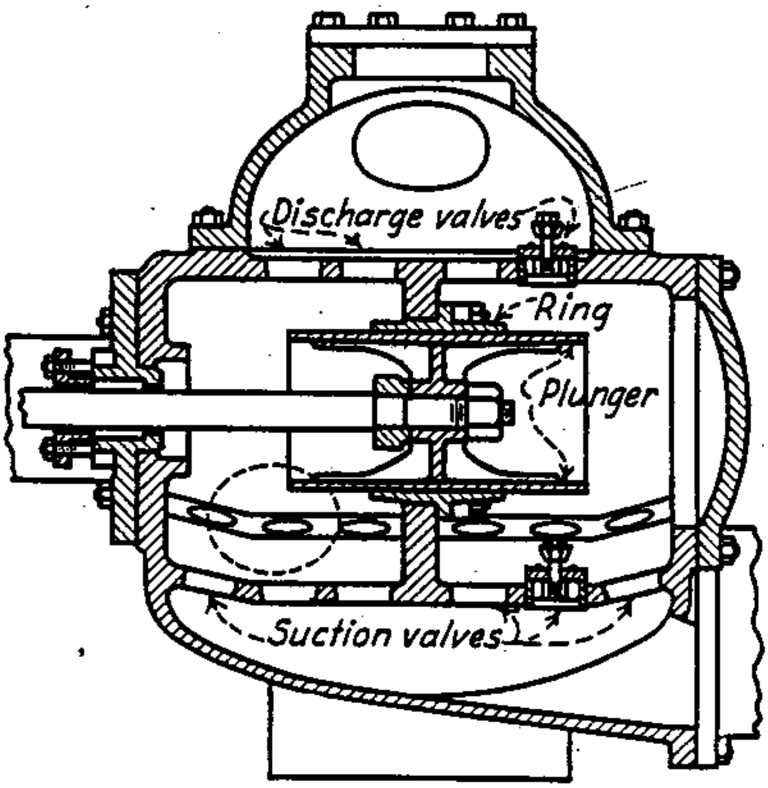


FIG. 13-14. Plunger and ring packing in straight-through liquid cylinder.

Question 13-28: What is a turret-type piston pump?

Answer: Figure 13-13 shows the liquid cylinder of such a pump. It has both suction and discharge valves in the cylinder casting. Access to suction valves is through removable covers on the sides of the turret, and to discharge valves through the top cover. This cylinder design, for low to medium pressures and relatively large capacities, is adaptable to the use of two or more valves of moderate size in each corner of the pump, instead of a single large one.

Question 13-29: What is a plunger-and-ring liquid cylinder?

Answer: The fire pump (Fig. 13-14) employs plunger-and-ring construction. Instead of a packed piston, this unit has a carefully finished plunger sliding in a closely fitted bronze ring. To reduce weight to a minimum the plunger is hollow, except for a center dividing wall that carries a boss for connection to the piston rod. The cylinder is a straight-through pattern, with suction valves below and discharge valves above the plunger. A large number of moderate-sized valves permit operation at about twice the piston speed allowable for continuous-duty pumps.

Plunger-and-ring construction eliminates the possibility of the piston packing swelling to inoperative tightness or disintegrating during stand-by periods. Valves of rubber composition have metal backing plates.

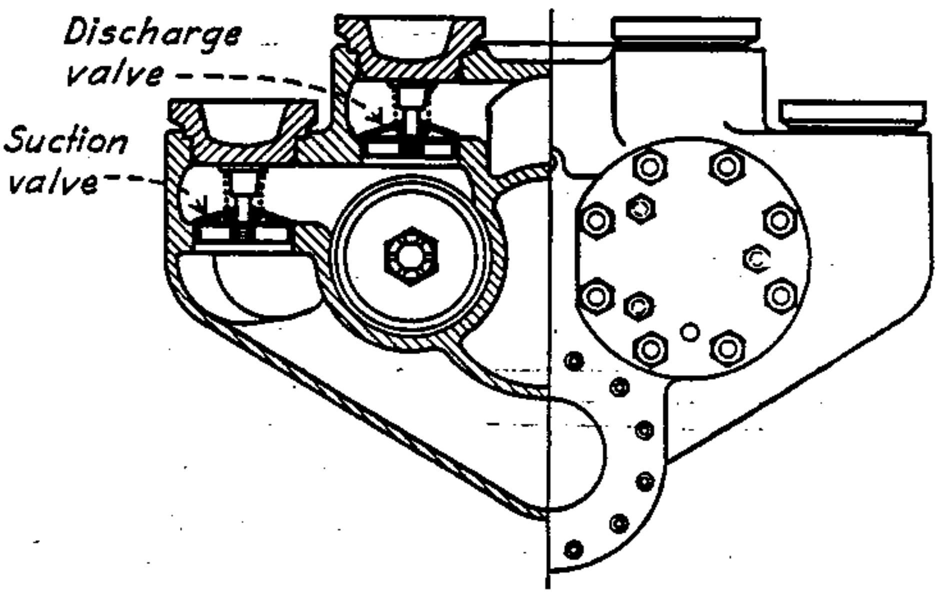


FIG. 13-15. Side pot type of duplex liquid cylinder.

Question 13-30: What is a sidepot-valve liquid cylinder?

Answer: Figure 13-15 shows such a cylinder. Suction valves are placed in individual pots on the sides of the cylinders and discharge valves in the pots above the cylinders. The relatively small individual valve-pot covers and the valve and cylinder arrangement permit design of maximum strength without excessive weight.

Sidepot-valve cylinders in a capacity range from moderate to large are widely used in refinery and oil field installations. This design is commonly employed for the maximum pressure practicable for a piston-pattern pump.

Question 13-31: What is a pressure pump?

Answer: It is primarily a packed-plunger rather than a piston type and is used frequently for hydraulic press service. For maximum pressure, the fluid end of such pumps is made from a single steel forging, with plunger chamber, valve holes, and suction and discharge passages bored in the solid forging. Valve-hole plugs on these pumps often have a metal-to-metal rather than a gasketed joint. For somewhat lower pressures and higher capacities, a pot-valve pump often has cylinders of heavy iron or steel castings. Figure 13-11 shows such a pump, with valves in individual pots bolted to plunger chamber.

Question 13-32: What is a tank pump?

Answer: It is a pump that discharges to a tank. Because of the relatively low discharge pressure required to pump to the elevated tank of an industrial water-supply system, the liquid piston of the pump is usually as large or larger than the steam piston. A tank pump requires a steam end with maximum cushioning because the water in suction and discharge lines tends to keep flowing, thus unloading liquid pistons as they come to rest at the end of the stroke.

Question 13-33: What is a differential piston pump?

Answer: This design obtains a double-acting pumping effect and loading of the power end with only half the usual number of valves. The piston rod of such a pump is, in effect, a plunger, in that it has one-half the area of the liquid piston (Fig. 13-16). On the forward stroke of the piston, the suction valve is closed and the entire displacement of the liquid piston passes through the discharge valve. Actual delivery is only half this amount because of flow from above the discharge valve through a passage to the plunger side of the piston. On the return stroke, this fluid is delivered past the top of the closed discharge valve, while the cylinder is taking a full charge through the suction valve. Thus the cylinder has a double-acting discharge but only a single-acting suction characteristic.

Question 13-34: What is a close-clearance pump?

Answer: A close-clearance fluid end nearly always applies to a simplex pump that handles highly volatile liquids in oil refineries. Fluid valves and passages are arranged for minimum clearance volume, and an extended liquid piston overruns the liner at each end of stroke, filling that portion of space between the valves (Fig. 13-17). Under most conditions, this pump can compress and discharge vapors that accumulate in the cylinder. Hand-operated

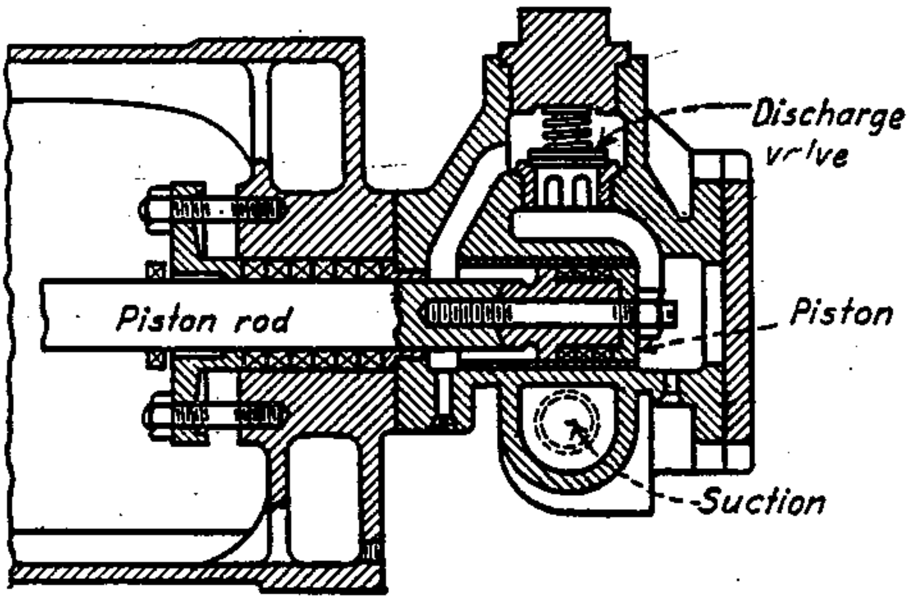


FIG. 13-16. Section through differential piston pump showing the single discharge valve. A single suction valve is located on the side of the cylinder between the suction manifold and the space below the discharge valve.

priming valves by-pass the discharge valves if the cylinder becomes vaporbound. Such a liquid-cylinder arrangement has little value on a duplex pump because of its tendency to short-stroke, particularly when not completely primed.

Question 13-35: What is a renewable or driven liner in a liquid cylinder?

Answer: It derives its entire support from the drive in the cylinder bore (Figs. 13-10 and 13-13). As a rule, such a liner is relatively thin and is commonly made from a centrifugal casting or a cold-drawn brass tube. After a driven liner is worn to where it must be replaced, it is usually removed by chipping a narrow groove along its entire length. This groove is cut as nearly as possible through the liner without damage to the wall of the cylinder bore. After chipping such a groove in the liner, it can be easily collapsed inward and removed.

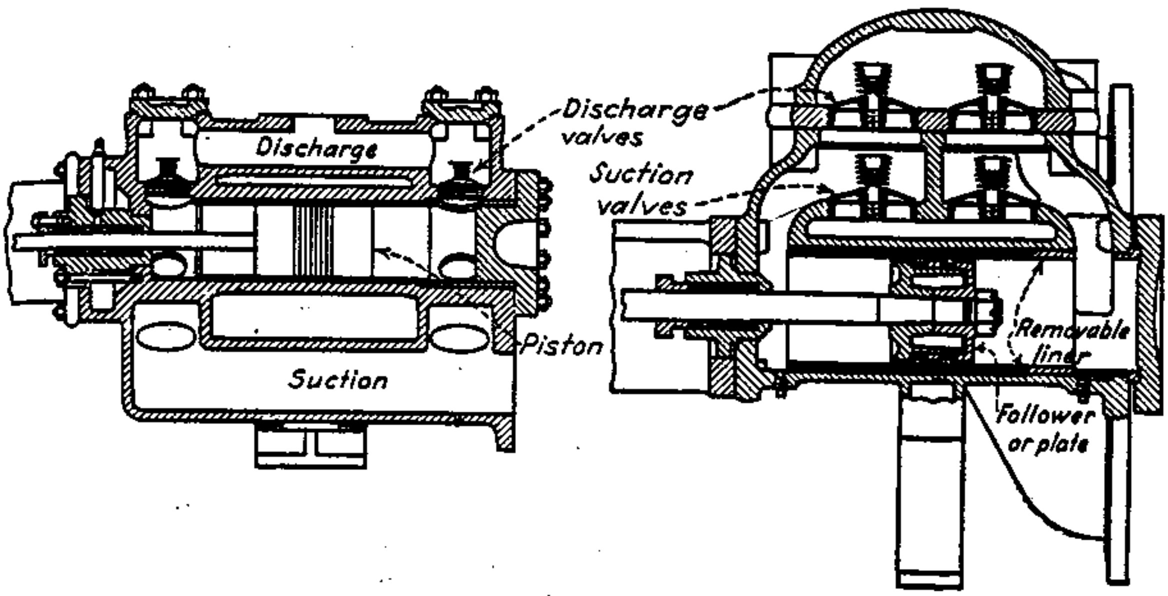


FIG. 13-17. FIG. 13-18.

FIG. 13-17. Close-clearance simplex liquid cylinder for refinery applications.
FIG. 13-18. Liquid cylinder with removable liner.

Question 13-36: What is a removable liner?

Answer: It may be removed and replaced without damage. Instead of being a driven fit throughout most of its length, it is located longitudinally by a flange at the cylinder end and is a driven fit only at the end of the cylinder bore (Fig. 13-18). For moderate pressures, the liner contacts cylinder heads and is sealed by a tight fit at the beginning of cylinder bore. For higher pressures, a flange fits into a recess at the beginning of cylinder bore; this flange is held in contact with a shoulder by jack bolts or a spacer between the cylinder head and end of liner. Sometimes a packing ring is used between the flange and shoulder for a more positive seal between the two cylinder ends. Removable liners are heavier than renewable ones.

Question 13-37: What is a wet liner?

Answer: This special form of removable liner simplifies the cylinder casting in vertical pumps by eliminating the need for a separate passage from the cylinder bottom up to the valve deck. It fits tightly only at the top end, just below the flange. Considerable space between the outside of the balance of the liner and the cylinder barrel (Fig. 13-19) forms a passage from the bottom of the cylinder to the outlet in the valve deck. A wet liner must have strength to withstand external loading of full discharge pressure throughout most of its length when piston approaches bottom of its stroke.

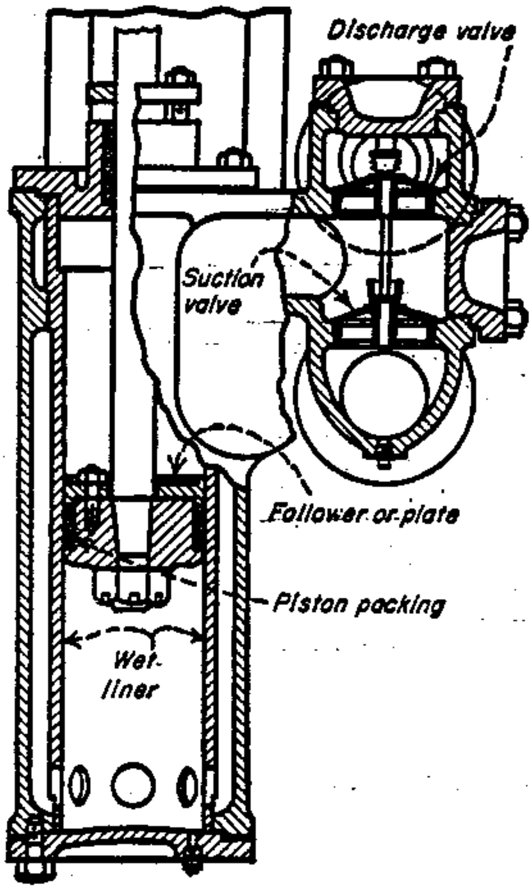


FIG. 13-19. Vertical liquid cylinder with wet liner.

Question 13-38: What is a follower liquid piston?

Answer: At one end it has a plate, or follower, removable for replacement of packing without disturbing the piston body (Figs. 13-10, 13-13, and 13-18). In smaller sizes, a single piston-rod nut holds both follower and piston in place on the rod (Figs. 13-10 and 13-18). In larger sizes, such a nut bears directly on the piston body; separate studs and nuts (Fig. 13-13) hold the follower plate in place. On vertical pumps, the follower plate is often on the top side of the piston, to permit repacking from above without disturbing anything but the top of the cylinder head and follower plate.

Question 13-39: What is a solid liquid piston?

Answer: A groove or grooves cut in the solid body receive the piston packing. To renew it, the piston must be withdrawn from the cylinder. Solid pistons are often used on heavy-duty pumps where the size of the piston rod and the depth of the packing groove make it impracticable to use a studded follower plate or to remove a large and extremely tight piston-rod nut to repack the piston.

Question 13-40: What is a wet-vacuum pump?

Answer: It is a special form of submerged piston pump (Fig. 13-20), generally used with reciprocating condensing steam engines to remove both air and condensate from condenser hotwell. The cylinder is lightweight because of low discharge pressure. It has

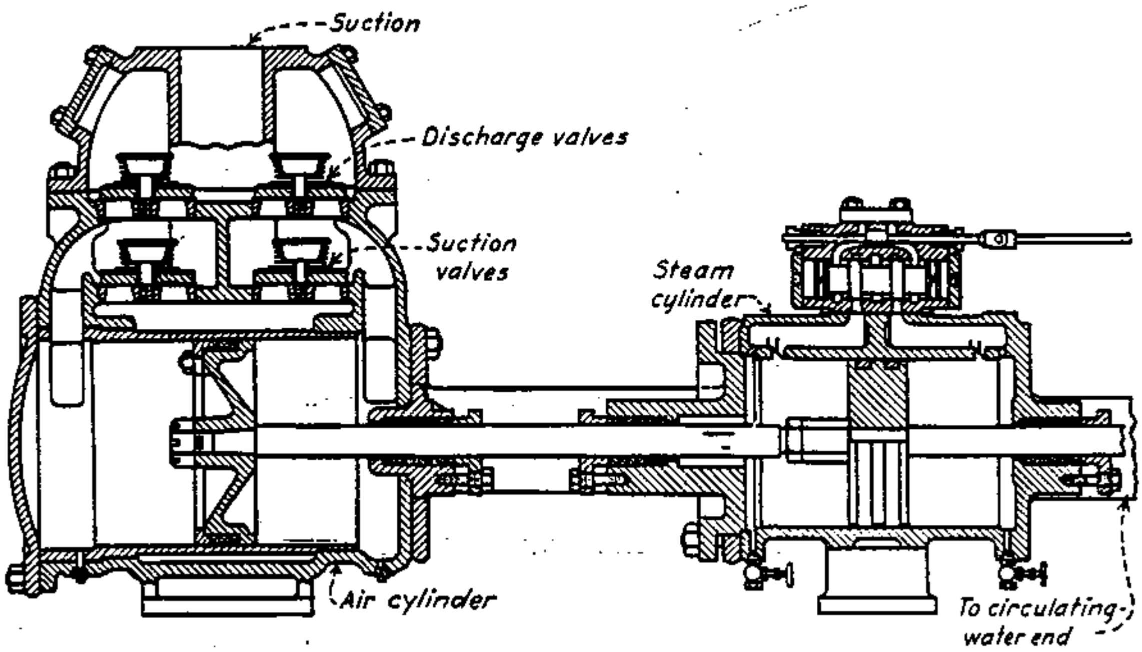


FIG. 13-20. Horizontal simplex wet-vacuum pump, usually built as a combined vacuum and condenser circulating pump driven by a single steam cylinder.

large passages and considerable space between suction and discharge valves, so that air may collect under the discharge valves with a minimum of mixing with the water that fills the clearance volume. The suction connection is usually on the top of the valve deck to permit a straight drop from the condenser hotwell to the pump suction chamber. Suction valves are sometimes set at an angle to permit free air flow from the higher side of suction valve to the bottom of the discharge valve.

Question 13-41: What materials are used for the liquid end of a direct-acting pump?

Answer: Most direct-acting pumps have cast-iron cylinders, heads, and valve plates and covers, and bronze piston rods and valves. Liquid pistons may be either cast iron or bronze. Rubber valves are often used for low pressure service and for vacuum pumps. Stainless steel is used for valves and seats; piston rods are often of stainless steel or monel metal. All bronze fluid ends are used for acid mine water and for salt water, where electrolytic action causes rapid deterioration when dissimilar metals are present. Fluid ends are all iron for certain alkalies and for oil.

CHAPTER 14

HORIZONTAL AND VERTICAL POWER PUMPS

Question 14-1: What is a power pump?

Answer: It is a piston or plunger reciprocating pump driven by a rotating crank or eccentric. This design is termed "power pump" because it is driven by an outside power source, such as an electric motor or internal combustion engine, instead of a steam cylinder, as in a direct-acting steam pump.

Question 14-2: What are its essential parts?

Answer: It has two major assemblies:

1. The fluid end, as in a direct-acting steam pump, includes liquid cylinders with pistons and rods, suction and discharge valves, and piping connections.
2. The power end consists of a frame to support or enclose moving parts and make connections to liquid cylinders, crossheads, connecting rods, a crankshaft or camshaft with bearings, and often built-in reduction gears. The driving motor or engine is not usually considered a power-pump part. The foundation of large open-frame horizontal power pumps might well be considered part of the unit, because it contributes to the lateral strength or resistance to weaving of the assembly.

Question 14-3: How are power pumps classified?

Answer: Classifications are:

1. Piston or plunger arrangement, such as horizontal, vertical, inverted vertical, radial, axial.
2. Open or enclosed frame.
3. Single- or double-acting fluid end.

4. Piston or plunger fluid end.
5. Number of crank- or eccentric-shaft throws.
6. Fixed or variable stroke.

Question 14-4: What is the general arrangement of a horizontal power pump?

Answer: Figure 14-1 shows a common form of horizontal duplex unit, complete with top-mounted motor and enclosed chain drive

FIG. 14-1. Horizontal duplex power pump with top-mounted motor and enclosed chain drive to the pinion shaft.

to the pinion shaft. This pump has a cap and valve-plate piston fluid end bolted directly to the extended power-end frame. The cradle portion of the frame encloses piston-rod stuffing boxes, which are reached for adjustment or repacking by removing sheet-steel covers *A*. Circular covers on the side of the power end house the main- and pinion-shaft roller bearings. Main cover *B* is removable for access to gears and connecting rods.

Figure 14-2 shows a longitudinal section through a pump like Fig. 14-1. Solid cylindrical crossheads run in guides bored in the power-end frame. The crosshead end of the connecting rod has a bronze bushing running on a steel crosshead pin. The crank end of the connecting rod carries a shim-adjusted split bronze bearing lined with babbitt. The steel crankshaft has an integral flange between crank throws, to which the gear is bolted. It is used with a double-acting duplex liquid end; hence its cranks are spaced 90 deg.

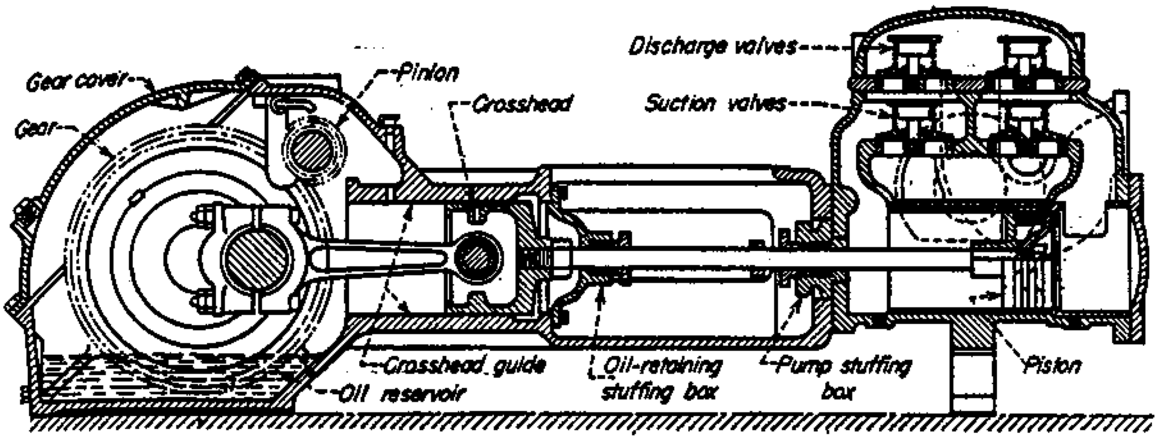


FIG. 14-2. Cross section through horizontal duplex cap and valve-plate power pump.

Question 14-5: What variations from the general setup in Question 14-4 are found in small units?

Answer: Smaller sizes seldom have integral reduction gears, because they can operate at speeds suited to a single reduction such as an external chain or belt drive. The smallest pumps often have a single overhung crank on one end of a shaft supported in sleeve or roller bearings, and a pulley or sprocket on the other end. Such pumps nearly always have a single-piston double-acting fluid cylinder. Somewhat larger pumps often have eccentrics on a straight shaft, the same as the duplex pump (Fig. 14-3). Since this unit has a single-acting plunger fluid end, eccentrics are set 180 deg apart. Used with a double-acting duplex liquid end, eccentrics are spaced 90 deg. Because of the large bearing area inherent in the eccentric, a solid or one-piece eccentric strap is practical instead of an adjustable two-part bearing.

Question 14-6: What variations from the setup in Question 14-4 are found in medium-sized units?

Answer: Such units are usually like Figs. 14-1 and 14-2, except that a sidepot-valve fluid end is used oftener than the cap and valve-plate fluid end. Some designers favor an eccentric shaft rather than a crank, to reduce power-end width and sometimes to permit using roller bearings for eccentrics as well as main bearings. The crosshead pin often has needle-roller bearings instead of bronze bushings. In larger sized enclosed-crankcase pumps, the piston fluid ends have divided instead of one-piece piston rods. For some applications, horizontal enclosed-crankcase pumps are triplex instead of duplex; one maker specializes in triplex and in quintuplex, or five-throw, pumps.

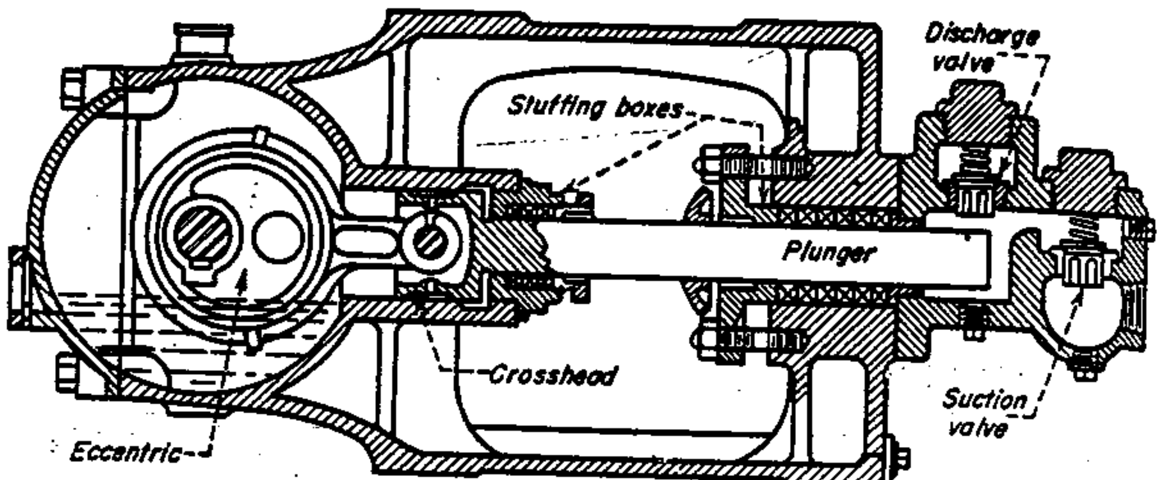


FIG. 14-3. Small-size power pump suitable for driving through a single external reduction from the motor.

Question 14-7: What variations from the setup in Question 14-4 are found in large units?

Answer: Enclosed-crankcase pumps (Fig. 14-4) are used up to about 300 hp. While the power end is generally like Figs. 14-1 and 14-2, the power-end frame connects to the fluid cylinders by a separate cradle. This cradle carries a second crosshead guide,

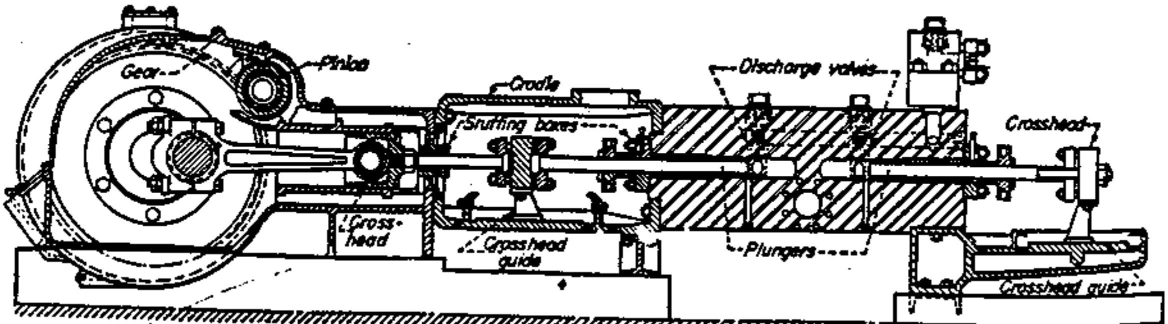


FIG. 14-4. Cross section through a large enclosed-crankcase power pump with forged-steel liquid cylinders.

which supports the weight of plunger, crosshead, and side-rod assembly independently of the power-end crosshead guide. A similar guide at the outboard end of pump carries the other end of this assembly. This unit generally does high pressure work, such as required by forging presses, and nearly always has a forged-steel fluid end. It has separate cylinder forgings, connected on the discharge by a forged-steel header. The high pressure gland has a wiper gland and drain, and the plunger crosshead guides have a separate pocket to catch stuffing-box drips and prevent fouling of the crosshead-guide oil.

Question 14-8: What is a rolling-mill or open-frame horizontal power pump?

Answer: The largest sizes are built up to 1200 hp in a single unit for heavy hydraulic press service, oil pipe lines, and refinery charge pumps. For each crankshaft throw, the crank pit, main bearing support, and crosshead guide are in a separate casting joined by cross pieces and a common foundation. The term "open-frame" does not literally describe these pumps now to the extent it once did, because modern pumps have sheet-steel covers over each crank throw, and reduction gears are housed separately. A rolling-mill frame offers great flexibility in pump setup. While the majority are duplex pumps, there are triplex units and special-service pumps with five- and six-throw crankshafts.

Question 14-9: What type of liquid end is used with horizontal power pumps?

Answer: This depends on the service. General industry and power plants use cap and valve-plate or sidepot liquid ends for low and moderate pressures. Hydraulic presses use forged-cylinder outside-packed plunger pumps at 1000 to 6000 psi or higher. Refineries use both sidepot piston pumps and forged-cylinder plunger pumps, depending on pressure and service. In oil well drilling and oil field gathering service, sidepot piston-pump pressures are much higher than in general industry. Piston-pattern gathering pumps are often used at 1000 psi, occasionally 1200 or even 1500 psi. Mud or slush pumps for oil well drilling, with special rubber-fitted pistons and valves, develop as high as 1500 to 2000 psi. Large pipe line reciprocating pumps have mostly forged cylinder and plunger constructions.

Question 14-10: What types of main bearings are used in horizontal power pumps?

Answer: Enclosed-crankcase pumps usually have tapered roller bearings. Shims under bearing-housing flanges set operating clearance and adjust for wear.

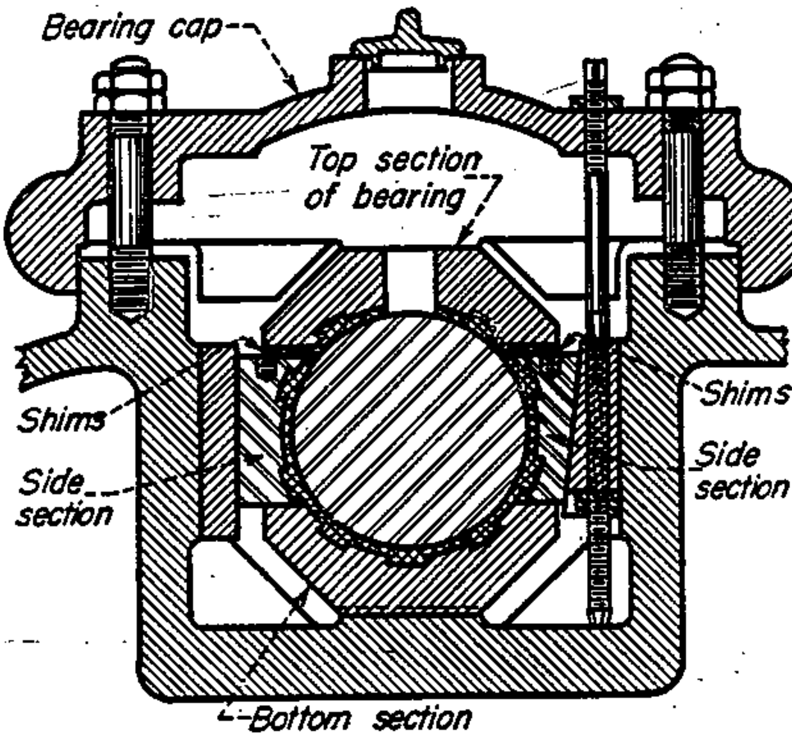


FIG. 14-5.—Four-part babbitted main bearing used on large power pumps.

Most rolling-mill-frame power pumps have been built with four-part babbitted bearings (Fig. 14-5). Shims provide vertical adjustment between the bottom section, sides, and top. Three-part bearings have also had side sections carried farther around the shaft at the top to make up for omitting the top section. Because of the complicated loading imposed on main bearings of horizontal machines, two-part bearings have not been used generally on horizontal pumps. A new type of two-part bearing for horizontal machines (Fig. 14-6) combines the adjustment flexibility of three- or four-part bearings with the assembly rigidity of two-part bearings.

Question 14-11: What type of connecting rods and bearings do horizontal units have?

Answer: Forged- or cast-steel connecting rods with removable bolted caps are usual. The crankpin bearing is a removable babbitt-lined bronze shell, with shims for adjustment. Crosshead-pin bearings are usually one-piece bronze bushings, with larger sizes babbitt lined. Needle-roller bearings running on hardened crosshead pins are sometimes used. On larger open-frame pumps, a

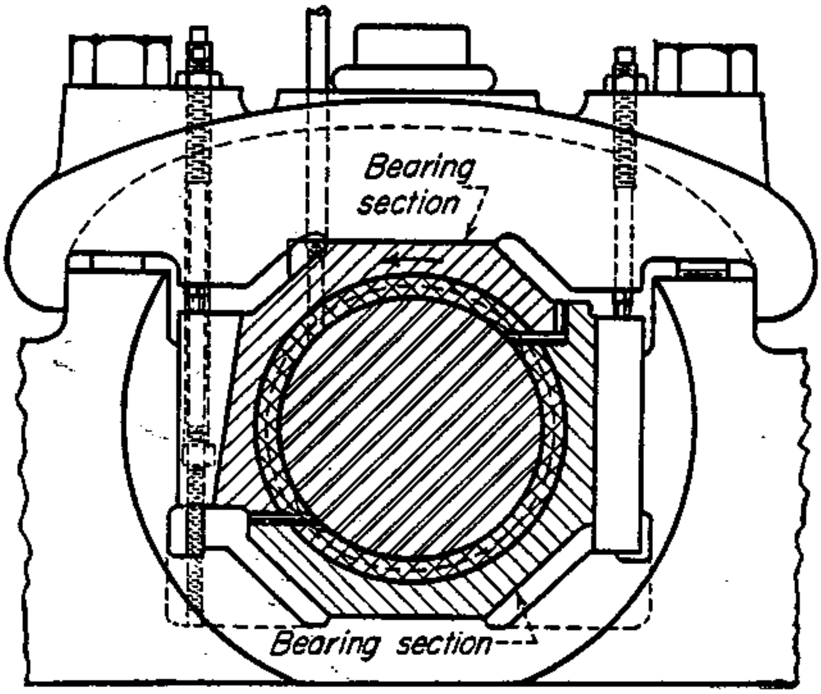


FIG. 14-6. Two-part babbitted main bearing having the adjustment flexibility of a three- or four-part bearing.

rod end like that in Fig. 14-7 adjusts the crosshead-pin bearing. Shims are not used in this bearing, and the wedge and bolt hold its movable half in position in the rod eye. Wedge-adjusted bearings have been used on the crankpin end of large connecting rods, but today most pumps have a marine type or bolted-on bearing (Fig. 14-7) or a movable bronze shell (Fig. 14-2).

Question 14-12: What types of drive do horizontal power pumps have?

Answer: Enclosed-crankcase pumps are made with a pinion shaft heavy enough to use an enclosed chain drive (Fig. 14-1) or a belt secondary reduction. In sizes below about 200 hp, the speed relation between the average driver and the pinion shaft requires secondary or external reduction. Oil fields commonly use multiple-V-belts, because their flexibility is an advantage. On large enclosed-crankcase pumps and nearly all open-frame pumps, pinion shafts couple direct to drivers. Since the pinion shaft is nearly always free to float with the main gear, a flexible coupling should transmit minimum thrust from driver to pinion shaft.

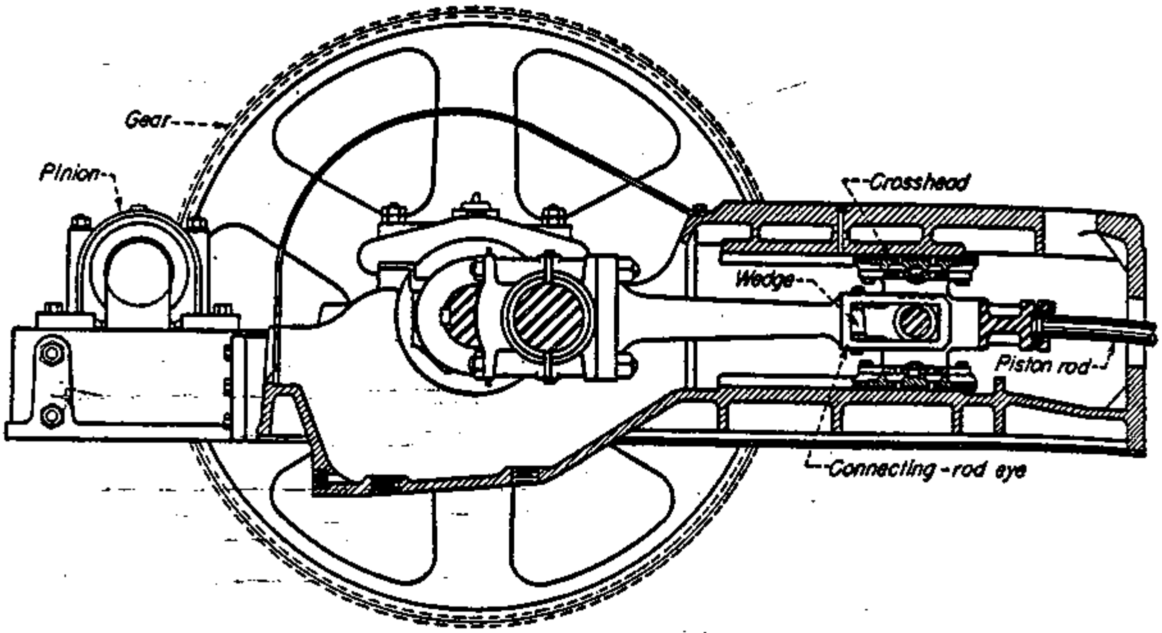


FIG. 14-7. Large open-frame power end with separate adjustable crosshead shoes and adjustable crosshead-pin bearing.

Question 14-13: How is the discharge of power pumps controlled?

Answer: This is a positive-displacement machine, so throttling cannot control its discharge, as in a centrifugal. Common controls are:

1. Start-stop.
2. Variable speed.
3. Automatic by-pass.
4. Synchronized suction-valve unloading.

Question 14-14: When is start-stop control used?

Answer: It is used where pumping may continue until the desired fluid quantity is delivered, as to fill or empty a tank. A float or pressure switch may start or stop the driver by hand or automatically.

Question 14-15: When is variable-speed control used?

Answer: For continuous pumping at a variable rate, as in feeding a boiler. Such a control applies down to two-thirds or one-half maximum delivery. Most variable-speed drives have limitations that demand by-pass control for a lower delivery rate.

Question 14-16: When is by-pass control used?

Answer: By-pass control may be:

1. Throttling, used with a constant-speed driver for continued but varying delivery, as for boiler feeding. It is wasteful, because the pump always works at full load.
2. Unloading, used in hydraulic systems for intermittent delivery to an accumulator. It is economical and satisfactory for small pumps, but in larger systems the by-pass valves usually operate with enough shock to reduce packing life and eventually damage the hydraulic system.

Question 14-17: What is synchronized suction-valve unloading?

Answer: This control system holds the suction valves open when delivery is not needed, and allows the valves to seat for normal operation. With valves held wide open, fluid passes freely in and out of the pulsation chambers, thus imposing minimum load on the driver. With proper synchronization, delivery may start and stop as often as desired without shock or disturbance. Delivery starts from zero at start of stroke of any one plunger, increasing gradually with plunger velocity during half the initial delivery stroke. During unloading, delivery tapers off with plunger velocity during the second half of the stroke of the last plunger to unload.

Question 14-18: What are major differences between vertical and horizontal power pumps?

Answer: Nearly all vertical pumps are single-acting, whereas double-acting pumps predominate in horizontal designs. Single-acting arrangements more or less dictate outside-packed plungers for all pressures in vertical pumps, as contrasted to piston liquid ends for a considerable pressure range in horizontal types. A three-plunger or triplex arrangement is common for vertical pumps, with five-plunger or quintuplex units as an alternate. Most horizontal pumps use two crank throws with a double-acting fluid end to obtain a four-cylinder effect. Vertical pumps tend toward shorter stroke, higher rotative speeds, and greater unit plunger loads than horizontal pumps of equivalent power ratings. Thus, for a given pressure, vertical pumps have larger plungers and shorter strokes than horizontal.

Question 14-19: What are the common types of vertical power pumps?

Answer: They are:

1. Open-frame, low- or moderate-speed.
2. Semienclosed frame, essentially a larger and higher quality version of an open-frame pump with arrangements for better lubrication of all moving parts.
3. Totally enclosed frame arranged with full circulating lubricating system for higher operating speeds.
4. Inverted vertical pump with totally enclosed frame, and with fluid cylinder above power end.
5. Variable-stroke, totally enclosed frame pump.

Question 14-20: What is an open-frame vertical power pump?

Answer: Figure 14-8 is a cross section through one cylinder of typical open-frame vertical triplex power pump. One casting makes frame, cylinders, and crosshead guides; valves and piping connections are in a separate bolted-on valve chest. The plunger and crosshead are combined in a single hollow casting, and stuffing boxes are integral with the cylinder. Except in larger sizes, the crosshead end of the connecting rod has a nonadjustable bronze bushing.

Question 14-21: What is a semienclosed vertical power pump?

Answer: It is generally like the open-frame pump plus sheet-steel enclosures around gears and crankshaft, connecting rods, and

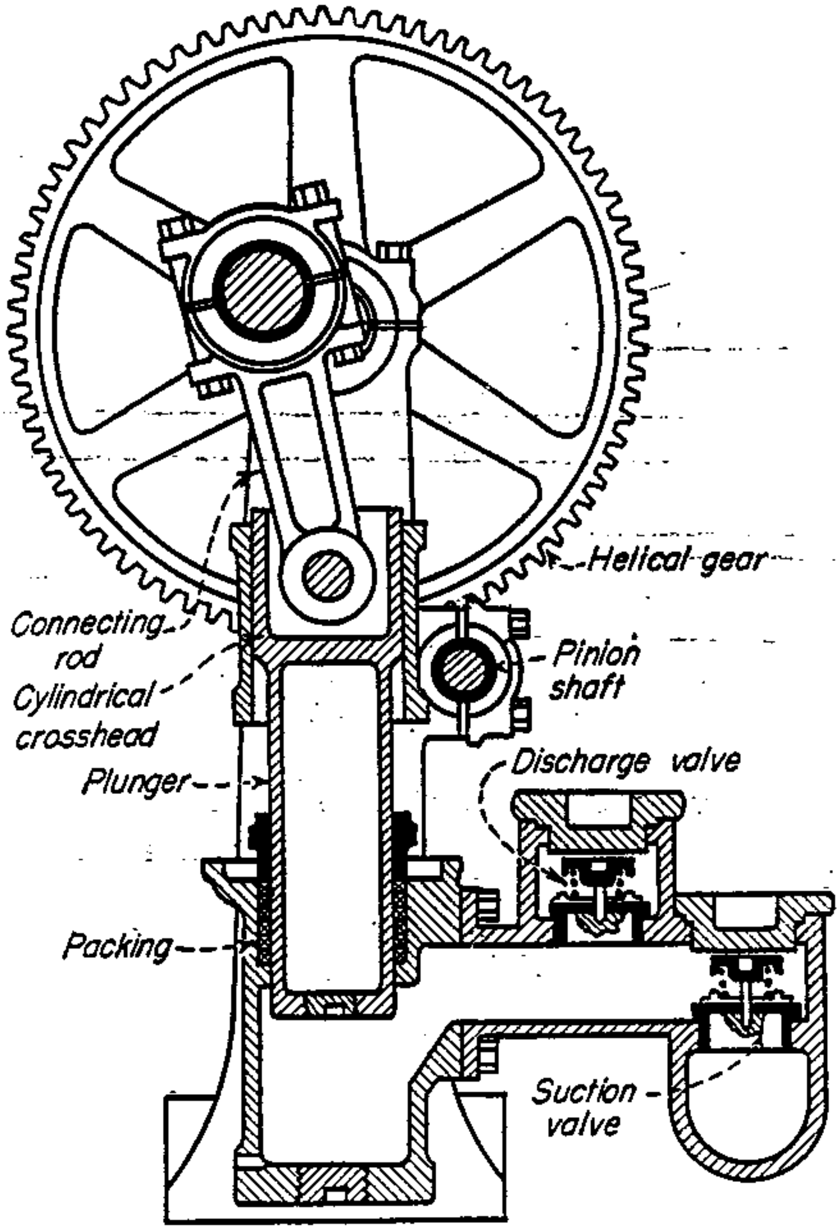


FIG. 14-8. Cross section through open-frame vertical triplex plunger-type power pump.

crossheads. Larger sizes are built in a variety of arrangements. This type has a better lubricating system than the common open-frame pump. Usually built for higher pressures, the fluid end has plunger pulsation chambers and valve passages bored in a solid steel forging.

Question 14-22: What is a totally enclosed vertical power pump?

Answer: All moving parts of the power end, including reduction gears, are enclosed in an oil-tight casing. Figure 14-9 shows this type designed primarily for marine boiler feed service. Roller

FIG. 14-9. Totally enclosed vertical power pump primarily for boiler feed service.

bearings carry a three-throw crankshaft and double reduction gears. The connecting rods are marine type, with removable babbitt-lined crankpin bearings. Crossheads and crosshead-pin bearings are nonadjustable. There is a removable section between the plunger and the crosshead, so the plungers can be removed without disturbing any major pump part. A separate chest bolted to the cylinder casting contains valves and piping connections.

Question 14-23: What is an inverted vertical power pump?

Answer: Figure 14-10 shows a modern inverted vertical triplex pump. This arrangement provides maximum accessibility to stuffing boxes and, at the same time, effectively separates the fluid

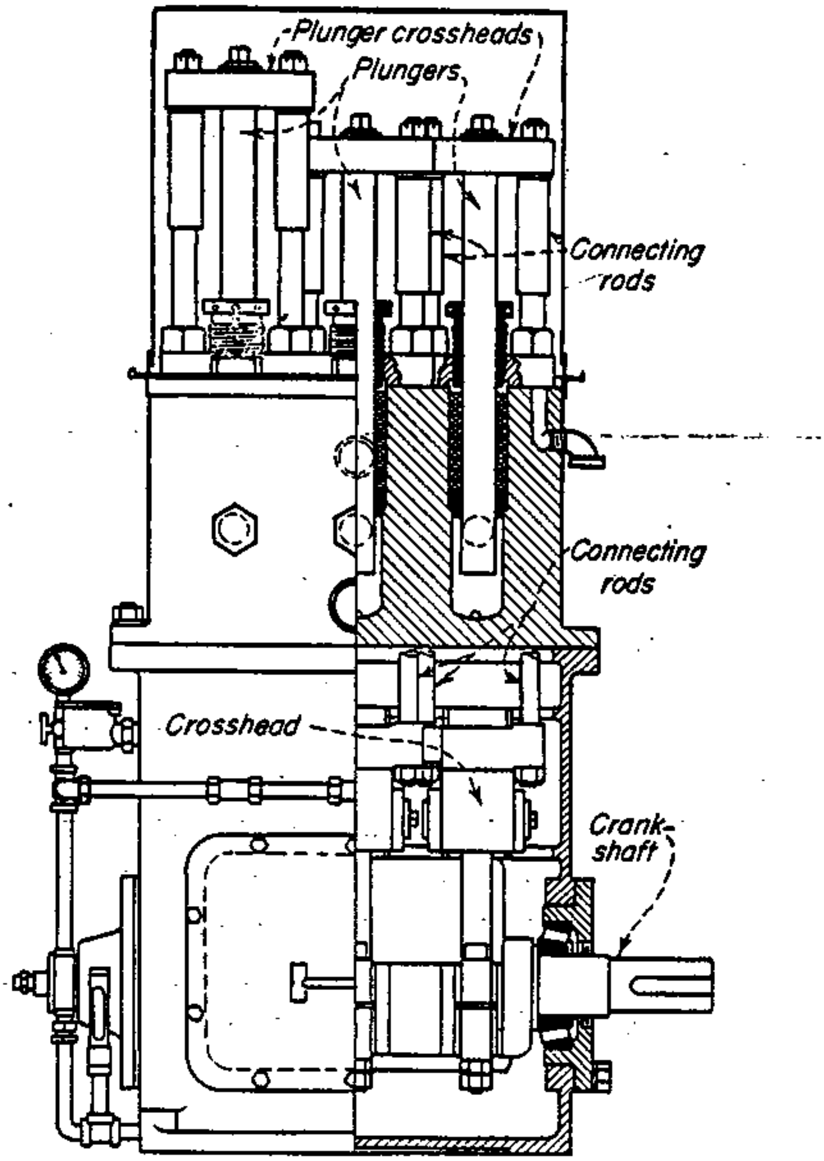


FIG. 14-10. Modern inverted vertical triplex power pump.

end from the enclosed power end. Rods extend through holes in the forged cylinder block from the power-end crossheads to the plunger crossheads above the cylinder. Telescoping pipes between the cylinder top and plunger crossheads prevent the entry of dirt or water into the power end. Plungers are easily removable for replacement, examination, or renewal of the plunger packing.

This type of pump runs at a sufficiently high speed to permit using a single-reduction belt drive from standard motors or direct coupling to gear-head motors. The largest sizes sometimes have built-in reduction gears.

Question 14-24: What is a variable-capacity power pump?

Answer: Plunger-stroke length of this special power pump can be varied between zero and maximum to obtain full control of discharge with constant speed. Variable-stroke power pumps fall into those which (1) provide complete separation of fluid and power ends, (2) have plungers and cylinders in the same compact housing with what we consider the power end of the ordinary pump.

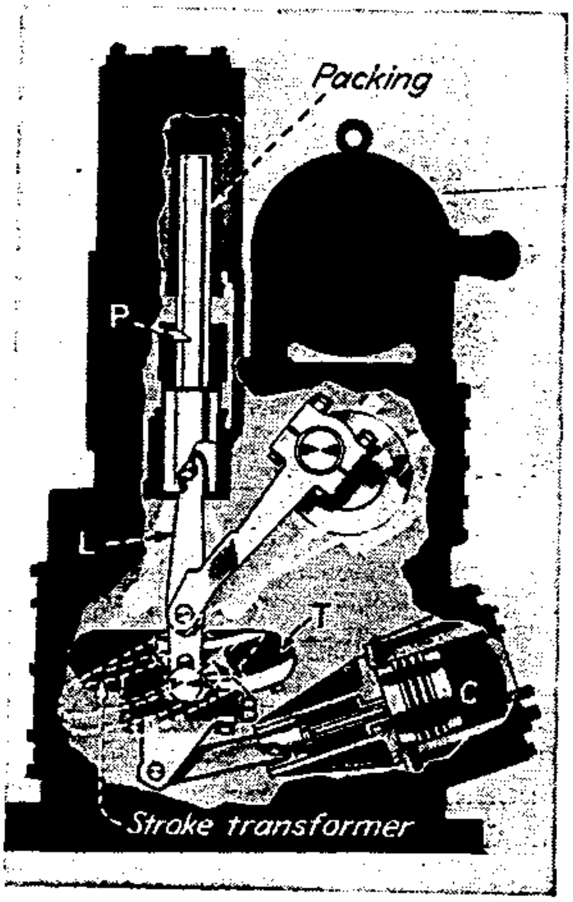


FIG. 14-11. Variable-linkage plunger power pump. Stroke and discharge are controlled by adjustable curved track.

The second arrangement is applicable only to specialized hydraulic services where the clean oil pumped is suitable to lubricate all pump parts. The general service pump for handling any fluid desired has conventional plungers, stuffing boxes, and valves in a separate fluid cylinder.

Pumps of the type first described may be divided into three groups: (a) variable-linkage, (b) variable-eccentric, and (c) swash-plate or wobble-plate.

Question 14-25: What is a variable-linkage power pump?

Answer: The two forms are: Aldrich-Groff (Fig. 14-11), and Worthington (Fig. 14-12). In the pump (Fig. 14-11), the top end of link *L* connects to plunger *P*, and its lower end is guided in an arc by a guide block B sliding in a curved track T in the stroke transformer. The radius of this track is such that with the track in a horizontal or zero stroke position the guide block B oscillates from one end of track T to the other without imparting motion to the plunger. For example, assume a clockwise rotation of the crank; as it rotates, it pulls the lower end of link L and its guide shoe to the right in the track. When the crank passes dead center,

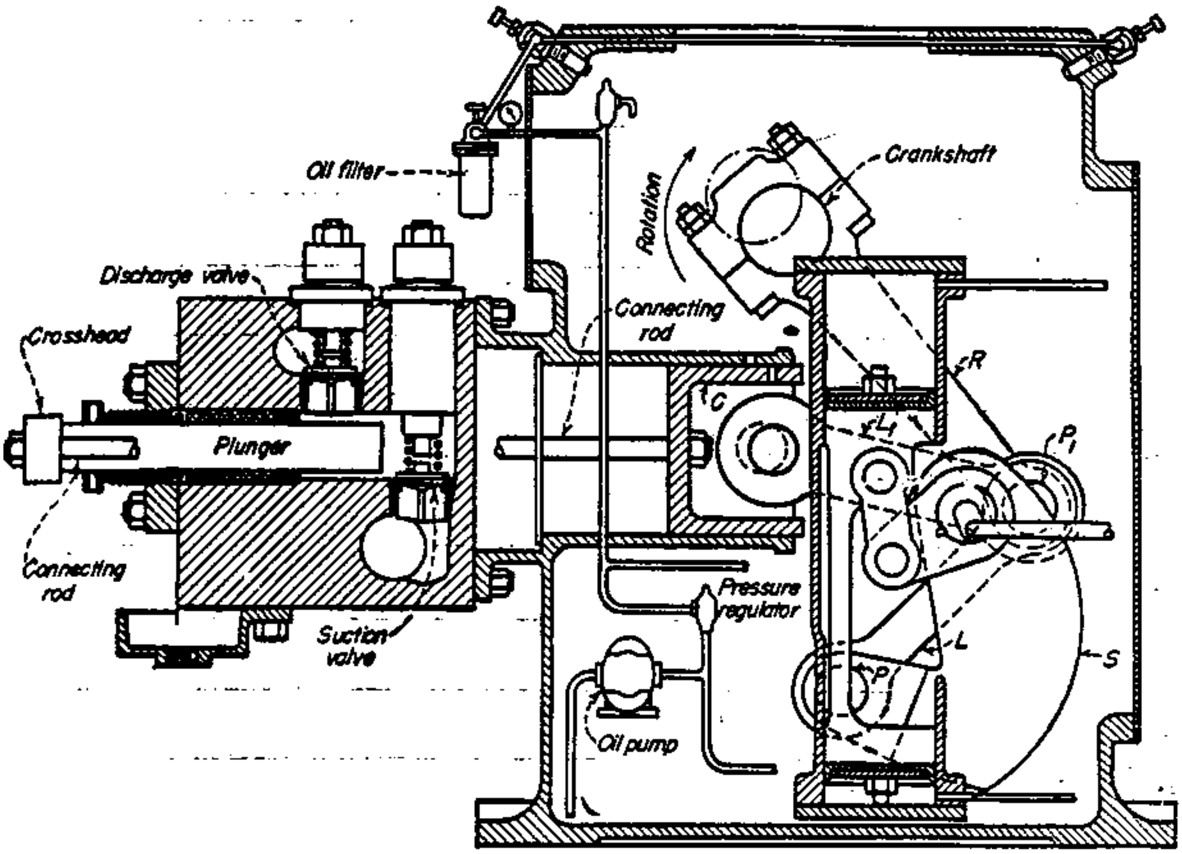


FIG. 14-12. A variable-linkage plunger power pump whose stroke and discharge are controlled by a link swinging from a fulcrum on an adjustable cradle.

it swings link L and its guide shoe to the opposite end of track T. Plunger P, however, remains motionless.

The stroke transformer is supported on two end trunnions to permit tilting the curved track. When hydraulic pressure is applied to the left end of the piston in adjustment-cylinder C, it moves to the right. A full piston stroke tilts the track T to the dotted-line position.

Now, when the crank turns clockwise, the lower end of the link and its guide block B are pulled up the track, thus imparting an upward motion to the plunger. During the other half of the crank's revolution, it moves guide block B with the link to the low end of track T. This operation pulls down the plunger for a full stroke. By moving the curved track toward zero stroke, any desired stroke length and discharge is obtained.

Figure 14-12 gives the same control of the plunger stroke and discharge as Fig. 14-11, by an additional link swinging from a

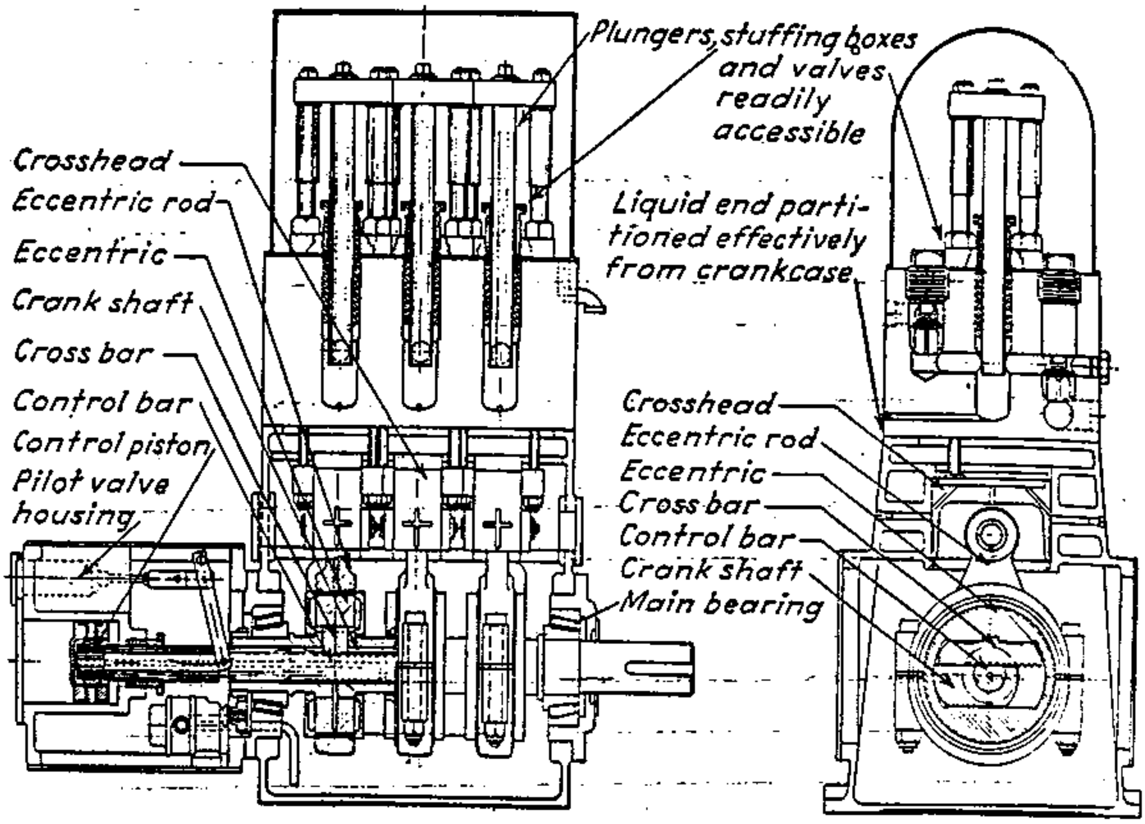


FIG. 14-13. Cross sections through variable eccentric power pump.

fulcrum on an adjustable cradle. In Fig. 14-12, fulcrum pin P is in position for maximum stroke. One end of fulcrum link L connects to stroke-changer S by fulcrum pin P. The other end of L connects to master pin P_1 along with one end of the plunger's crosshead link L_1 and connecting rod R. Assume clockwise rotation of the crankshaft. As it rotates, it pulls master pin P_1 to the left, with connecting link L_1, plunger crosshead C, and plunger. This is the suction stroke. When the crank goes over dead center, it begins to return the linkage to the position shown, to complete the discharge stroke.

If fulcrum pin P and link L are swung up until they are parallel with crosshead link L_1 the plunger remains at rest. The only result of crank rotation is an up-and-down motion of master pin P_1 and link ends connecting to it. As there is no movement of the pump plunger, discharge is zero. By positioning fulcrum pin P between zero and maximum stroke, the discharge from the pump is varied between zero and maximum values.

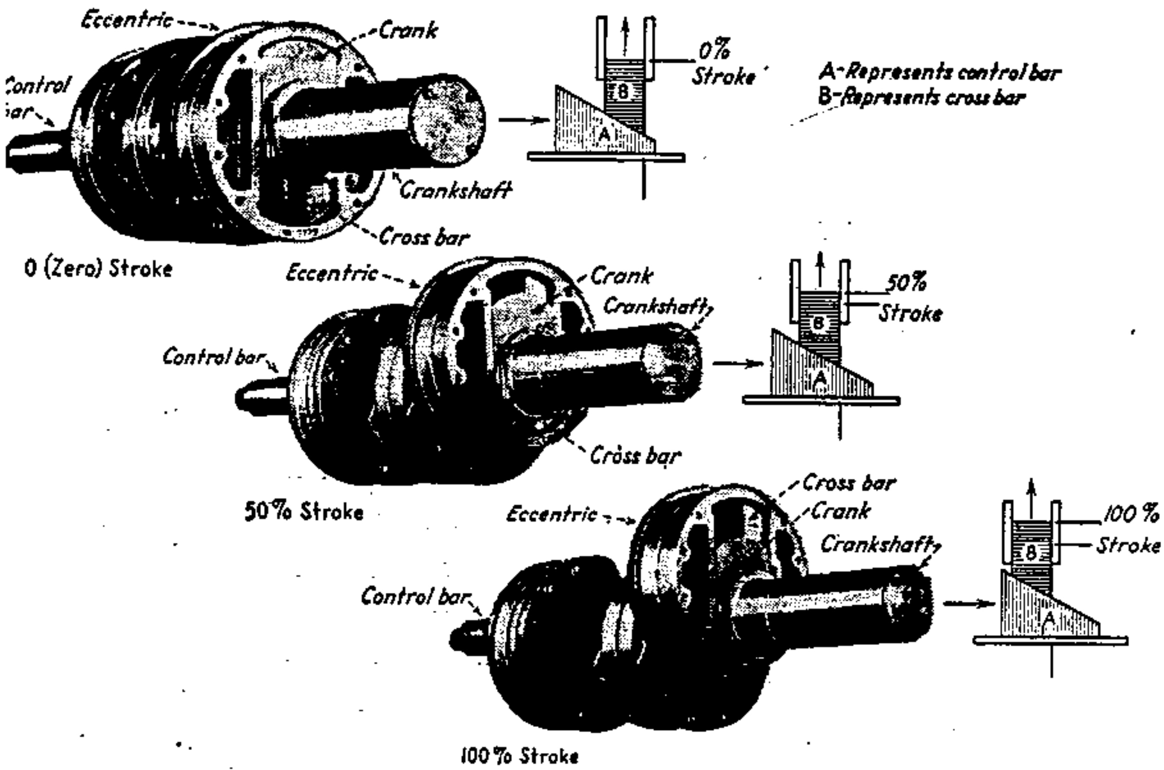


FIG. 14-14. Eccentric shaft assembly with eccentrics in various positions showing wedge action of control bar with eccentric cross bars.

Question 14-26: What is a variable-eccentric power pump?

Answer: Figures 14-13 and 14-14 show a variable-stroke arrangement built into the eccentric of this pump. Except for the variable-stroke feature, this pump is like the inverted vertical-triplex (Fig. 14-10). In Figs. 14-13 and 14-14, the variable eccentric is concentric with the crankshaft and, therefore, in position of zero stroke and discharge. If the eccentric were moved to the left against the crankshaft, it would have maximum throw, to produce maximum plunger stroke and discharge. Eccentric throw

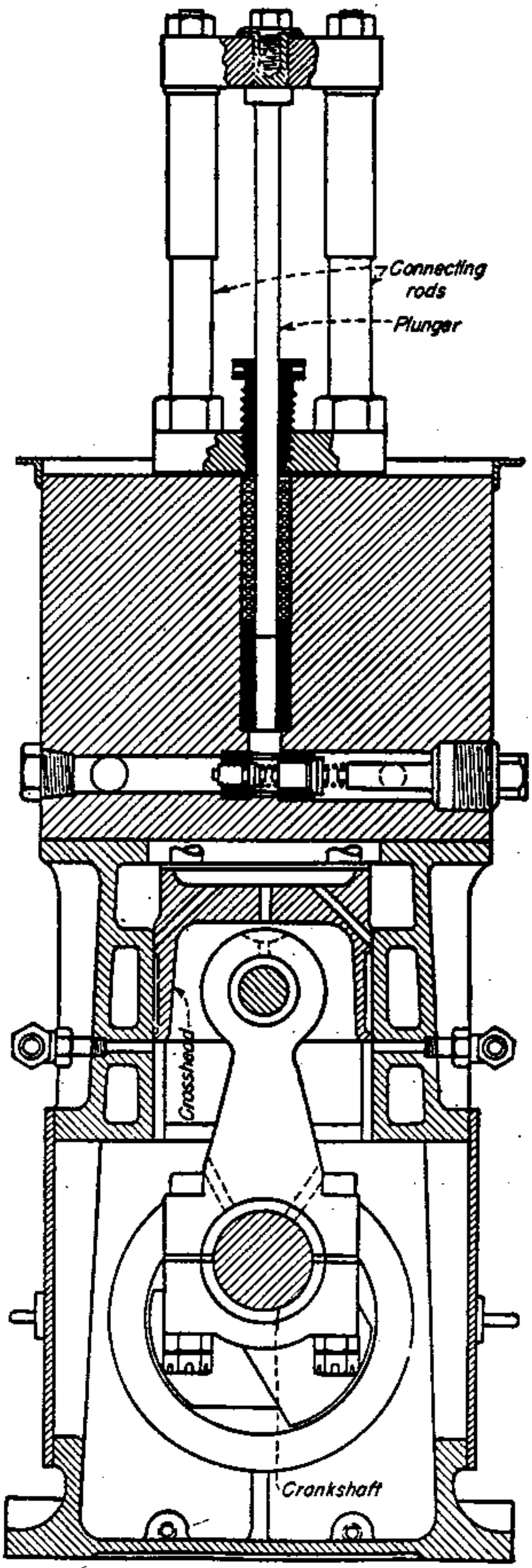


FIG. 14-15. A close-clearance high-pressure pump for high volumetric efficiency.

is adjusted by a mechanism within a hollow crankshaft operated by a hydraulic cylinder.

Question 14-27: What is a swash- or wobble-plate variable-stroke pump?

Answer: There are several basic arrangements. The general service one has its plungers in a circle with their axes parallel to the center line of the drive shaft and attached to the wobble plate by ball-jointed connecting rods. The wobble plate is prevented from rotating, but rides on roller bearings against a tilting plate that adjusts the length of the plungers' stroke and discharge.

Question 14-28: What is a close-clearance plunger pump?

Answer: This special pump is used when the discharge pressure is high enough for the compressibility of the fluid pumped to have a decided effect on the volumetric efficiency of the pump. A close-clearance design is desirable when pumping water to pressures above 5000 psi. Most oils, particularly the lighter fractions, have a compressibility much greater than water, and a close-clearance design may be desirable at pressures as low as 2000 psi. Figure 14-15 shows a small hydrostatic test pump for 10,000 psi. Despite high pressure, its close clearance makes it possible to obtain volumetric efficiency in excess of 90 per cent.

Question 14-29: What types of bearings are used in vertical power pumps?

Answer: Except main crankshaft bearings, vertical and horizontal power pumps use very much the same types of bearings. Unlike the horizontal pump, the vertical pump has main-bearing working loads along the same line as the load due to the weight of the parts. Thus, vertical pumps may use simple, two-part, horizontally split main crankshaft bearings. Figure 14-8 shows this type. Since the principal working load is vertically upward, the bearing cap must be firmly held, as in this case by four instead of two studs. Large semienclosed pumps sometimes have wedge-adjusted main bearings. Totally enclosed vertical triplex pumps often use tapered roller bearings at each end to support the crankshaft, although the largest sized triplex pumps usually have sleeve bearings between each crank throw.

Question 14-30: What types of variable-capacity plunger pumps are used for closed circuit oil hydraulic applications?

Answer: Two basic types are often used: (1) axial-piston variable-stroke, and (2) radial-piston variable-stroke. They are built in several designs by the American Engineering Company, Northern Pump Company, Oilgear Company, Sundstrand Machine Company, Vickers, Inc., Hydraulic Press Manufacturing Company, and the Superdraulic Corporation.

CHAPTER 15

CHARACTERISTICS OF STEAM AND POWER PUMPS

Question 15-1: What are the major advantages of a direct-acting steam pump?

Answer: A direct-acting pump is simple, flexible, reliable, and low in cost. It is a complete unit, consisting of the pump or fluid end and the driving element or steam end, and does not require a separate motor or other power source. Simple and inexpensive to install, the pump is easy to operate and control, and most maintenance does not require highly skilled mechanics. Because of this unique combination of advantages, a large number are still built with virtually no fundamental changes in design or construction from those established almost a century ago.

Question 15-2: For what pressure and capacity ranges are direct-acting pumps built?

Answer: Their greatest uses are in the moderate-capacity and -pressure field, and they are often used as stand-by pumps. Where the ratio of pressure to capacity is large or where the viscosity of the fluid pumped is high enough to be important, these pumps have distinct advantages. They are most economical where the exhaust steam is used for heating or process work.

Question 15-3: What factors govern the maximum pressure that direct-acting pumps develop?

Answer: Primary factors are:

1. Available steam pressure.
2. Ratio of steam-piston and fluid-piston areas.
3. Operating speed.

Exhaust pressure and suction pressure or lift also influence the pressure developed.

Figure 15-1 diagrams a direct-acting pump. The motion of the two pistons, and the piston rod connecting them is proportional to the unbalance in forces acting on the pistons. The maximum or stalling discharge pressure is nearly equal to effective steam pressure times steam-piston area divided by fluid-piston area. As the pump speed increases, developed discharge pressure decreases because of fluid friction in the steam and liquid ends, and packing and mechanical friction.

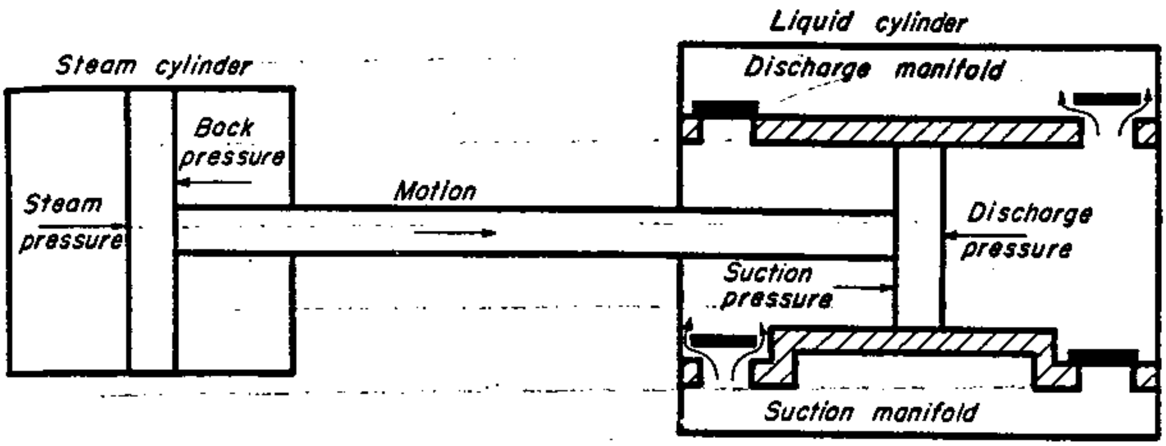


FIG. 15-1. Simplified diagram of direct-acting pump showing direction of forces on pistons. Rate of motion is proportional to unbalance of forces acting on them.

Question 15-4: What method is used to control the delivery of a direct-acting steam pump?

Answer: The steam supply is throttled, either by the operator or by an automatic device to meet the requirements of a specific service. Since a direct-acting pump is a variable-speed machine, it is never necessary to throttle its discharge, as on a constant-speed centrifugal. A direct-acting pump may be controlled by the liquid level, through a float and linkage to a throttle valve, or by a constant-pressure or excess-pressure governor. The latter is well adapted to a boiler feed pump where the boiler pressure may fluctuate through relatively wide limits. A by-pass is not needed for low or zero capacity, because the governor may stop or start a direct-acting pump to meet any conditions.

Question 15-5: Is a pressure-relief valve necessary on the discharge side of a direct-acting pump?

Answer: While many such pumps operate for years without a relief valve, it is usually a good investment. Stalling pressure for most pumps is at least 50 per cent greater than operating pressure, because of losses in the pump and pressure drop in the steam line and control valves. When the steam-piston area is much larger than the liquid-piston area, as on boiler feed pumps, maximum discharge pressure may be from 2 to 2.5 times boiler pressure. Under certain conditions, simplex pumps occasionally build up pressures greatly in excess of their normal rated pressure.

Question 15-6: Is the inherently high steam consumption of direct-acting pumps a disadvantage?

Answer: It is not, when exhaust steam is used for heating feed water, for building heating, or process work. Because these pumps can operate with a considerable range of back pressure, it is possible to recover nearly all the heat in the steam required to operate them. Since they do not use steam expansively, they are actually metering devices rather than heat engines, and as such consume heat from the steam only as it is lost in radiation from the steam end of the pump. These pumps act, in effect, like a reducing valve to deliver lower pressure steam that contains nearly all its initial heat.

Question 15-7: How does the operating steam pressure affect the steam consumption of a direct-acting pump?

Answer: The weight-to-volume relationship of saturated steam at various pressures is such that the theoretical steam consumption in terms of pounds per hp-hr is a minimum in the 400-to-500 psi range, and increases more and more rapidly as pressure decreases. It was generally believed that a compound or triple-expansion pump was required to get steam consumption less than 80 or 90 lb per liquid hp-hr. Marine experience during the last ten years shows that a simple pump can operate on less than 50 lb per liquid hp-hr when the steam-cylinder pressure approaches 400 psi and the steam end is properly insulated. Where steam consumption is an important factor and sufficient steam pressure is available, considerable saving can be made by selecting a steam-cylinder size requiring the highest practical steam pressure.

Question 15-8: Does the steeply sloping exhaust line of indicator card (Fig. 15-2) indicate serious restriction in the exhaust passages or piping?

Answer: While it is not common for operators to take indicator cards from direct-acting steam pumps, a card like Fig. 15-2 should not cause alarm. The moving assembly of these pumps is a delicately balanced mechanism, sensitive to pressure change on both sides of each piston. If the discharge pressure builds up gradually during most of the stroke, the piston assembly starts moving as soon as the steam-piston exhaust pressure drops appreciably below the steam pressure. Motion continues at whatever rate is required to maintain the moving balance of forces, with exhaust pressure dropping as discharge pressure rises.

FIG. 15-2. Steam-end indicator card taken from simplex direct-acting pump.

This card is typical of a simplex pump with a small-discharge cushion chamber on the water end, in which the pressure drops sharply while the pump is reversing and builds up gradually during most of the stroke. The same pump would produce an almost rectangular indicator card if discharging through a short, open line to a boiler or to a large receiver with practically constant pressure. With uniform resistance on the discharge side of the fluid piston, the pump's pistons will not move until the steam-cylinder exhaust pressure approaches the exhaust-line pressure.

Question 15-9: Because a direct-acting pump is a positive-displacement machine, capable of priming itself at considerable lift, does it have good "sucking" qualities?

Answer: No machine or device can "suck" fluid from a lower level. Piston motion can do no more than lower the pressure within the cylinder to the point where atmospheric pressure on an open suction supply can force the liquid up the suction pipe into the pump cylinder.

Question 15-10: How does clearance volume affect the priming ability of direct-acting pumps?

Answer: In a dry pump—without liquid in the cylinders—the ratio of clearance volume to displacement has a direct effect on the pump's priming ability. To evacuate the suction line, the pump must compress air within the cylinder and discharge some of it on each stroke. Before additional air can be taken in from the suction line, the air remaining in the clearance volume must expand to less than the suction-line pressure. Thus a large clearance volume greatly limits the priming ability of a dry pump.

Filling the clearance volume with fluid largely nullifies its effect. In a submerged piston pump, such as the common cap and valve-plate design, filling the cylinder with liquid makes the pump self-priming at any reasonable lift.

Question 15-11: What is the maximum practical suction lift of a direct-acting pump?

Answer: With cold water at sea level, an average pump can operate at a lift of about 22 ft. Figure 15-3 shows the influence of temperature and altitude on a pump capable of a maximum lift of about 26 ft. The difference between the maximum theoretical lift, 34 ft, and the maximum practical lift represents losses through the valves and entrance losses, velocity head, and pipe friction in the suction line.

Question 15-12: What vacuum can be maintained with a wet-vacuum pump?

Answer: A wet-vacuum pump, when well designed, can maintain a vacuum nearly equal to condensate vapor pressure, as long as air leakage is not excessive. This is high enough to obtain the best over-all efficiency from a compound or even a triple-expansion reciprocating steam engine. Special wet-vacuum pumps applied to process work maintain vacuums as high as 29 in. Hg.

Question 15-13: With water at 212°F, what suction head does the average direct-acting pump require?

Answer: Assuming standard atmospheric pressure on the water surface, the average pump at normal speed requires a positive suction head of about 10 ft. This should be expressed as net positive suction head (NPSH) of 10 ft. The NPSH, or head in feet of

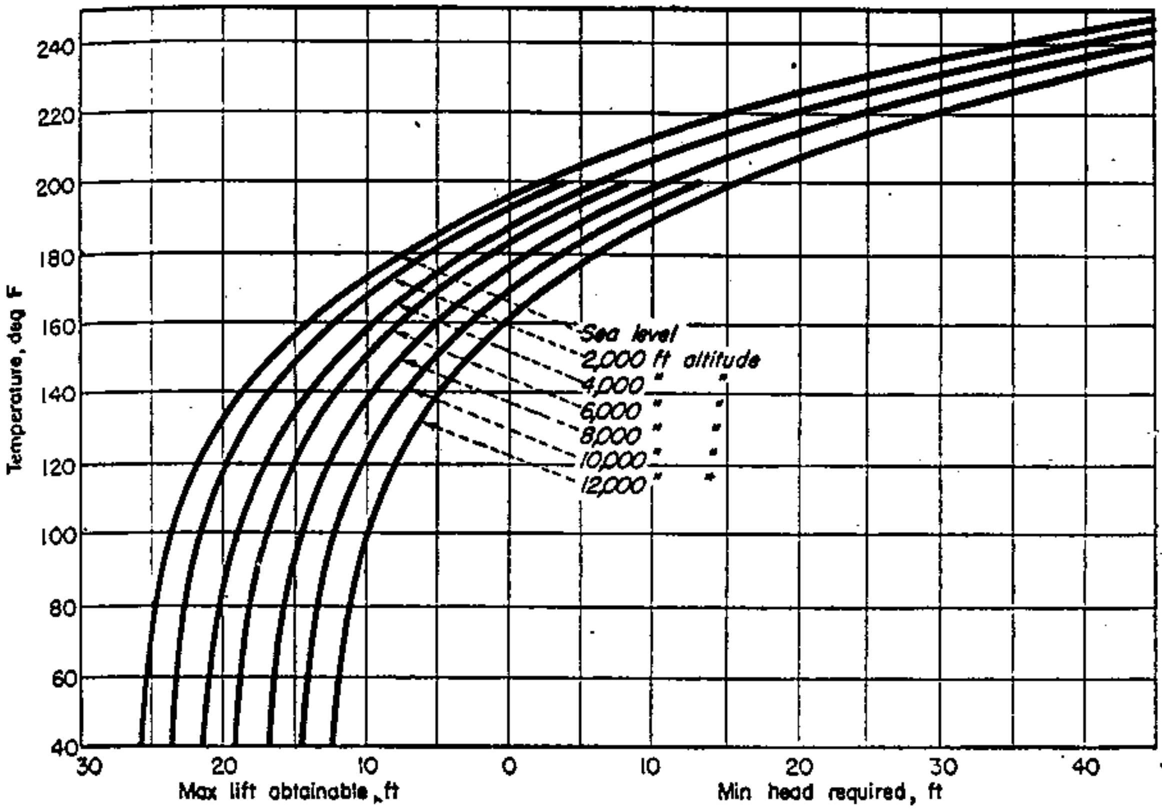
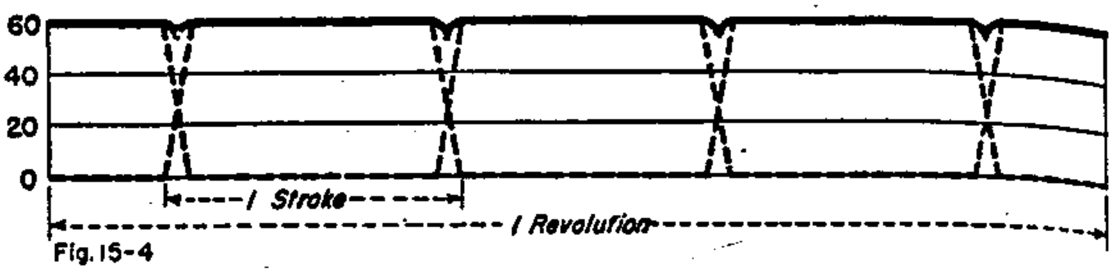


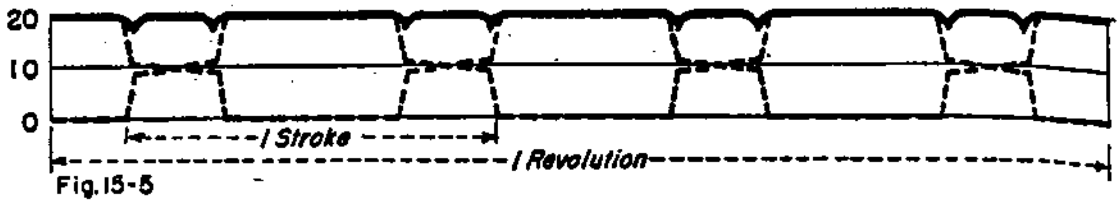
FIG. 15-3. Curves showing maximum suction lift obtainable and minimum suction head required by a reciprocating pump handling water at different temperatures.

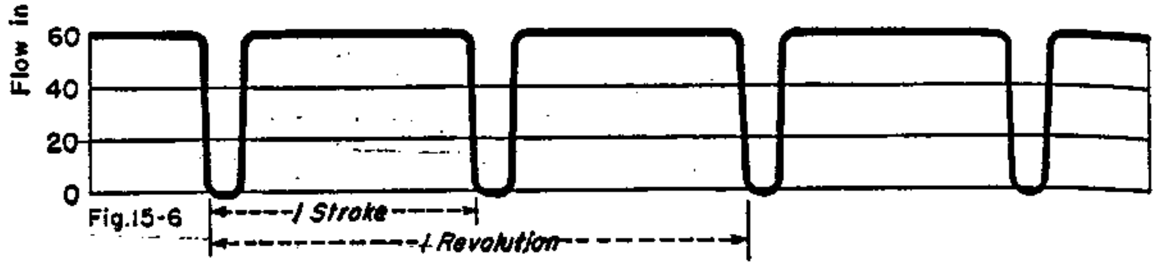
liquid pumped in excess of vapor pressure, should always be the basis of comparison. The NPSH of 10 ft of 212°F water is about equal to lift of 24 ft of 65°F water. For details on NPSH see Question 4-45.

Question 15-14: What are the flow characteristics of a duplex direct-acting pump at normal speed?

Answer: In theory, a duplex pump should maintain steady flow and constant pressure. Practically, influences of moving parts' inertia, mechanical and fluid friction, slip, and unbalanced area of valves make this ideal unattainable. A properly adjusted duplex pump should have a flow characteristic approximating Fig. 15-4. While actual flow may be nearly as shown, pressure may fluctuate at each reversal because of the rapid acceleration and slowing down of moving parts. At rated speed, there is almost no







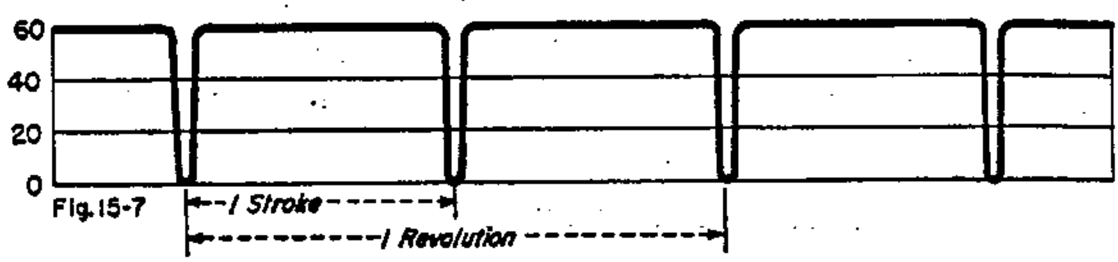


FIG. 15-4. Fluid end flow curves for a 6×4×6 duplex pump operating at full speed.
FIG. 15-5. Same as Fig. 15-4 with pump at one-third speed.
FIG. 15-6. Flow curve of 8×5×12 simplex pump under boiler-feed conditions.
FIG. 15-7. Flow curve of 5½×5½×7 simplex pump operating under tank-pump conditions.

overlapping of strokes on the two sides of the pump. One piston is at rest while the other is moving at a velocity nearly constant throughout 90 to 95 per cent of its stroke. At the end of the stroke, this piston stops quite sharply while the other starts almost as quickly. At normal speed, the average pump makes a stroke slightly greater than the rated stroke.

Since one side of a duplex pump is idle most of the time, simplex-pump enthusiasts claim that the duplex is no better than a simplex of the same cylinder size. However, there is a great difference between the sustained, if not the steady, flow of a duplex and the interrupted flow of a simplex. Very few people appreciate how much effect minor adjustments can make in the smoothness of operation of a duplex steam pump.

Question 15-15: What are the flow characteristics of a duplex pump at reduced speed?

Answer: Figure 15-5 shows the flow characteristics of a duplex at reduced speed. The overlapping stroke frequently assumed to be characteristic of the duplex is obtained, but unfortunately only at a reduced stroke. Each piston moves independently during the middle half of each stroke, and at a velocity essentially constant. When the opposite piston starts at low speed, the velocity of the already moving piston is reduced quickly to the point needed to maintain approximately average delivery. When one piston stops, the other quickly accelerates to reestablish average delivery. At full speed, the moving parts have sufficient inertia to maintain nearly constant piston velocity. This is not true at low speed, when varying frictional forces exert a stronger influence, causing minor but frequent variations in piston velocity.

Question 15-16: What are the flow characteristics of a simplex steam pump?

Answer: Figure 15-6 shows the flow characteristics of a simplex steam pump under boiler feed conditions, where the discharge pressure is fairly well sustained except when the pump is at rest. Figure 15-7 shows performance under tank-pump conditions, where the discharge pressure is nearly proportional to the piston velocity. In a simplex, a pause is necessary at each end of the stroke during the time required for the forces on both sides of the steam and liquid pistons to reverse.

The time of this piston pause may seem exaggerated even to the experienced operator, but actual tests show that under boiler feed conditions the pause may seem short and still take 15 to 25 per cent of the time of a complete single stroke. This pause at reversal may seem a serious disadvantage but is necessary to the proper functioning of the fluid valves. Since the piston velocity is essentially constant until near the end of the stroke, the liquid valves cannot begin to close until just before reversal. An appreciable pause is needed for the valves to seat gently instead of being slammed against their seats by a sudden flow reversal.

Question 15-17: Can a direct-acting simplex boiler feed pump designed for a 400-psi boiler be used on a system operating at 185 psi maximum?

Answer: Yes, but all pump parts are unnecessarily heavy and costly for such operation. All steam pumps for boiler feed service are designed with the steam-piston area from 2.25 to 2.5 times the liquid-piston area, to ensure sufficient pressure to overcome boiler pressure, plus resistance of the boiler feed regulator and pipe friction. The average boiler feed pump never requires the full available steam pressure, even at excess speed with high exhaust pressure; instead, it always has plenty in reserve to meet any condition. Operation at lower boiler pressure may be compared to conditions existing while a boiler is being brought up to pressure. During this period, the direct-acting pump has no difficulty in feeding water to the boiler at 50, 100, or 200 psi, even though the plant is designed for 400 psi.

Question 15-18: What are the primary advantages of power pumps?

Answer: The primary advantages are:

1. High efficiency. At or near full load, large power pumps attain an over-all efficiency in excess of 90 per cent. Even in the smaller sizes, efficiency is rarely less than 80 per cent. With a reasonable amount of attention to the condition of valves and plunger packing, these efficiencies may be maintained indefinitely, since normal wear does not result in gradually increasing internal leakage.

2. Ability to develop high pressures, even at small capacities, without materially reducing efficiency.
3. Simplicity and reliability. Modern self-oiling power pumps may be run for long periods with a minimum of attention.
4. Applicability to viscous liquids. Viscosities up to 2500 SSU have little effect on power-pump performance. Moderate-speed pumps operate satisfactorily up to 10,000 SSU, and special low-speed pumps are built to handle liquids having a viscosity as high as 100,000 SSU.

Question 15-19: Through what range of capacity and pressure are power pumps applicable?

Answer: Small power pumps are applicable from moderate to the highest pressures. Small power pumps have been built for 10,000 to 15,000 psi, and pumps for 10 to 12 gpm are being constructed for higher pressures. In small power plants and marine installations, motor-driven reciprocating boiler feed pumps gain in advantage as operating pressures rise above 450 psi. Power pumps serving hydraulic presses and shears operating at 4500 to 6000 psi and requiring 1200-hp drivers were built for war plants. Modern reciprocating oil pipe-line pumps are built for 1200 to 1800 gpm and 800 to 1200 psi. The large moderate-pressure power pump of thirty years ago has been replaced by the centrifugal pump; but as capacities are reduced and pressures increased, the power pump becomes more and more applicable.

Question 15-20: What volumetric efficiency is attained by power pumps?

Answer: Most power pumps have a volumetric efficiency of 96 to 98 per cent, as long as valves and packing are in reasonably good condition. As pump speed is increased, more attention must be given to valve action and to the valve spring loading. While the delivery of each plunger drops gradually to zero as the end of the stroke is approached, instant reversal makes it necessary to load the valves sufficiently to bring them almost to their seats at the end of the stroke. If this is not done, considerable backflow or slip results, and considerable shock accompanies valve seating.

Packing leakage from outside-packed plunger pumps is readily noticeable and is easily kept under control, but packed-piston pumps require more care to ensure against abnormal packing leakage. At extremely high pressures, fluid compression may reduce volumetric efficiency considerably unless special attention is given in the design to keep the clearance at a minimum.

Question 15-21: Does volumetric efficiency have a direct influence on the over-all efficiency of a power pump?

Answer: Loss in volumetric efficiency due to slip of valves and packing directly influences over-all efficiency. At higher pressures and with certain fluids such as the lighter oils, fluid compression has a marked influence on volumetric efficiency. While some of the work of fluid compression is lost in heat, most of it is recovered during expansion at the start of the suction stroke, and hence it has relatively little influence on over-all efficiency. One maker of a small high-pressure oil pump having a relatively large clearance volume publishes test curves showing an over-all efficiency about 30 per cent greater than volumetric efficiency at full pressure.

Question 15-22: Can we say that a power pump has definite operating characteristics?

Answer: Yes. Each type of power pump has certain flow and acceleration characteristics, which are modified to some extent by details of design, but which are not changed by the piping and other features of the rest of the pumping system. A direct-acting steam pump adapts itself to some extent to the characteristics of the entire pumping system. A power pump has no such flexibility and labors to make the flow in the entire system correspond to its particular flow pattern, except where adequate cushion chambers may be used on both suction and discharge of the pump. While the flow pattern of a pump cannot be changed, its influence on a given piping system is subject to simple analysis to determine whether cushion chambers must be used or the piping modified to suit pump characteristics.

Question 15-23: What are the flow characteristics of a double-acting duplex power pump?

Answer: Figure 15-8 shows how the velocity and hence the rate of displacement or flow from the individual plungers resembles the sine curve of simple harmonic motion, but is distorted by the effect of the moderate-length connecting rod. The amount of this distortion is decreased as the ratio of connecting-rod length to

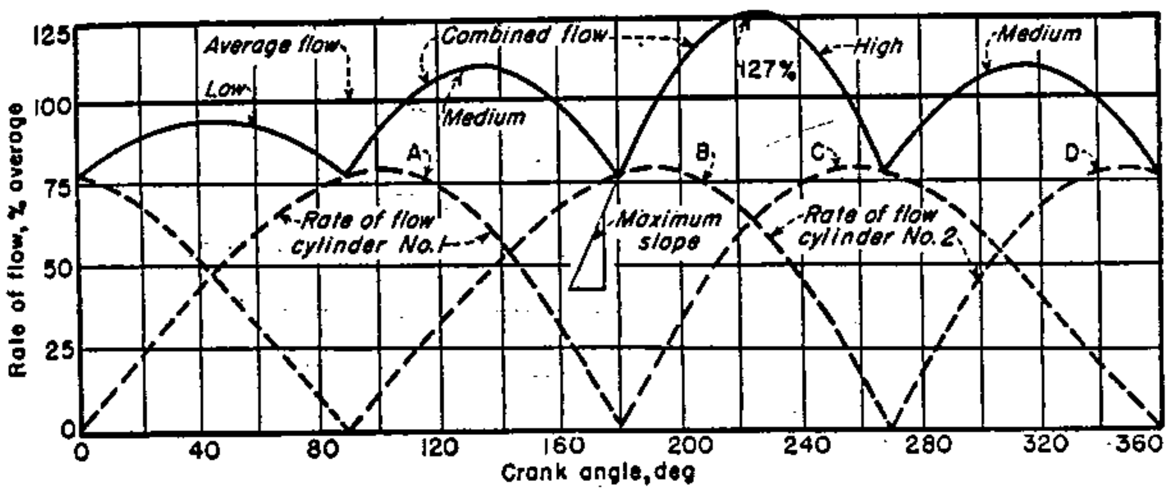
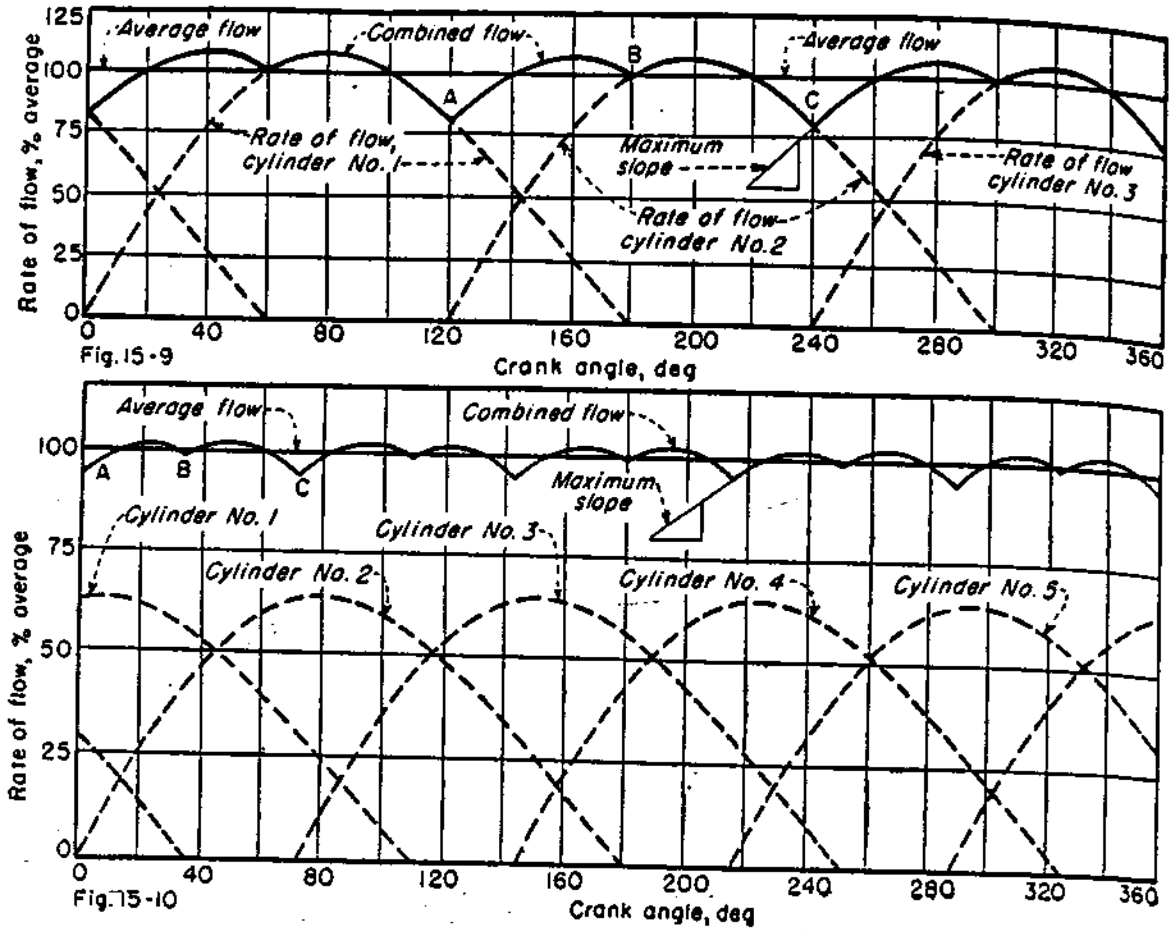


FIG. 15-8. Flow characteristics of each cylinder and combined discharge curve, for a double-acting power pump when cranks are displaced 90 deg.

crank radius is increased. As plotted, curves *A* and *B* lean to the right and *C* and *D* to the left. This causes the periodic variation in the combined velocity or flow curve, which has four peaks: low, medium, high, and medium.

Question 15-24: What are the flow characteristics of single-acting triplex and quintuplex power pumps?

Answer: Figures 15-9 and 15-10 show the flow curves for these pumps. Since they are single-acting, all the curves of the individual plungers lean in the same direction, and there is no variation in peak flow, as for a double-acting duplex pump (Fig. 15-8). Instead, an identical variation in flow is repeated three and five times per revolution. Because of the incomplete overlapping of strokes, half of the combined curve of the triplex pump is the sum of two individual curves, *A* to *B*, and half is the peak of an individual curve, *B* to *C*. The quintuplex combined curve is the sum of the first three, *A* to *B*, and then two individual curves, *B* to *C*.



FIGS. 15-9 and 15-10. Flow characteristics of a single-acting triplex and of a single-acting quintuplex power pump, respectively. All curves lean in the same direction.

Question 15-25: Is the variation in flow rate during each cycle of primary importance in comparing the characteristics of different power pumps?

Answer: No. Usually the *rate of change* in flow or velocity rather than the *amount of change* is responsible for pressure fluctuation. The maximum rate of change in velocity is proportional to the product of the pump speed and a factor representing the greatest slope of the combined velocity curve. Figures 15-8, 15-9, and 15-10 show that the maximum slope of the double-acting duplex curve is almost twice that of the triplex curve, which in turn is less than twice as steep as the quintuplex curve. For pumps of the different types having the same capacity at the same speed, pressure fluctuation will be in this proportion. This is true except in the unusual case of a pump discharging through a short length of pipe and a nozzle, where the pressure varies as the square of flow variation.

Question 15-26: Will a small high-speed power pump cause less pulsation in pressure than a larger low-speed pump of the same type and capacity?

Answer: No, at least within the speed range of pumps now in use. The small pump at a higher speed has the same total variation in flow as the larger pump, with the difference that it takes place more frequently. Thus the *rate of change* is greater for the small high-speed pump, and in the average system where the inertia of the associated fluid column is important, the fluctuation in pressure will be greater.

Question 15-27: Are all power pumps capable of operating with a suction lift?

Answer: No. Suction requirements vary widely with the design and the service for which the pumps are intended. Piston pumps for low and moderate pressures are usually capable of operating with a suction lift of 10 to 20 feet of water, depending on the speed and relative valve area. Horizontal duplex sidepot pumps are usually offered with a considerable range of liner and piston sizes for each size of pump cylinder. Thus, with a small liner and piston, the valve area is relatively large, and a higher suction lift is permissible. With the largest liner and piston for a given cylinder, the relative valve area is less, and hence the pump must be operated with less suction lift.

Question 15-28: Are all power pumps self-priming with a suction lift?

Answer: No. This depends on the clearance volume and on the required valve loading. Moderate-speed piston pumps for operation at moderate pressures are usually self-priming with some suction lift. If the clearance volume is filled with liquid, the pump should be self-priming at any reasonable operating lift.

Where priming is no problem, plunger pumps for higher pressures are usually designed to operate with flooded suction.

Question 15-29: What types of power pumps require the greatest net positive suction head?

Answer: Speed and pressure influence the suction head required. High-pressure pumps necessarily use heavy valves, and valve and passage areas cannot be made overly large without greatly increasing pump weight and cost. Thus, high-pressure hydraulic pumps usually require considerable suction head, which is provided by bringing all returns to an elevated suction tank. High-speed pumps also require more suction head, unless made with unusually large suction-valve area. High-speed hydraulic or boiler feed pumps may require as much as 15 to 20 psi net positive suction head.

Question 15-30: What piston speed is permissible for power pumps?

Answer: Modern 24-in.-stroke horizontal pumps are run at piston speeds of 300 to 400 ft per min. High-speed vertical pumps of 6- to 10-in. stroke are run at 200 to 250 ft per min. Smaller vertical pumps vary between 100 and 200 ft per min. Permissible piston speed for power pumps is usually two to four times that for direct-acting steam pumps.

Question 15-31: Should every power pump have a discharge relief valve?

Answer: Yes. This cannot be overemphasized. Directly connected to the discharge manifold of every pump should be a relief valve capable of discharging its full capacity at a reasonable pressure increase. For most installations, set the relief valve for a pressure not more than 10 per cent above the maximum operating value. If a power pump is accidentally started against a closed discharge valve, the average electric motor develops sufficient starting torque to build up a pressure three or more times the normal operating pressure. If the discharge valve is closed with the pump running, the inertia of the rotating parts will add to the quick build-up of sufficient pressure to cause serious damage.

Question 15-32: Is a relief valve advisable on the suction side of a power pump?

Answer: While little attention has been paid to the protection of the suction manifold and piping, a small relief valve may prevent considerable damage under certain conditions. There is little danger when a pump is in operation, but during stand-by periods faulty seating of some of the valves in a system may cause a damaging pressure on the suction side of a pump. If, for instance, the discharge stop valve of a stand-by boiler feed pump leaks slightly, and this leakage seeps back through the discharge check valve and the pump valves to a tightly seating suction stop valve—under these conditions, little additional water is required to raise the pressure in an already full system to full boiler feed pressure. The danger of this condition developing may be eliminated by installation of a small relief valve set at a safe pressure for the suction piping and manifold.

CHAPTER 16

LIQUID VALVES FOR RECIPROCATING PUMPS

Question 16-1: What is the operating principle of valves used in the liquid end of reciprocating pumps?

Answer: Valves are self-acting in the fluid end of all direct-acting steam and most power pumps, in contrast to the mechanically operated designs in the steam end of direct-acting pumps. The former are opened by the liquid passing through them, and closed by a spring, plus their weight. Mechanical operation of fluid-end pump valves is not only unnecessary, but impracticable, because of the perfect timing required to avoid shock from fluids that are practically incompressible at normal pressures.

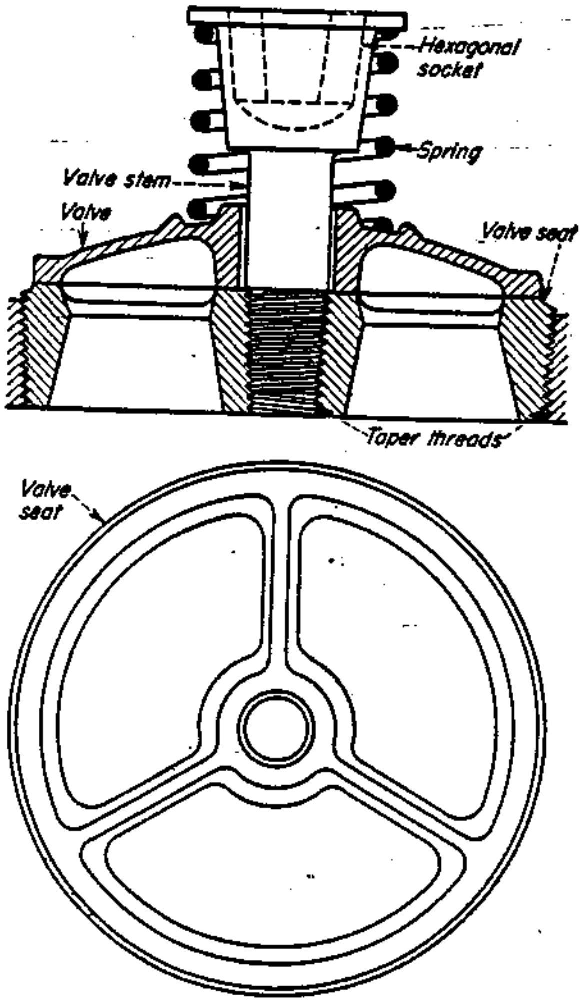


FIG. 16-1. Stem-guided disk valve for low-pressure service.

Question 16-2: What type of valve is used for low pressures?

Answer: The stem-guided disk type (Fig. 16-1). Seat and stem are usually of bronze, as are valves for hot-water boiler feed and medium-pressure general service. For lower pressures, cold-water valve disks are often of rubber. Valve, seat, stem, and spring form a complete assembly, usually screwed into the cylinder casting or valve plate with a tapered thread. At normal speeds, the spring limits the maximum valve lift, but under extreme conditions, the valve may strike the enlarged portion of its stem.

Question 16-3: What type of valve is used for moderate pressures?

Answer: The wing-guided valve (Fig. 16-2), made in both flat-faced and bevel-faced designs. A projection on the valve cover forms a seat for the spring and limits the valve lift under extreme conditions. The seat for a wing-guided valve is a plain ring with the outside taper ground for driving into the bore in the cylinder. Wing-guided valves and their seats are made of cast bronze, or are forged from stainless steel and heat-treated for more severe services.

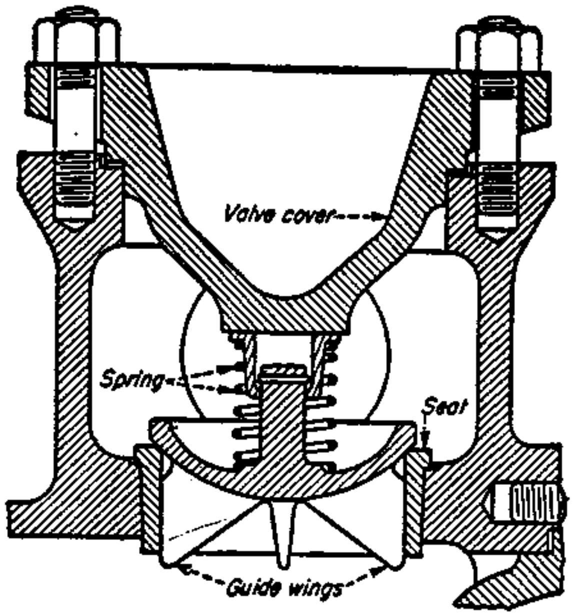


FIG. 16-2. Wing-guided valve for medium pressure may be either flat or bevel-faced.

Question 16-4: What types of valves are used for high-pressure service?

Answer: Bevel-faced wing valves are used most frequently for higher pressures (Fig. 16-3), while ball valves and special seal-type valves are used under some conditions. Hydraulic valves are usually forged from stainless steel and heat-treated, but some designers prefer them

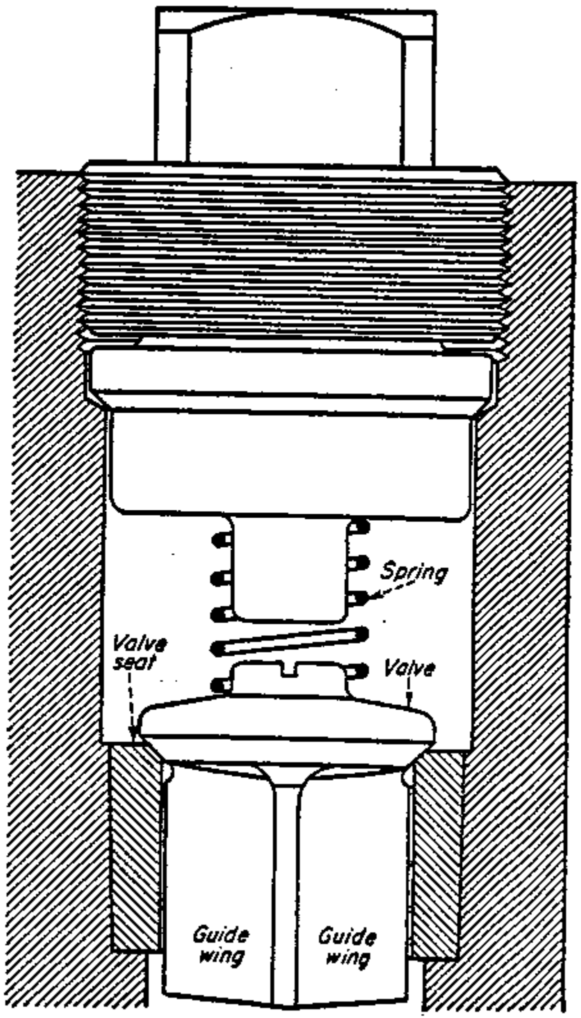


FIG. 16-3. Bevel-faced wing-guided valve for high-pressure service.

cast of hard bronze. They are necessarily quite heavy and in the older slow-speed pumps were often used without springs. Today, common pump speeds require valve springs in spite of the weight of the valve.

Question 16-5: What types of valves are used for viscous liquids?

Answer: Ball or semispherical valves (Fig. 16-4). Both designs have an unobstructed passage through their seats. Semi-

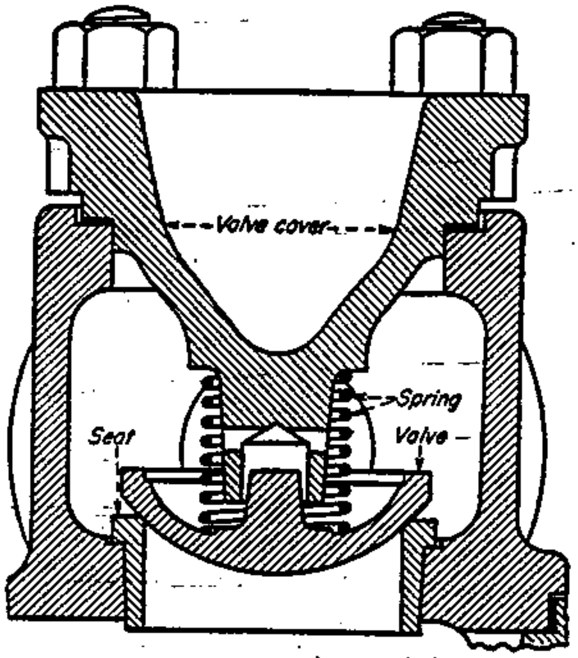


FIG. 16-4.

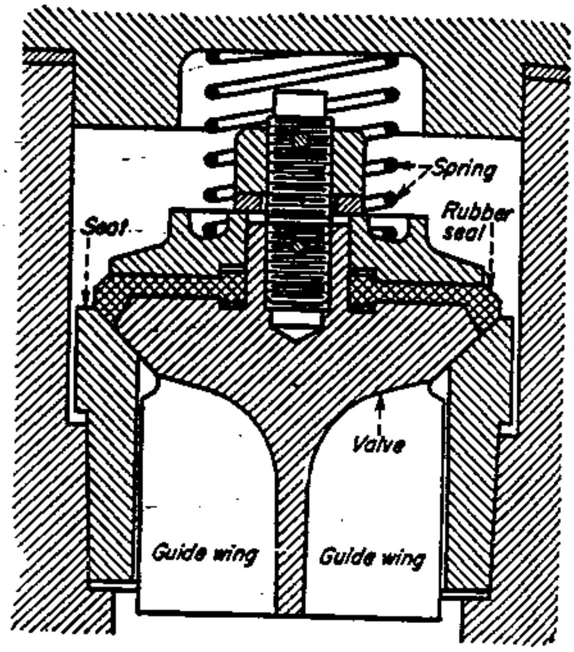


FIG. 16-5.

FIG. 16-4. Semispherical valves permit maximum pump speed with viscous liquids.
FIG. 16-5. Wing-guided sealed valve for high-pressure hydraulic service.

spherical or Rollo valves are better suited to operation at somewhat higher speeds, because, with their lighter weight, a spring can control their action more easily. They also require less space above them, since the fluid passages are not obstructed by the cage needed to retain and guide a ball valve. Spherical-faced valves seat with a line contact on a conical seat, which should be lapped, using as a lap a wing-guided bevel-faced valve rather than the regular valve.

Question 16-6: What is a seal valve?

Answer: Figures 16-5 and 16-6 show two forms. Figure 16-5 is a wing-guided valve for high-pressure hydraulic service, supported by the inner portion of its conical seat. It is sealed against wire-drawing by an oil-resistant synthetic rubber disk, clamped between the valve and its cover. Figure 16-6 shows a valve for a mud or slush pump in oil well drilling. It is sealed by a rubber disk above its body, and is supported by the lower guide-bushing face as well as the seat. Because of the abrasive material pumped, the lower guide is a separate, renewable bushing.

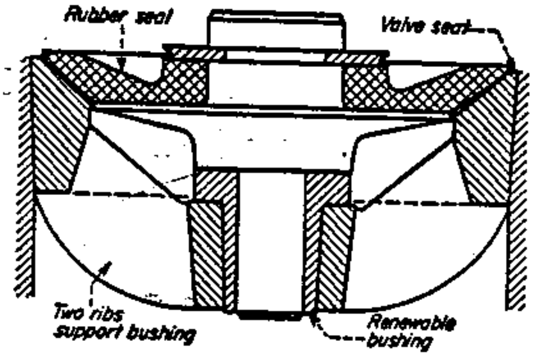


FIG. 16-6. Sealed valve for a slush or mud pump used for oil well drilling. It is sealed by a rubber disk above its body and is supported by a lower guide bushing as well as seat.

Question 16-7: What is meant by valve area?

Answer: This term is usually defined as the free area through the valve seat. Applied to a stem-guided disk valve, this comprises the area between the seat's rim and central boss, less the area occupied by supporting ribs. The area of wings must be subtracted from the area through the wing-guided valve seat. A ball or Rollo valve leaves the area through its seat free of obstructions, and hence provides maximum valve area in a given space.

Question 16-8: Is valve area, as defined above, of primary importance in considering flow through a valve?

Answer: No, because the outlet area between the valve and its seat is determined by the height the valve lifts and the type of seat. With a flat-seated valve, outlet area is the product of lift and valve circumference. The outlet area of a bevel-seated valve at any lift is reduced in proportion to the angle of the bevel.

Question 16-9: How high must a disk valve lift for its outlet area to equal the valve area?

Answer: Somewhat less than one-quarter of diameter D through the valve seat, depending on how much of this opening the ribs or guide wings fill. Neglecting such obstructions, the area through the seat equals $\pi D^2/4$, and the outlet area is $\pi D \times$ valve lift in inches. If valve lift equals one-quarter of D, then outlet area equals $\pi D \times (D/4)$, which is the same as the area through the seat.

Question 16-10: Why are several small valves frequently used instead of a large one having the same valve area?

Answer: As explained in Question 16-9, a valve must lift one-quarter of its diameter for its outlet area to equal the valve area. On larger pumps, the required valve area with reasonable lift can be obtained only with multiple valves. For direct-acting steam pumps, valves are seldom larger than 4½ or 5 in. in diameter. Power pumps, which operate at considerably higher speeds, particularly in smaller sizes, use proportionately smaller valves.

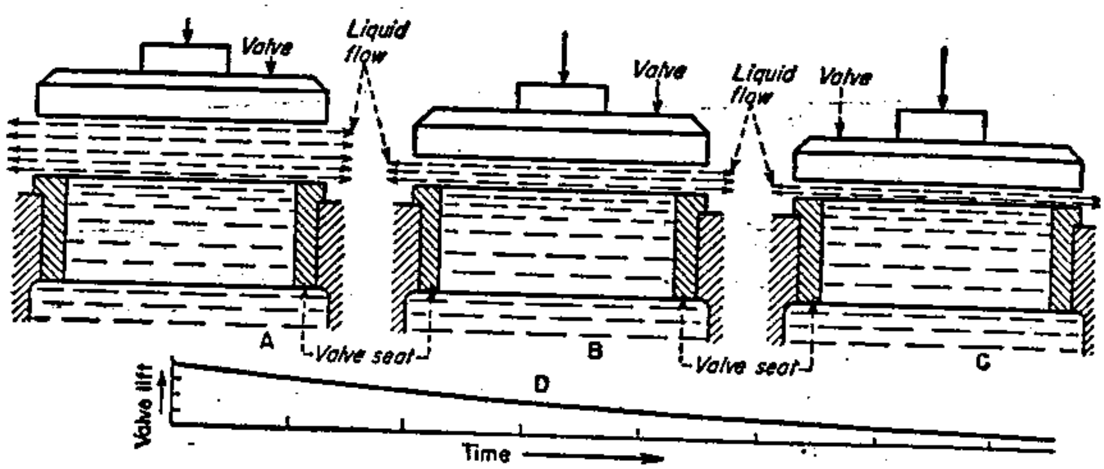


FIG. 16-7. The volume of liquid between a self-acting pump valve and its seat must be displaced through its outlet area to obtain closure without shock.

Question 16-11: What primary factor controls the closing of self-acting pump valves?

Answer: To close without shock, a valve must displace through its outlet area the volume of fluid between it and its seat. Spring loading, valve area, and outlet area determine the rate at which this fluid can be displaced. Consider the discharge valve of a duplex pump at the end of a stroke. Since there is a pause while the pump's other side is making a stroke, the cylinder may be considered as a closed vessel with the valve open above it, as in Fig. 16-7. For each increment of downward movement, the valve must displace a definite amount of fluid through a smaller and smaller outlet area.

Thus, the rate of closing is retarded as the valve approaches its seat as at *B* in Fig. 16-7, where it is half-way down, and at *C*, where it is at one-quarter initial lift. *D* in Fig. 16-7 is the curve of valve lift plotted against time for the valve positions shown. If the initial closing rate is fast enough and the pause long enough, the valve settles on a film of fluid. In practical operation, the piston usually starts its suction stroke before the valve seats, thus providing space in the cylinder for the last of the fluid displaced by the valve, and quickly brings it to its seat.

Question 16-12: Starting at the same lift, can a 4-in. valve be closed as quickly as a 2-in. one?

Answer: Yes, but only by providing about 40 per cent more spring force for each square inch of valve area. Consider both valves starting at ½-in. lift. The 2-in. valve must displace $3.14 \times 1^2 \times \frac{1}{2} = 1.57$ cu in. through an opening that has initially $3.14 \times 2 \times \frac{1}{2} = 3.14$ sq in. of area. The 4-in. valve must displace $3.14 \times 2^2 \times \frac{1}{2} = 6.28$ cu in. through an opening that has initially $3.14 \times 4 \times \frac{1}{2} = 6.28$ sq in. of area. Thus, when a valve size is doubled, the rate of flow through its outlet area must be doubled. To close as quickly, the larger valve must have its lift restricted by a heavier spring. Increased spring tension increases the loss through the valve. That is the reason why one 4-in. valve cannot pass as much fluid as four 2-in. valves, although the valve area of both is the same.

Question 16-13: What is the primary disadvantage of several small valves in place of one larger valve?

Answer: For best performance, all the small valves must open and close simultaneously. This means that the springs for each set of valves must be matched and that the passages to and from the valves must provide a smooth, even flow. If one valve of a set has a stronger spring than the others, it may not open at all. On the other hand, if one spring is weak, a valve may remain open until flow reversal drives it against its seat.

Question 16-14: What is the advantage of a double-ported valve?

Answer: This type of valve (Fig. 16-8) is used in large, relatively high-speed pumps. It provides almost twice the outlet area of a single valve of the same size, and displaces when closing only about half as much fluid. Thus, in a single valve it supplies the characteristics of a multiple installation of much smaller ones. The area through the seat is restricted, but this is offset by the excellent approach formed by the converging passage. A double-ported valve frequently has the disadvantage of high unbalanced area.

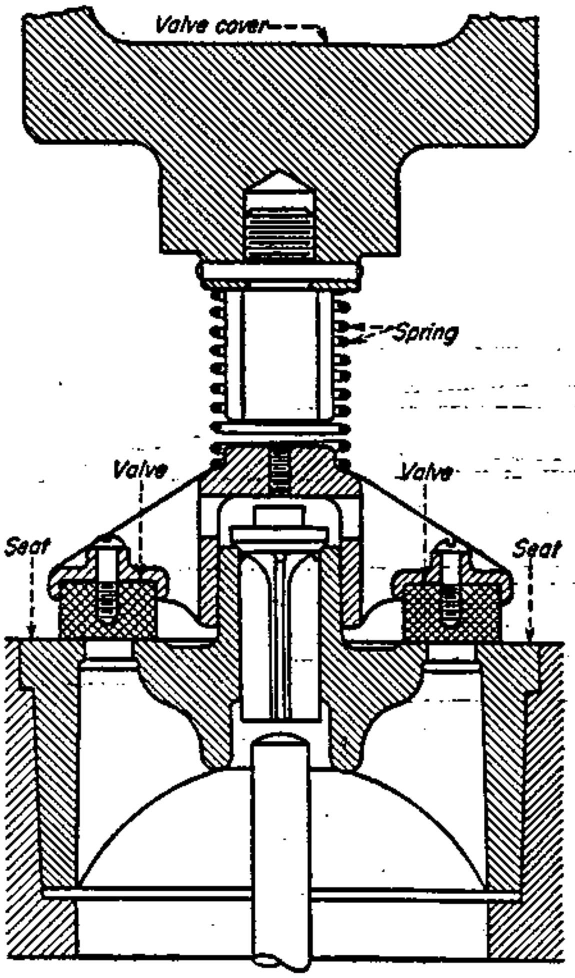


FIG. 16-8. A double-ported valve with a separate bevel-faced valve in the center of seat permitting the use of synchronized suction-valve unloading control.

Question 16-15: What is unbalanced area and how does it affect the valve action of a pump?

Answer: Unbalanced area is the area of the seating surface, or the difference between the area of the valve and the area through the seat. With a properly seating valve, fluid pressure on the seating surface drops to the vapor pressure of the liquid just as the valve begins to open. The valve is held down by discharge or cylinder pressure acting on area A (Fig. 16-9) and is lifted by cylinder or suction pressure acting on area A_1. If the ratio of these two areas is high, an excessive pressure difference is required to lift the valve away from its seat. Thus, the contact area should be kept as small as materials and loading permit. Sometimes wide-seated valves are chamfered to reduce the unbalanced area.

Question 16-16: How does the continuously changing flow rate of a power pump, with no pause at the end of the stroke, affect valve action?

Answer: Power-pump valves open and close gradually throughout the stroke of the piston or plunger. This would be ideal for the valve action if it were not for the effects of valve displacement, particularly in higher speed pumps. Figure 16-10 shows this action. The solid line is the plunger displacement and the dash line the actual rate of flow through the valve, as influenced by valve displacement, which is represented by the shaded area. During the first half of the stroke, the rising valve provides space below it for part of the piston displacement. During the second half of the stroke, valve displacement is added to piston displacement to get the actual flow through the outlet area.

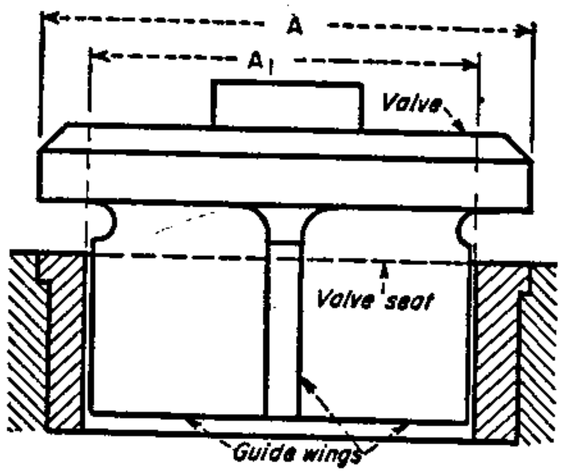


FIG. 16-9. Unbalanced area is the valve area minus area through the valve seat.

The piston reaches the end of its stroke with the valve some distance above the seat and still displacing fluid as it settles. This means that reversal of piston displacement controls the final closing of the valve. Because of this, power-pump valves must have enough spring loading to force them relatively close to their seats at the end of the plunger's stroke. Valve movement lags slightly behind plunger movement, and this lag must be held to a minimum to prevent excessive shock and volumetric loss.

Question 16-17: How much spring pressure should be used on steam-pump liquid valves?

Answer: On suction valves the spring pressure should be equal to ½ to 1 lb per sq in. of valve area for average conditions. This loading may be doubled for unusually high suction heads. A discharge valve is sometimes loaded more heavily than a suction valve, although normally there should be no difference. Where

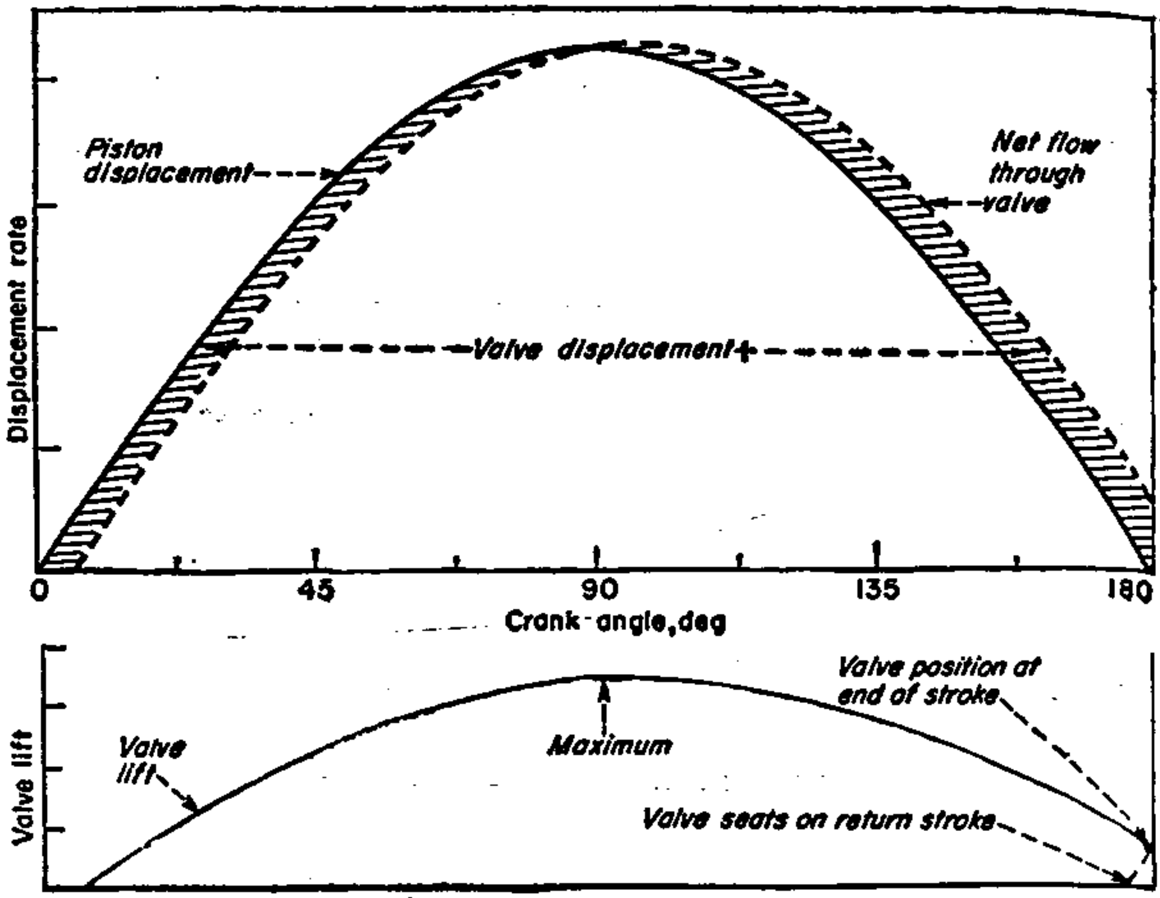


FIG. 16-10. Solid line represents plunger displacement; dash line, actual flow rate through valve of power-driven pump, as influenced by valve displacement.

the valve area is equal in suction and discharge, the only reason for a difference in spring loading is to permit operation at a high suction lift by reducing suction-valve spring tension below that desirable for normal conditions.

Question 16-18: How much spring pressure should be used on power-pump liquid valves?

Answer: Correct spring loading of power-pump valves varies widely with pump design and speed. Slow-speed pumps require about the same spring loading as direct-acting steam pumps. For 100 rpm and higher speeds, spring loading should be increased about in proportion to the speed increase above 50 rpm. This is true in spite of the relatively large valve area of high-speed power pumps. If an existing pump is to run at a higher speed, increase the valve-spring loading proportionally to the speed increase squared. This is necessary because the valves must pass proportionately more fluid and, in addition, must open and close more frequently.

CHAPTER 17

PACKING FOR RECIPROCATING PUMPS

Question 17-1: What is pump packing?

Answer: It is any material used to control leakage between a moving and a stationary pump part, such as between the piston rod and cylinder head, or between the piston and the cylinder liner. Packing is (1) expendable, thus reducing wear on more costly parts, (2) flexible to conform to the shape of adjacent parts, and (3) usually relatively soft. With the exception of piston packing, most packing is installed to permit adjustment for the best balance between leakage and wear of both packing and the parts in contact with it.

Question 17-2: What materials are used for pump packing?

Answer: Packing may be anything from a strip of leather to a cast-iron snap ring. Flax braided and impregnated with tallow, and braided asbestos impregnated with graphite and oil, are common materials for rod and plunger packing. Layers of cotton duck bonded with rubber are commonly used for piston packing for cold water, and special compounds of the same type are used for hot boiler feed water. Canvas impregnated and bonded with bakelite, and sometimes graphitized, is used in the same way. Soft metals, such as lead, babbitt, aluminum, and copper, are used in shredded form, for instance (1) foil wrapped or crumpled around a softer core, together with suitable lubricants, or (2) segmental rings of solid metal, flexibly supported in contact with the rod or plunger. While many different materials may be used, satisfactory performance requires proper selection, installation, and care.

Question 17-3: What is the most important element in good packing performance?

Answer: It is the care with which the packing is handled, installed, and adjusted. Relatively poor inexpensive packing may give excellent service if properly handled, while carelessness may quickly spoil the best material. Cleanliness, a first consideration, applies to both the stuffing box and any surface used in cutting or forming the packing. A little grit on a new packing ring may seriously damage a rod or cylinder liner. Second, packing must be properly fitted, with allowance for expansion under existing conditions. This applies to both the initial fitting and running adjustment. Some leakage is needed for lubrication and cooling. An overly tight packing runs hot.

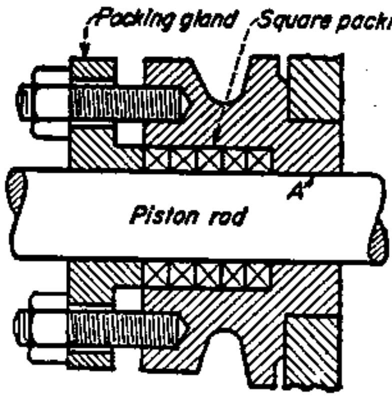


FIG. 17-1.

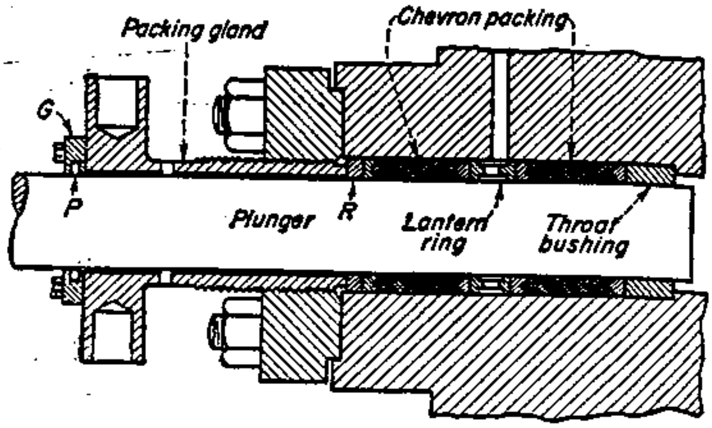


FIG. 17-2.

FIG. 17-1. Simple piston-rod stuffing box with square packing.
FIG. 17-2. Stuffing box and chevron packing used on the plunger of a high-pressure hydraulic pump.

Question 17-4: What are the essential parts of a piston-rod stuffing box?

Answer: Figure 17-1 shows a simple piston-rod stuffing box, consisting of a cavity to receive the packing, a close fit around the rod at A to prevent the packing from extruding into the cylinder, and an adjustable gland to close the outer end of the box. On small rods, a single nut surrounding the gland is often used instead of studs. Most stuffing boxes are fitted with a replaceable bushing, usually of bronze, at the stuffing-box base.

Question 17-5: What type of stuffing box is used for the plunger of high-pressure hydraulic pumps?

Answer: Figure 17-2 shows such a box. The gland threads into a heavy flange, which is centered by a counterbore in the cylinder and bolted solidly in place. A separate metal packing ring *R*, closely fitting the plunger and stuffing-box bore, is used between gland and packing. This construction provides ample clearance between gland and plunger, and permits ready adjustment without danger of scoring the plunger by unequal tightening of two or more gland nuts. Gland and flange are removed as a unit when repacking the stuffing box.

Question 17-6: What is a lantern ring or gland?

Answer: The relieved metal ring at the center of the packing space (Fig. 17-2) is known as a lantern ring or gland. Holes through its center section permit free circulation of fluid or lubricant around the plunger and to the packing on either side of the ring. On vacuum pumps, this ring permits the use of sealing fluid to prevent the entry of air. On high-pressure pumps, it distributes grease to the packing around the plunger.

Question 17-7: What is a wiper gland?

Answer: The main gland (Fig. 17-2) has a light auxiliary gland *G* with a small packing space *P* to control leakage flow from the high-pressure packing. The wiper keeps the high-pressure packing leakage from traveling along the plunger and dripping on the crosshead guide. Auxiliary glands are used also when it is necessary to inject water or oil to absorb and dispose of leakage of volatile or inflammable fluids, such as light hydrocarbons.

Question 17-8: What are the major differences between a piston packing and a rod or plunger packing?

Answer: Figure 17-3 shows a simple piston with soft, fibrous packing. Clearance *C* between packing and packing space permits fluid pressure to act on one end and the inside of the packing to hold it against and seal the other end of the space and the cylinder bore. No adjustment is possible when the pump operates, nor is it required, since all wear takes place on the packing's outside face.

In a rod or plunger stuffing box, the adjustable gland cannot seal the outer end of the stuffing-box bore; hence the packing must fill the space and seal along both its inner and outer surfaces. While some packings, when first installed, may move in the stuffing box to maintain contact with the gland, it is usually necessary to compensate for packing wear by tightening the gland to press the packing against the rod or plunger.

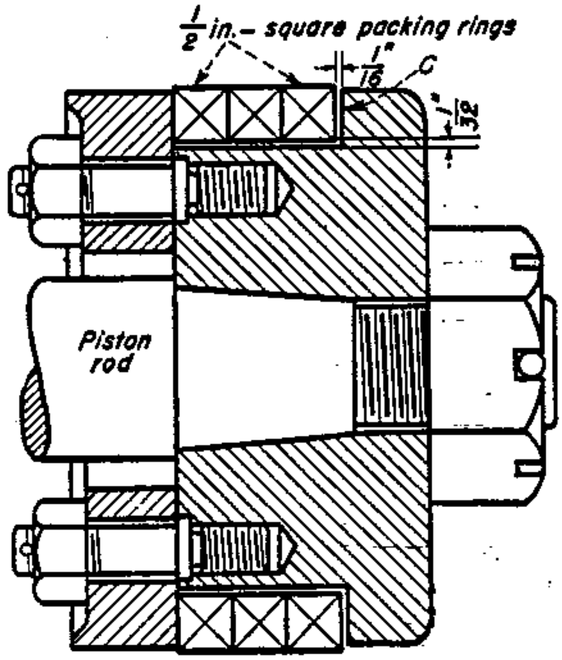


FIG. 17-3.

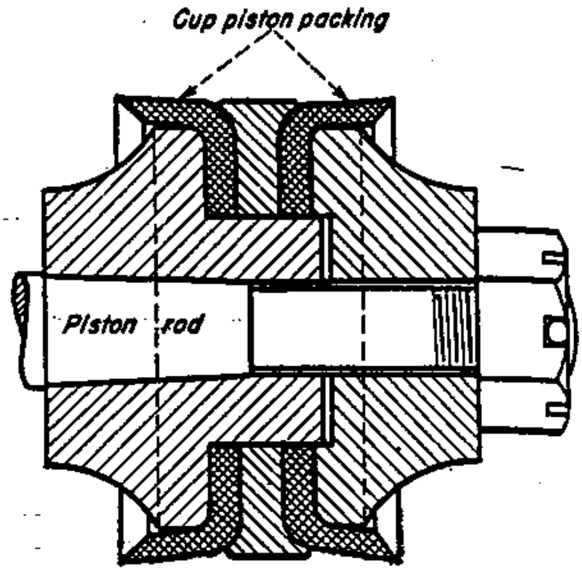


FIG. 17-4.

FIG. 17-3. Pump piston packed with three rings of soft fibrous packing.
FIG. 17-4. Self-sealing cup packing on a double-acting piston.

Question 17-9: How does a cup packing (Fig. 17-4) seal a pump piston?

Answer: Fluid pressure on the inside of the cup presses the lip out against the cylinder bore, forming a tight seal with a small area of contact. On a double-acting piston, two cups are placed back-to-back (Fig. 17-4), and the leading cup seals in each direction of motion. Cup piston packing is one of the oldest and simplest forms. Until recently it has had relatively little use in pumps, except at low pressure, because of the concentration of wear at the heel of the cup, which made its life too short. Development of rubber-compound and fabric cups capable of long life under severe conditions has led to more general acceptance of this form of packing.

Question 17-10: How does a relatively soft, fibrous packing seal the fluid piston of a pump?

Answer: As mentioned in Question 17-8, fluid pressure on one end and on the inside of the packing holds the other end in contact with the piston and the outside in contact with the cylinder bore. This pressure also holds individual rings in sealing contact with each other, and there is a sharp drop in fluid pressure between the packing and the cylinder bore, starting at the first packing ring. On reversal, the packing transfers its sealing contact to the piston's other end. Such packing must be fitted with enough clearance to permit this floating action, and individual rings must have end clearance to prevent jamming and interference.

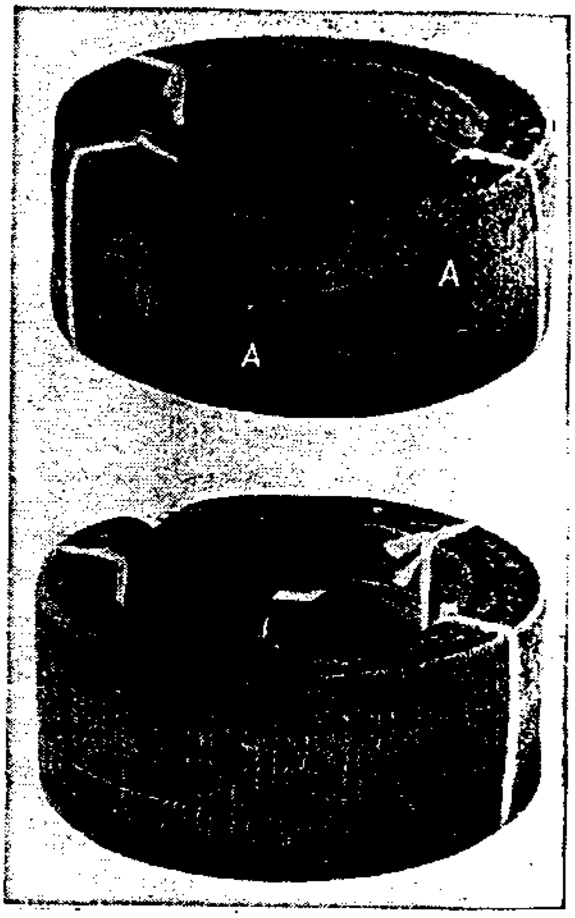


FIG. 17-5. Packing removed from service showing damage from excessive swelling.

If the piston and follower clamp the packing when first installed or the packing swells so that its movement is restricted, slight wear permits excessive leakage, or excessive friction causes short strokings and other irregularities. Figure 17-5 shows a set of packing in which swelling caused excessive friction and severe liner wear. Operation in the resulting oversize bore opened the gap in each ring, permitting the adjacent rings to swell into the space thus formed, as at *A*.

Question 17-11: How does the action of a relatively hard packing on a piston differ from that of a soft, fibrous one?

Answer: A relatively hard packing, such as the bakelite-impregnated canvas used extensively for boiler feed pumps at pressures and temperatures too great for soft packings, often fails to seal between the individual rings when multiple rings are used in a single groove. This causes rapid wear of the two end rings while the intermediate rings are doing no work. Figure 17-6 shows a piston for bakelite packing in which the pressure is broken down gradually from groove to groove and wear is divided between all the packing rings.

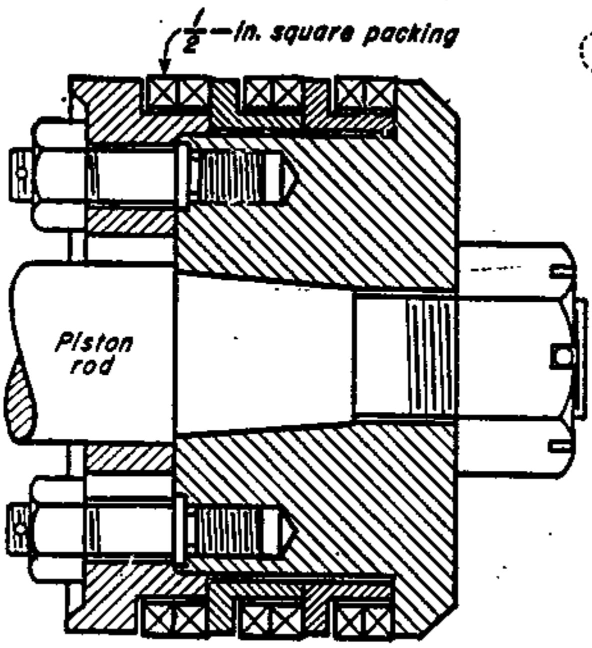


Fig. 17-6.

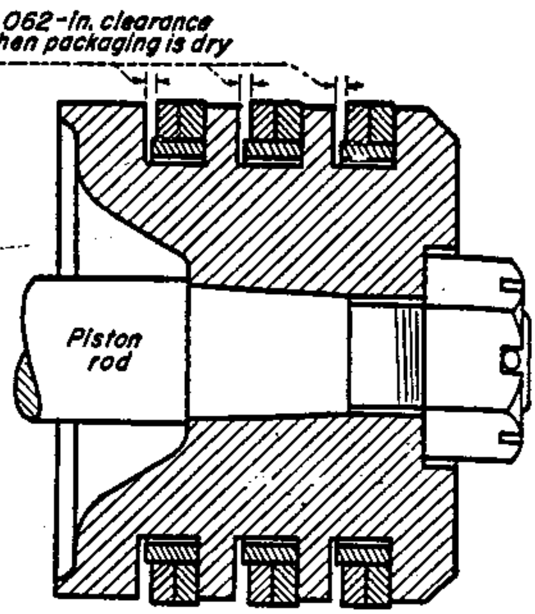


Fig. 17-7.

Fig. 17-6. In piston with bakelite packing, pressure is broken down gradually from groove to groove.
Fig. 17-7. Piston for relatively high pressure packed with double-width sealing ring behind each pair of wearing rings.

Question 17-12: What is the function of a sealing ring placed behind the wearing rings of a piston packing assembly?

Answer: Packing rings must always be installed with enough gap between their ends to keep them from butting tightly together when the packing swells in service. Unless some means is used to seal this gap, it permits leakage from behind the last ring in each groove. Figure 17-7 shows a piston for relatively high-pressure packed with a double-width sealing ring behind each pair of wearing rings to seal this gap.

Question 17-13: What two basic types of packing rings are used on both pistons and rods or plungers?

Answer: They are the square packing rings (Figs. 17-1 and 17-3) and self-sealing rings or cups (Figs. 17-2 and 17-4). In the first type, fluid pressure on the end or inside of the ring holds the entire face of the packing in sealing contact. In the second, fluid pressure in the flexible lip acts to maintain contact.

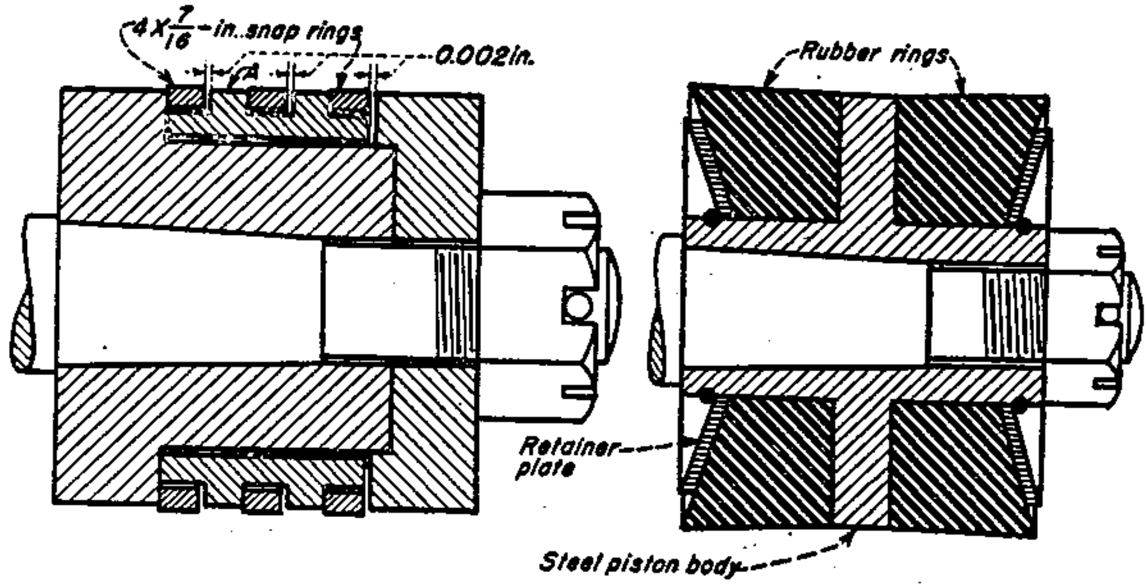


FIG. 17-8. FIG. 17-9.

FIG. 17-8. All-metal piston packing for oil pumps.
FIG. 17-9. Slush-pump piston packing with solid rubber rings.

Question 17-14: What is a bull and snap-ring packing, and when is it used?

Answer: It is an all-metal piston packing used on oil pumps; Fig. 17-8 shows a common form. Spacer *A* is a solid metal ring, which fills most of the packing space of an ordinary follower-type piston. The snap rings are conventional iron or bronze piston rings with an angle or step-cut joint. Their natural tension keeps them in contact with the cylinder liner, assisted by the fluid pressure under the ring. A tighter packing of this type may be obtained by using a two- or three-piece compressor ring in each groove.

Question 17-15: What type of packing is used on the pistons of mud pumps for oil well drilling?

Answer: Usually a solid rubber packing of the flexible-lip type is used for this severe service. The heavy rubber disks, which make up most of the piston, may be clamped between retainer plates on either side of the piston body (Fig. 17-9) or may be bonded to the piston body.

Question 17-16: What type of stuffing box is used for high-temperature pumps?

Answer: Figure 17-10 shows a stuffing box for temperatures of 500°F and higher. The stuffing box and extension are jacketed for

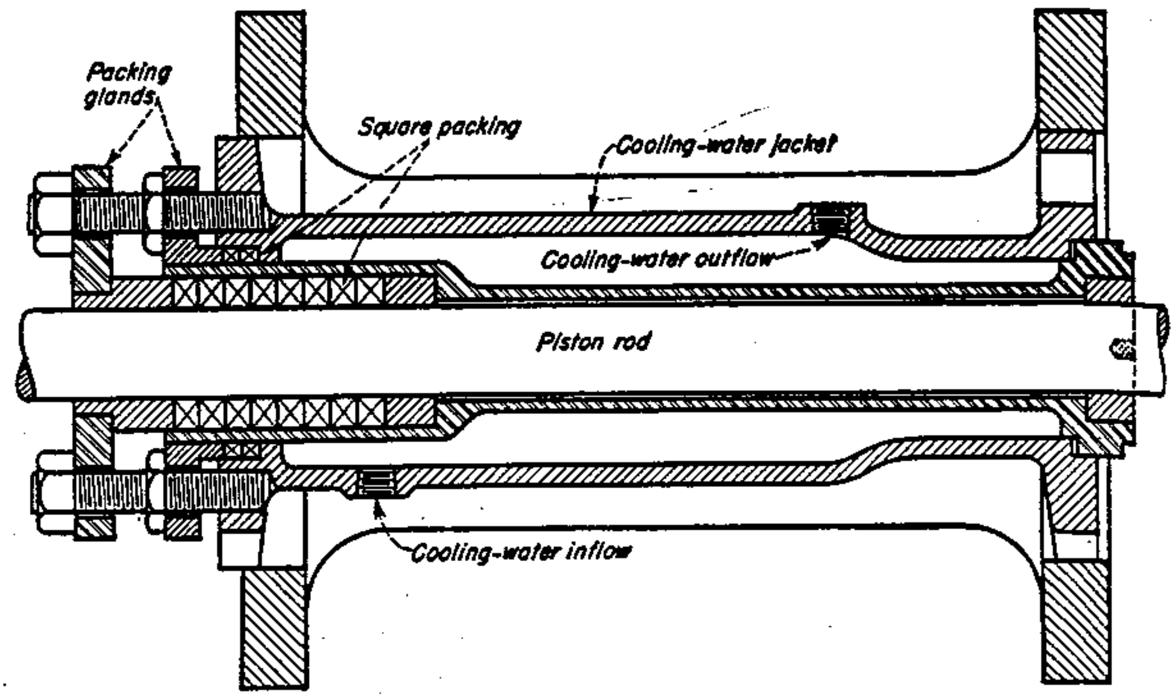


FIG. 17-10. Jacketed stuffing box for pumps handling 500°F and higher temperature liquids with extension so section of rod passing through packing does not project into cylinder.

water cooling, and the piston-rod section, which comes in contact with the packing, does not travel beyond the inclosing sleeve and hence does not reach the hot liquid. Large plunger pumps for high-temperature oil have, in addition, hollow plungers, inside of which cooling water is circulated.

Question 17-17: What procedure should be followed in packing a deep stuffing box for high pressure?

Answer: For self-sealing chevron packing, each ring should be snugly seated before adding the next one. But do not pull the gland hard against the packing, because it depends on the free action of each lip and must not be solidly compressed. Square packing requires the opposite treatment. Each ring is compressed into firm contact with the plunger and stuffing-box bore by split spacers and the plunger gland before adding the next ring. After the last ring is in place, back off the gland slightly to permit normal leakage and expansion under running heat. Heavy gland pressure cannot compensate for loosely installed rings at the bottom of a deep stuffing box. Instead, a gland that is too tight will soon destroy the first few rings of packing and damage the plunger.

CHAPTER 18

CUSHION CHAMBERS FOR RECIPROCATING PUMPS

Question 18-1: What is an air chamber?

Answer: Figure 18-1 shows a common form of air chamber used on the discharge of a double-acting duplex pipe-line pump. Such a chamber is partly filled with air or gas, which contracts and expands to absorb most of the pressure changes caused by variation in flow from a reciprocating pump.

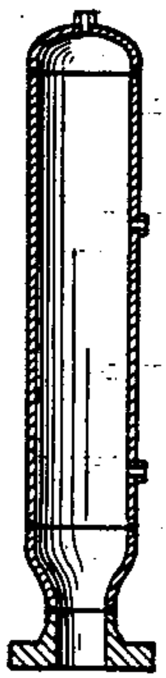


FIG. 18-1. Common form of cushion chamber used on discharge of double-acting duplex pipe-line pump.

Question 18-2: What is a vacuum chamber?

Answer: It has long been customary to speak of a cushion chamber on the discharge side of a pump as an "air chamber" and of a similar chamber on the suction side as a "vacuum chamber." In the writers' opinion, it would be far less confusing and more specific to use the terms "discharge cushion chamber" and "suction cushion chamber."

Question 18-3: How should cushion chambers be connected to a pump?

Answer: A cushion chamber is far more effective if connected so that flow is directly into the chamber. Figures 18-2 to 18-4 show three methods of connecting a suction cushion chamber. Where the pump suction chest has two opposite connections, best results are obtained with the cushion chamber opposite the suction pipe (Fig. 18-2). For pumps with a single suction opening, Fig. 18-3 is almost as effective. Figure 18-4 shows the wrong way to connect a suction cushion chamber. The same rules apply when con-

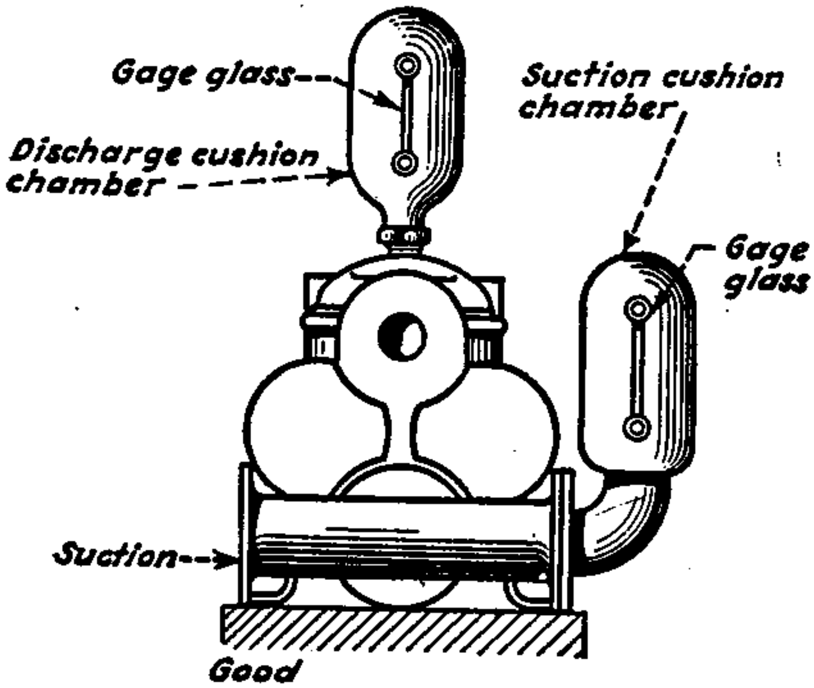


FIG. 18-2.

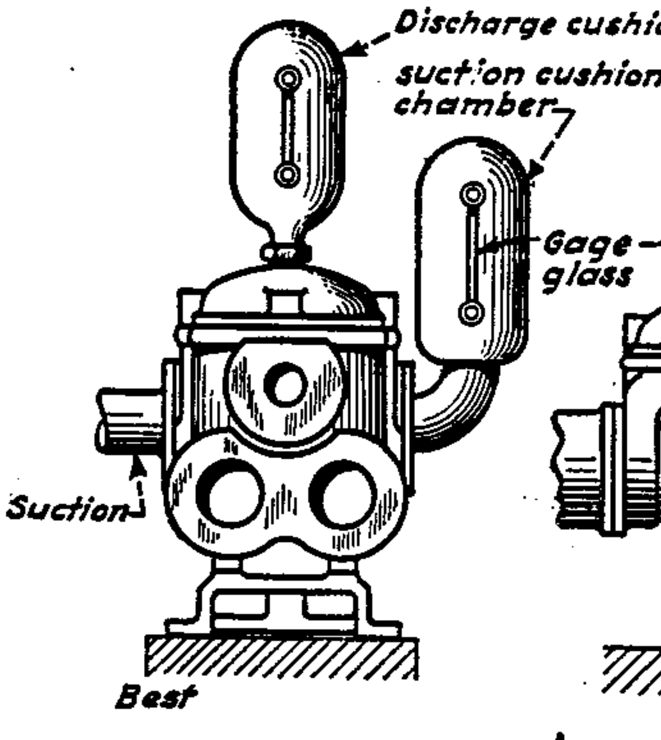


FIG. 18-3.

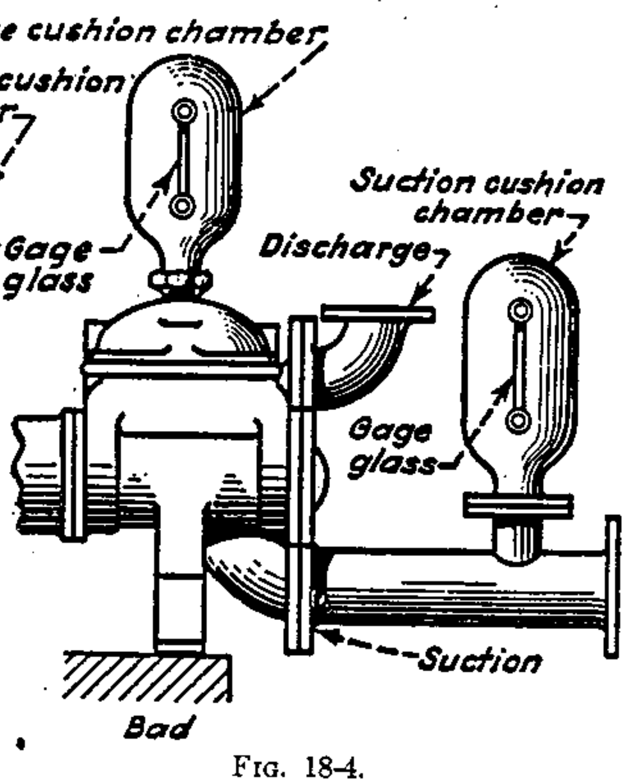


FIG. 18-4.

FIGS. 18-2 to 18-4. Three methods of connecting a cushion chamber to the suction of a double-acting duplex reciprocating pump.

necting a discharge cushion chamber, although the difference in results is not so great as for a suction chamber.

Question 18-4: Why are cushion chambers desirable in connection with reciprocating pumps?

Answer: As brought out in discussion elsewhere, the flow rate of every reciprocating pump, either direct-acting steam or power, goes through a periodic variation. Unless it is practical to use cushion chambers, the entire fluid column, both suction and discharge, must follow this flow pattern. With cushion chambers, flow in the piping is more or less constant, since variation at the pump is absorbed by flow in and out of the chambers.

Question 18-5: When should cushion chambers be used?

Answer: Use them when the care required to keep them charged brings a worth-while return in smoother pump operation and pipe flow. An exception is where the cushioning medium will contaminate the fluid pumped, as would occur with an air-charged chamber on a pump handling deaerated boiler feed water. On reciprocating pipe line and oil well drilling pumps, relatively elaborate apparatus is justified to charge and retain the charge in discharge cushion chambers.

Question 18-6: Why do cushion chambers require charging?

Answer: The reason is that air or gas dissolves into fluids commonly pumped, at a rate proportional to pressure in the chamber. Under extreme conditions, it may be necessary to charge a discharge cushion chamber every hour. Conversely, under a high suction lift it may be necessary to remove accumulated air from a suction cushion chamber to maintain good volumetric efficiency.

Question 18-7: Should cushion chambers have a gauge glass to show the liquid level?

Answer: Yes, install one where there is any question of the amount of air or gas in the chamber. A waterlogged cushion chamber is worse than none at all, because it gives the false impression of helping the installation. A gauge glass shows both the amount of cushion and the effectiveness of the chamber-charging device.

Question 18-8: Which should be used oftener, a suction or discharge cushion chamber?

Answer: Generally a suction cushion chamber is more important than a discharge cushion chamber. On the suction side of a pump, a relatively small pressure fluctuation may momentarily reduce pressure to the point where the fluid vaporizes. Severe pounding results when this vapor condenses under increased pressure. A similar or considerably greater pressure fluctuation on the pump's discharge side may have no noticeable effect. On direct-acting steam pumps, a discharge cushion chamber alone may cause disturbances that can be eliminated only by installing a suction cushion chamber. Another factor strongly in favor of a suction cushion chamber is that its effective operation is easily maintained.

Question 18-9: How are cushion chambers charged?

Answer: A suction cushion chamber on a pump operating with a suction lift frequently accumulates air released from the water because of reduced pressure. This condition provides a self-regulating cushion. Where the amount of air released is excessive, it may be necessary to use a vacuum pump or an ejector to remove air from the chamber. Where pressure in a discharge or suction chamber is greater than atmospheric, the best means of charging is with compressed air or gas. Where air is not available at a suitable pressure, snifter valves or a charging device (Fig. 18-5) may be used. Air enters cylinder P through check valve C during the suction stroke. Then the discharge stroke forces this air into the cushion chamber through check valve C_1. Valves V and V_1 are used to regulate the amount of air handled by each pump stroke.

Question 18-10: How large a cushion chamber does a direct-acting steam pump need?

Answer: The best size for any installation can be determined only by trial, because at reversal the action of a steam pump is changed somewhat by a cushion chamber. Experience shows that results are usually good on a simplex pump with a chamber volume six to eight times the displacement of a single stroke. The cushion-chamber volume of a duplex pump is usually three to four times the single stroke displacement. Sometimes a larger or smaller chamber may be necessary for either a simplex or a duplex steam pump.

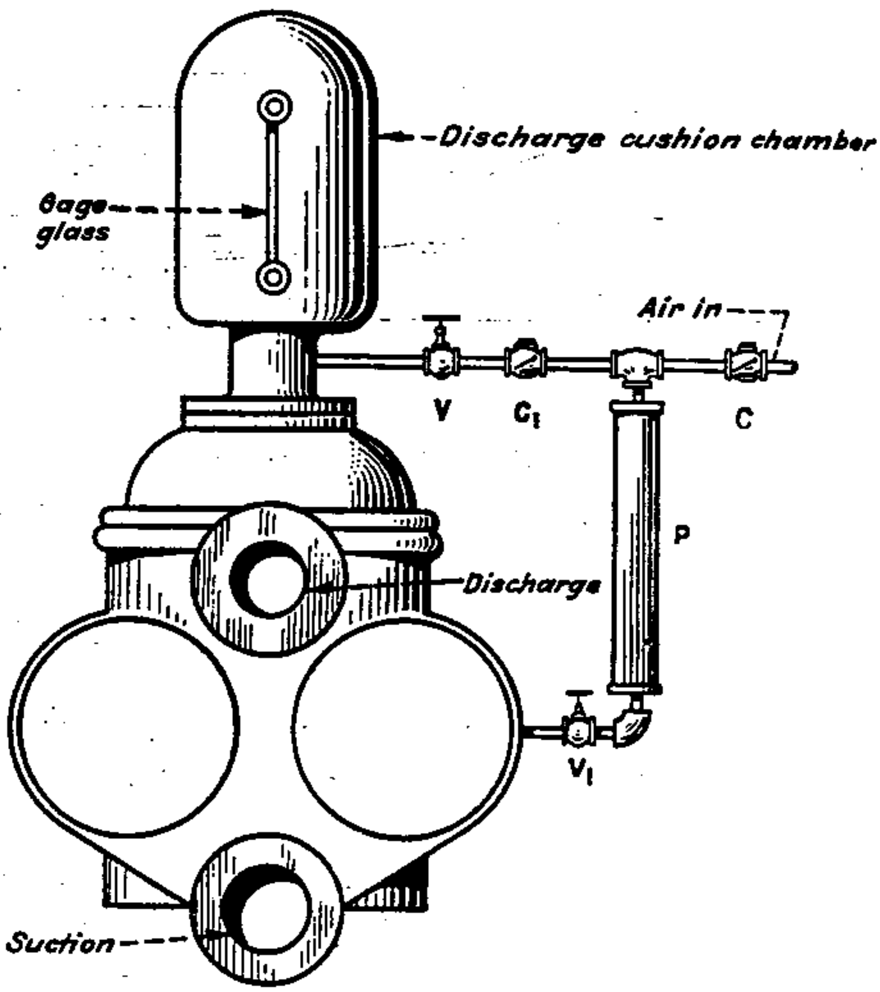


FIG. 18-5. Snifter-valve arrangement for charging air into discharge cushion chamber of duplex reciprocating pump.

Question 18-11: What size cushion chamber does a double-acting duplex power pump need?

Answer: This pump goes through a complete cycle once each revolution, and the total flow to and from a cushion chamber is about 3.5 per cent of the volume delivered during a complete revolution. If the air volume in the chamber is 30 to 40 times the amount absorbed, or 1 to 1.5 times the displacement of one pump revolution, pressure fluctuation should not exceed 3 to 5 per cent of the average pressure. A larger air volume provides a proportionate reduction in pressure fluctuation.

Question 18-12: What size cushion chamber is required for a triplex power pump?

Answer: Triplex pumps, both single- and double-acting, have three cycles per revolution, and flow to and from the cushion chamber during each cycle is slightly over 1 per cent of the pump displacement per revolution. For slow-speed pumps, this makes it possible to use a relatively small cushion chamber, but since modern triplex pumps operate at higher speeds, the frequency with which a small quantity of fluid must enter and leave the chamber requires consideration. For pump speeds in excess of 100 rpm, make the air volume 100 to 150 times the flow variation, or 1 to 1.5 times the pump displacement per revolution, thus making the change in air pressure negligible. Also, instead of necking down the lower end of the chamber and using a reduced-size connection to the pump, as is customary for direct-acting and other low-speed pumps, the connection to the valve chest of the pump should be as short and as large as is practicable.

Question 18-13: What size cushion chamber does a quintuplex power pump require?

Answer: On both single- and double-acting pumps of this type, the actual volume flowing to and from a cushion chamber at each cycle is too small to use as a basis for calculating chamber size. If the chamber is large enough to hold its charge a reasonable length of time, air-pressure variation will be negligible. On the other hand, the higher speed of quintuplex pumps and the 5 cycles per revolution make it doubly important to use a short, maximum-diameter connection to the valve chest as mentioned in Question 18-12.

Question 18-14: How much difference does a short, full-sized connection to a cushion chamber make on a moderate-speed power pump?

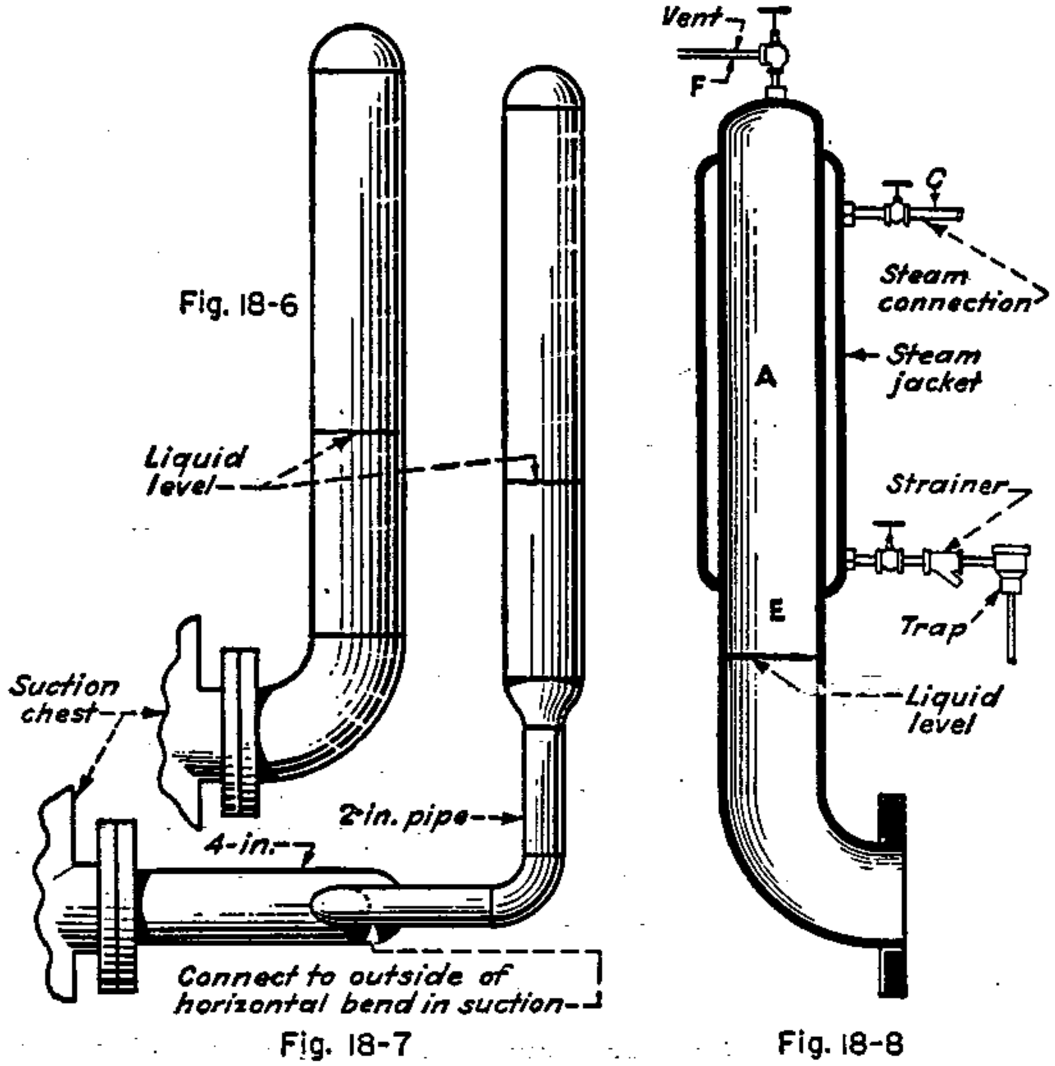


FIG. 18-6. Proper method of connecting suction cushion chamber to high-speed power pump.
FIG. 18-7. Less effective method of connecting cushion chamber.
FIG. 18-8. A steam-jacketed suction cushion chamber for reciprocating boiler-feed pumps. Cushion space contains steam at suction pressure.

Answer: Figures 18-6 and 18-7 show suction cushion chambers as arranged by different designers for the same triplex boiler-feed-pump installation. Figure 18-6 provides a cushion volume of about 2.5 times the pump displacement per revolution, directly connected to the pump suction chest by a long-radius 5-in. elbow and about 9 in. of 5-in. pipe. In Figure 18-7, cushion-chamber volume is 1.5 times the pump displacement, and the chamber is connected to the 4-in. suction line about 1 ft from the suction chest by 2 ft of 2-in. pipe and 1 ft of 4-in. pipe. Both chambers are large enough to absorb the flow variation with very little pressure change in the cushion space. But the connection and mounting (Fig. 18-6) reduce pressure fluctuation in the pump chest to about 1.5 ft of water, while Fig. 18-7 will reduce pressure fluctuation to only 12 to 15 ft of water. This shows that a well-designed installation may be ten times as effective as a poor one.

Question 18-15: Because an air chamber seriously contaminates deaerated feed water, how can the suction of a boiler feed pump be cushioned?

Answer: Figure 18-8 shows the Worthington steam-jacketed suction cushion chamber for reciprocating boiler feed pumps. Cushion space *A* contains steam at a suction pressure produced by evaporating a little of the feed water in the chamber. There is no connection between cushion space *A* and the steam jacket. The latter is supplied with saturated steam through connection *C* at a pressure from 10 to 100 or 150 psi greater than the pump suction pressure. As the jacket steam condenses, it drains through the strainer and trap.

When the jacket steam is first turned on, heat transfer through the chamber wall to the feed water is rapid, and enough feed water evaporates to force the water level quickly down to *E*. Stabilization of the water level at this point is positive since heat transfer from the cushion steam to the water, a few inches below the steam jacket, is equal only to the heat loss from the chamber. Insulate the chamber to prevent unnecessary heat loss. A vent may be provided at *F* to bleed off air before putting the system in operation.

Question 18-16: What are the advantages of a steam-jacketed cushion chamber?

Answer:

1. The pumped fluid is not contaminated.
2. The chamber requires no attention or recharging. The liquid level remains practically constant over a wide pressure range

in the cushion space and steam jacket as long as the jacket pressure is higher than the pump suction pressure.

3. Very little heating steam is required. A chamber like Fig. 18-8 with a cushion space of 2 to 3 gal uses only 3 to 4 lb of steam per hour. The only problem is to find a steam trap that consistently handles this small quantity of condensate without blowing steam.
4. The liquid level may be placed close to the pump suction for most effective cushioning, since it changes little during a gradual pressure change. Even with a sharp reduction in suction pressure, the amount of steam that may expand into the suction chest will condense in the suction stream without disturbing pump operation.

Question 18-17: Is a steam-jacketed cushion chamber practical for the discharge side of a boiler feed pump?

Answer: No, since boiler feed pump discharge pressure is necessarily higher than the pressure of any available saturated steam. It is possible to obtain the higher temperature required by using superheated steam, but the advantages of self-regulation and automatic operation are lost. Also, the higher temperature and poor heat transfer from dry superheated steam to the outside surface of the cushion chamber make it relatively difficult to design a chamber that works, even with careful and more or less continuous hand regulation.

CHAPTER 19

OPERATION OF RECIPROCATING PUMPS

Question 19-1: What is the most important consideration in the installation of any pump?

Answer: No matter how heavy and rigid the pump, it may be seriously distorted when tightening its foundation bolts. To obtain satisfactory service, every machine must be firmly and evenly supported. A suitable foundation for a small direct-acting steam pump may be a firm wooden floor. But a heavy foundation of reinforced concrete is required for a large rolling-mill-frame pump. Foundation details are subject to adjustment for local conditions, such as soil and the proximity of other machinery.

Question 19-2: What factors should be considered in selecting the location for a pump?

Answer: The following factors should be considered:

1. The pump should be placed as near the source of supply as practical, but do not elevate a unit to get it close to an overhead supply as that would reduce the suction head.
2. The pump should be accessible for cleaning, regulating, adjustment, and repairs, with sufficient space for the removal of piston rods and other parts.
3. Operators will generally take better care of a pump if it is installed in a clean, dry pump room of adequate size.

Question 19-3: Should a horizontal direct-acting steam pump be firmly anchored at both ends?

Answer: No, because it must be free to expand under operating temperatures. Usually the liquid-cylinder support should be firmly bolted to the foundation and the steam-cylinder support provided with a sole plate (Fig. 19-1) on which it may slide with hold-down bolts, left loose enough to permit free longitudinal movement.

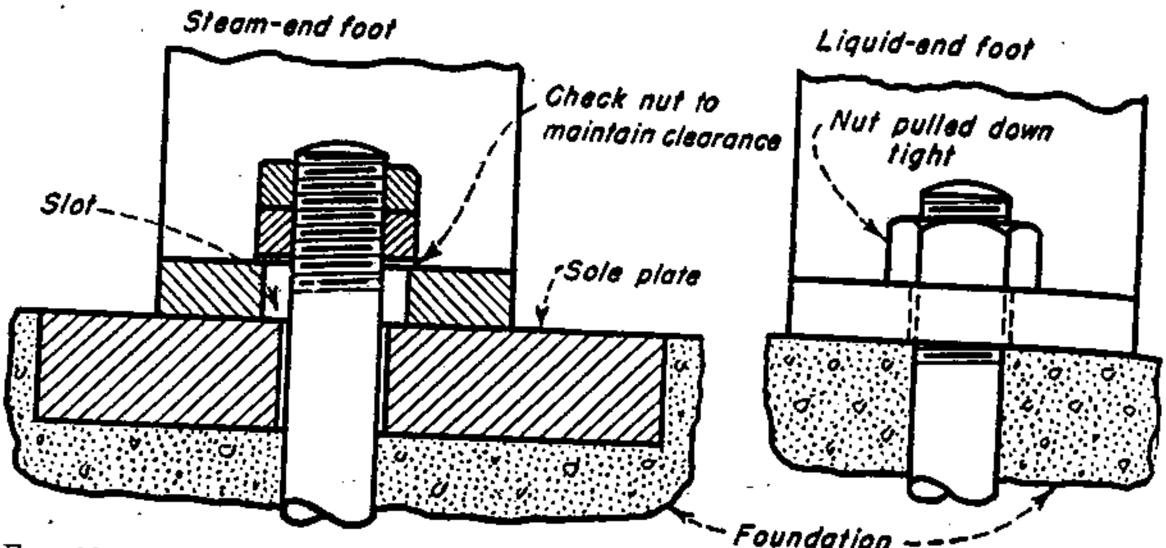


FIG. 19-1. Method of fastening the feet of steam and liquid cylinders of a horizontal direct-acting steam pump to its foundation.

Question 19-4: How are vertical direct-acting steam pumps supported?

Answer: The primary support of all vertical pumps is at the liquid end. The lower end of the liquid cylinder is made with an integral flange or brackets for support by a suitable foundation

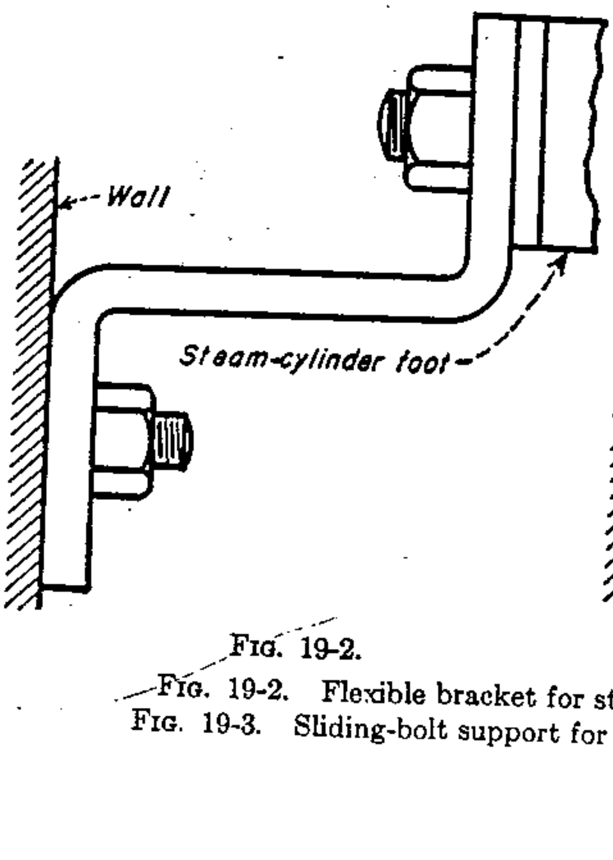


FIG. 19-2.

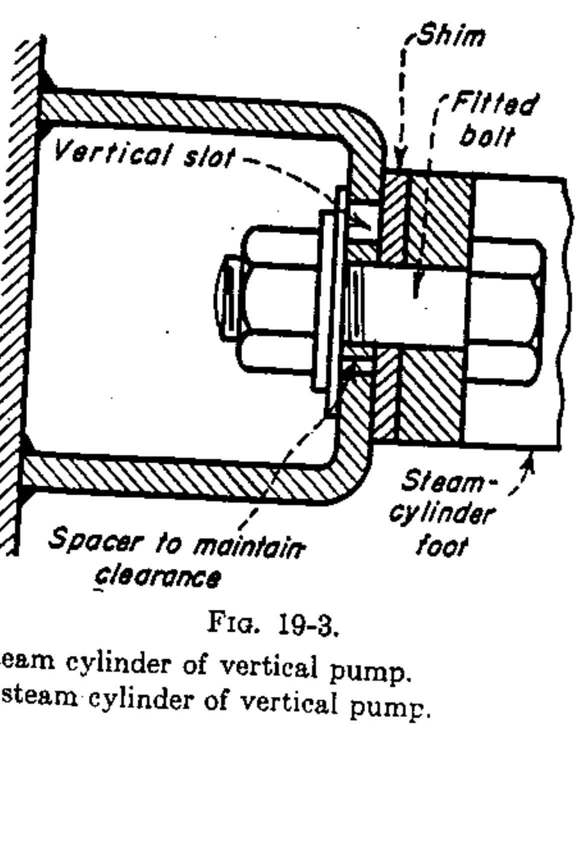


FIG. 19-3.

FIG. 19-2. Flexible bracket for steam cylinder of vertical pump.
FIG. 19-3. Sliding-bolt support for steam cylinder of vertical pump.

(Fig. 19-4). Direct-acting boiler feed pumps are often made with integral brackets at the upper end of the liquid cylinder, at about the same height as the suction and discharge piping (Fig. 19-5). Such a design leaves most of this cylinder free to expand downward when heated by the boiler feed water. The bracket at the steam-cylinder end of vertical pumps carries no weight and is arranged merely as a sway brace. Vertically flexible brackets (Fig. 19-2) or sliding bolts (Fig. 19-3) are used to permit free expansion.

Question 19-5: Are pump flanges heavy enough to support connecting piping?

Answer: While most pump flanges support considerable weight in addition to the hydraulic load, it is poor practice to hang the pipe on the pump. Firmly support liquid piping as close to the cylinder as practicable, to prevent transmission of forces from piping to pump. Support steam piping a little farther away from the cylinder, to permit free expansion of the cylinder and still isolate it from forces caused by the expansion of steam and exhaust piping. Where practicable, check all piping for alignment by loosening the joints at the pump when nearly normal operating temperature exists.

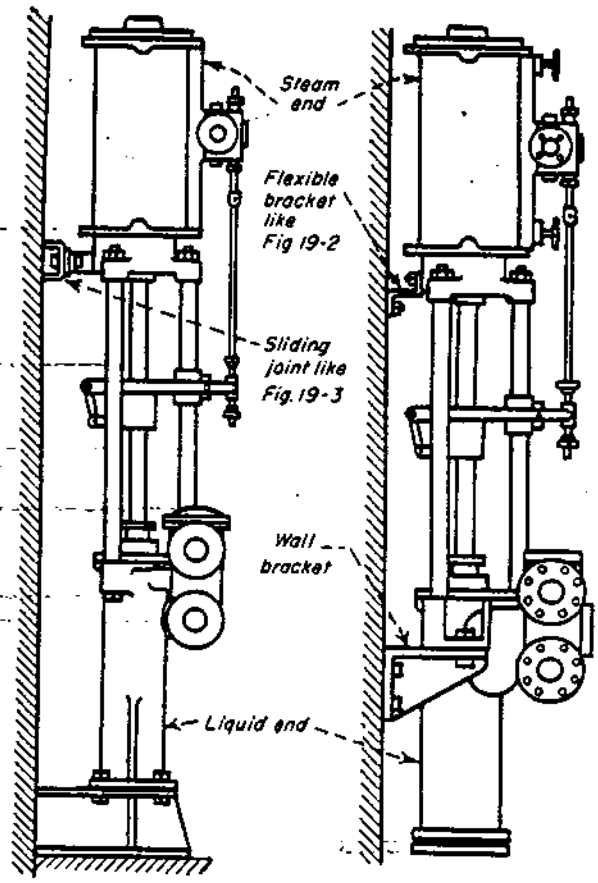


FIG. 19-4. FIG. 19-5.

FIG. 19-4. Vertical pump supported on foundation by an integral flange on the lower end of liquid cylinder.

FIG. 19-5. Direct-acting boiler-feed pump supported at upper end of liquid cylinder on a wall bracket.

Question 19-6: What special precautions should be followed in laying the suction line of a pump?

Answer: Have the suction line as short and direct as possible. It should never have a smaller diameter than the suction opening of the pump. Slope it up uniformly to the pump (Fig. 19-6) to avoid forming an air pocket. If an obstruction must be by-passed,

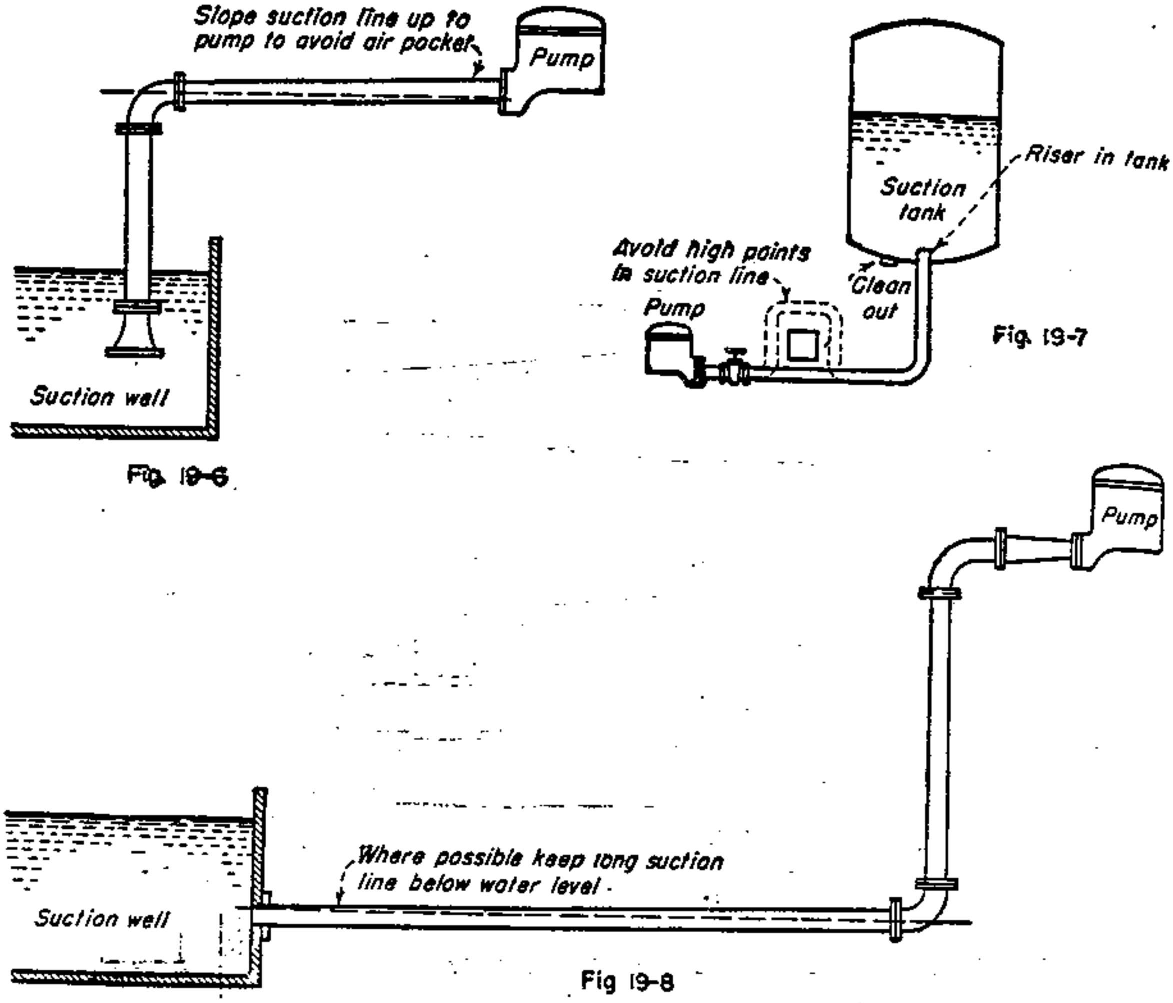


FIG. 19-6. Suction lines should slope uniformly up to pump to avoid air pockets.
FIG. 19-7. To avoid air pockets, suction lines should be run in horizontal plane around obstructions, not under or over them.
FIG. 19-8. When the suction line is long and suction lift is high, pipe should be kept, as much as possible, below suction liquid level.

go around it in a horizontal plane rather than over or under, to avoid an air pocket (Fig. 19-7) Make unusually long lines larger than normal size, using an offset fitting at the pump to avoid an air pocket (Fig. 19-8). When the suction lift is high and the line long, run as much of the pipe as possible at or near suction-well level; then let it rise vertically to the pump (Fig. 19-8). This will reduce air release from the liquid under reduced pressure. An oversized foot valve is desirable on an installation having a long suction line or a high suction lift.

Question 19-7: **When should strainers be used in pump suction lines?**

Answer: Whenever the fluid contains debris or foreign matter that may clog or damage the pump or the pumping system. When water is taken from a lake or river, install a strainer or trash rack at the entrance to the suction line. On a ship's bilge pump or on similar units, a strainer is placed at the pump suction connection rather than at the ends of the suction lines leading to various compartments.

Ordinarily, strainers are not used in a closed system such as a boiler feed circuit in a condensing plant. But to catch pipe scale, welding spatter, and slag or other dirt in the piping of practically every new system, install a temporary fine-mesh strainer in the suction of each positive displacement pump. This precaution is not needed after the plant has been in operation for some time. Such strainers would have saved many a pump liner and piston from early destruction during the recent war shipbuilding program.

Question 19-8: **How should the suction connection be arranged at the bottom of a heater or condensate return tank?**

Answer: When the suction is taken from the bottom of a tank, there should be a riser (Fig. 19-7), extending from one-half to one pipe diameter up into the tank, to protect the pump from the pipe scale and dirt that accumulates in any return system. Provide a clean-out opening near the suction connection for periodic inspection and removal of foreign matter.

Question 19-9: **What valves should be used in reciprocating pump piping?**

Answer: Where the liquid supply is above the pump, use a gate valve in the suction line near the pump. If the pump is located above the liquid supply, a suction gate valve is not required, although a foot valve is recommended if the suction line is long or if there is a high suction lift. On the discharge side, put a relief valve next to the pump, then a check valve, and finally a gate valve. Where power pumps are to start against load, connect a by-pass line of one-half the discharge diameter ahead of the check valve. This by-pass with its stop valve may connect to the suction side of the pump or return to the suction tank. Do not use globe valves, particularly on the suction side, because of their high friction loss.

Question 19-10: What are snifter valves and when are they used?

Answer: A snifter valve (Fig. 19-9) is a screw-down or stop-check valve used on the suction side of a pump to admit air into the suction under certain conditions. Such valves are most useful on drainage pumps, such as bilge pumps on ships, to admit just enough air to cushion the pump at maximum speed, even with long and sometimes restricted suction lines. Individual snifter valves may be used above each suction valve where the suction lift is small. Or a single valve may be connected to the suction line or manifold. Snifter valves are also used on moderate-pressure pumps to admit enough air to keep their discharge air chambers properly charged. (Also see Question 18-9.)

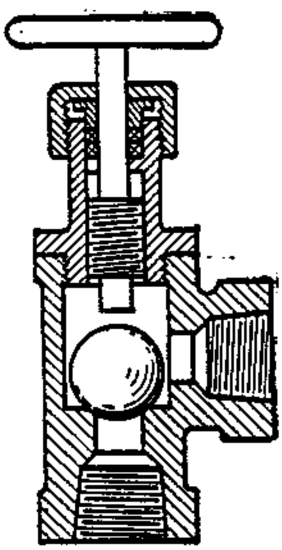


FIG. 19-9. Snifter valve used on the suction side of a pump.

Question 19-11: How is a vertical power pump installed?

Answer: Most vertical power pumps built today are small enough to handle as an assembled unit except for the driver. Their foundations must be heavy enough to maintain the pump and its driver in alignment. For a belt drive, provide a sliding-rail or tension base for the driver, to adjust belt tension. Level the pump frame both ways, and support it firmly and evenly. A rigid concrete foundation is recommended, except possibly for the smallest pumps. Support the frame at intervals and adjacent to all foundation bolts. leaving at least one-half inch of space for grouting.

After the grout has set, tightly pull down the foundation bolts. Be sure that the frame is not distorted during any stage of the installation.

Question 19-12: How is an enclosed-crankcase horizontal power pump installed?

Answer: In the smaller sizes with piston liquid ends, the pump may be handled like a vertical pump. Level the frame both ways, using the pinion shaft or finished surfaces on top of the frame in one direction and the crosshead guides in the other. Check the gear alignment to make sure the frame is not twisted. On larger pumps and those using forged-steel cylinders, locate the power ends first and then install the cylinders in proper alignment. Make sure that the cylinders are level both ways and that the plungers run true in the stuffing boxes for the whole stroke, before installing packing.

Question 19-13: How much backlash is required for pump gears?

Answer: About 0.010 in. per in. of circular pitch, with a minimum of 0.005 to 0.006 in. for small gears. To check backlash, set a dial indicator against the flank of a pinion tooth and rock the pinion back and forth against the main gear.

Question 19-14: How should the alignment of gears and tooth bearing be checked?

Answer: Level and parallel both shafts. If the shafts are symmetrical on both sides of the gears, use a pin gauge to check center distances at both ends of the shafts. If these are the same and both shafts are level, they are parallel. Check the tooth bearing by lightly coating the pinion with prussian blue and rotating to mark the contact with the gear teeth. Herringbone and spur gears may also be checked by inserting narrow strips of paper in the meshing point at both ends of a tooth and noting the pressure on both strips. Where gears are accessible, thickness gauges or feelers may be used to check both backlash and alignment.

Question 19-15: What precautions should be taken when starting a new direct-acting steam pump?

Answer: Blow and flush out the steam and water lines so that they are clean before connecting them to the pump. Clean the pump of slushing compounds, pack and lubricate it. Open exhaust, suction, and discharge valves. Be sure that there is liquid available on the suction side of the pump. Drain the condensate from the steam line, open cylinder drains, and crack the steam valve to warm up the pump cylinder. Gradually increase the steam pressure until the pump operates slowly, but do not close cylinder drains until most condensation has stopped. Increase the load and speed gradually, so that operating details may be watched and necessary adjustments made.

Question 19-16: What steps should be taken in starting a new power pump?

Answer: Flush the water lines clean, pack and lubricate the pump as in Question 19-15. Turn the pump through a full revolution by hand before applying power. Check rotation of the driver. Horizontal pumps run "over," so that pressure is down on the crosshead. Open vertical pumps run so that the load is down on the pinion bearings. Enclosed vertical pumps are marked for the direction of rotation required by the lubricating system.

Open suction and discharge valves, prime the pump if practicable, and open the hand by-pass, if provided. After starting the unit, run for a reasonable time at no load or with the by-pass open before gradually applying the load. Give the pump time to warm up before applying the full load; a new pump should have a further "run in" period. Check lubricating-system operation as soon as the unit starts; watch the bearings and stuffing boxes for excessive heating as load is applied.

After the pump is running satisfactorily at rated pressure, check the discharge relief-valve operation by throttling the outlet from the pump. It should begin to open at not more than 10 to 15 per cent above the rated pressure.

CHAPTER 20

STEAM- AND POWER-PUMP TROUBLES

Question 20-1: What are the common types of trouble?

Answer: Operating, or hydraulic, and mechanical. Direct-acting steam pumps and power pumps have many troubles in common.

Question 20-2: Why does a direct-acting steam pump sometimes fail to start?

Answer: Low steam pressure is a common condition that prevents a duplex steam pump from starting. Condensate in the cylinders or steam line seldom prevents the unit from starting slowly, although the line and cylinders should be drained of condensate. A simplex pump may stroke to one end, and then refuse to reverse. Often condensate causes this difficulty; it can be corrected by installing drain cocks on the steam chest covers. If a simplex pump has been standing idle for some time, heavy oil may prevent the movement of the steam-thrown valve piston. Disassembly and cleaning is the best remedy for this condition, but steam pressure and heat usually loosen such a piston.

Question 20-3: What prevents delivery of liquid?

Answer: A pump may fail to prime because of too high a suction lift, air leaks in the suction line, leaky pump valves, or worn liquid-piston packing. When starting against a discharge head, the pump may be unable to discharge even a small amount of air from the cylinder or suction line. The unit thus fails to deliver until the air is released from the discharge side at low pressure. When changing over from hot to cold water, the liquid cylinder and its contents must be cooled if the suction head is to be lower when pumping cold water. A hot pump quickly changed to operate with a suction lift becomes steambound. Operation under this condition quickly damages liquid-piston packing (see Check Chart, page 331).

Question 20-4: What causes a pump to deliver less than its rated capacity?

Answer: If a steam pump will not run at the desired speed, low steam pressure, high back pressure, and excessively tight packing are the usual causes. Steam-valve and piston-ring wear and leakage also limit pump capacity. If the unit runs at the rated speed and fails to deliver full capacity, the cause may be worn and leaking piston packing and valves, and air leaking into the suction.

Question 20-5: What makes a pump vibrate?

Answer: Common causes are misalignment caused by faulty installation, distortion due to expansion of the pump or piping, or warping of the foundation. Tight packing, particularly on the liquid piston, causes uneven action and vibration at low speeds (see Check Chart, page 334).

Question 20-6: What causes the rapid scoring and wear of piston rods and plungers?

Answer: Misalignment of the pump, unduly tight packing, packing that has lost its lubricant and become hardened, or the wrong kind of packing for the service. Dirt and scale from the pipe line or dirt in the liquid being pumped also score rods and plungers. The same considerations apply to liquid cylinder liners and pistons.

Question 20-7: How can the steam end of a pump be tested for abnormal leakage?

Answer: By operating the pump against load with the exhaust open to atmosphere. A tight pump exhausts relatively little steam during the stroke, but has a sharp and heavy exhaust at the moment of release when the steam valve passes center.

Question 20-8: How can the liquid end of a steam pump be tested for leakage?

Answer: By operating a fully primed pump at slow speed against a closed discharge valve. Throttle the steam supply to obtain no more than normal discharge pressure, and note the time for each stroke and for a complete revolution of the pump; this time should be 25 to 50 times that at normal speed. This is termed a "slip test"; the time required is a good indication of liquid-end tightness.

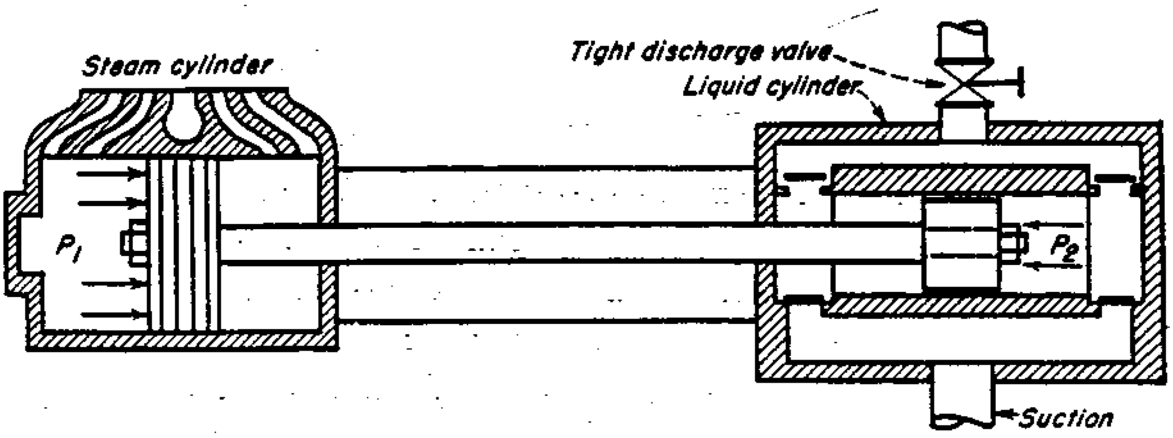


FIG. 20-1. If suction valves and discharge stop valves are tight and piston packing leaks freely, slip tests show how easily a pump can develop enough pressure to burst the liquid cylinder.

Question 20-9: What precautions must be observed in making a slip test?

Answer: Carefully control the discharge pressure by throttling the steam supply, even if the discharge is protected by a relief valve. Particularly in a simplex pump, if the suction valves and discharge stop valve are tight and the liquid freely by-passes the liquid-piston packing, the piston rod enters the cylinder like the plunger of a single-acting plunger pump (Fig. 20-1). Steam pressure P_1, acting on the full steam piston, must be balanced by liquid pressure P_2, acting on the piston-rod area only.

This action, termed "hydraulic-ing," may develop sufficient pressure to burst the liquid cylinder, if the piston packing is badly worn. On a demonstration the authors have seen a pump that normally requires 200 psi steam to develop 300 psi liquid pressure, build up a liquid pressure of 750 psi on only 25 psi steam. This operation was on a new pump, before the liquid-piston rings were seated against the cylinder liner; badly worn packing may cause an even more severe pressure increase.

Question 20-10: How can a slip test be used to distinguish between leaky valves and leaky piston packing?

Answer: Leaky piston packing usually permits the piston to slip at nearly the same rate for both directions of motion. Two or more valves seldom leak at the same rate. Thus the rate of slip

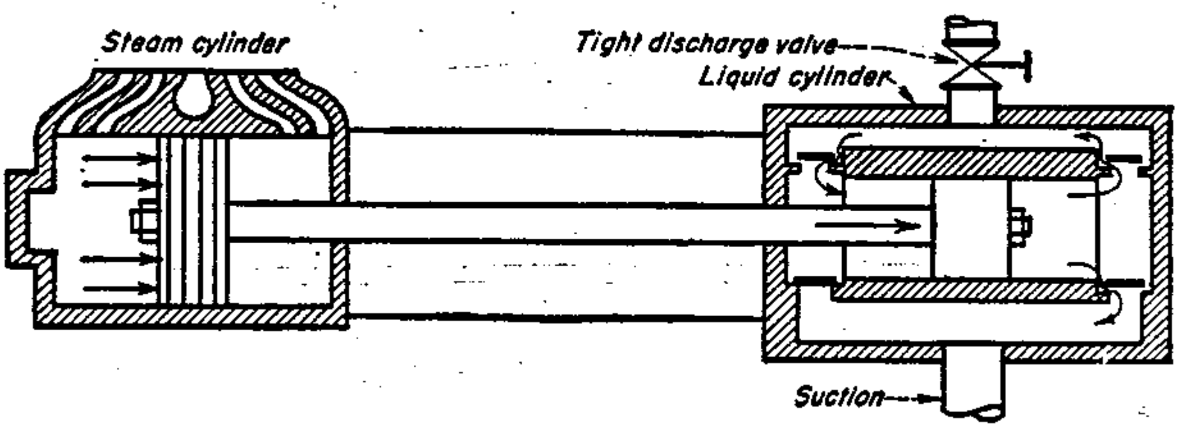


FIG. 20-2. On a slip test, leaking suction valve on pressure side of piston or leaking discharge valve on suction side permits movement as indicated.

varies with the direction of the motion. A high rate of slip in one direction may be caused by a leaking suction valve on the pressure side of the piston, allowing loss of water back to the suction with little or no delivery. A leaking discharge valve on the suction side of the piston may allow water from the discharge to nearly balance the pressure on both sides of the piston. Figure 20-2 indicates the effects of these faults.

Question 20-11: A duplex steam pump operated satisfactorily at low and medium speeds, but as the speed increased, the pump hesitated, then stalled when the right-hand steam piston reached the head end of stroke. What caused this condition?

Answer: Investigation showed that the left-hand steam-valve adjustment was off center, and that there was less than normal clearance between the cradle end of this valve and the inside of the steam chest. As the pump stroke lengthened with increasing speed, the valve moved farther and farther, until contact with the steam chest stopped it. Further motion of the valve rod then lifted the valve from its seat, opening the steam chest and all ports on the left-hand side of the pump to exhaust. Figure 20-3 shows this action greatly exaggerated. After the chest was chipped out

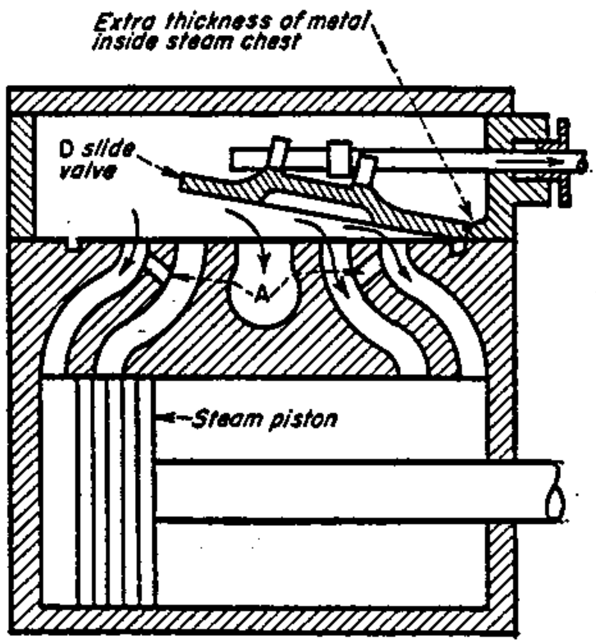


FIG. 20-3. When a "D" slide valve was adjusted off center, it was lifted off its seat by being pulled against a lump of metal inside the steam chest, stalling the pump.

to provide normal clearance and the valves correctly adjusted, the pump ran perfectly at all speeds.

Question 20-12: Two identical vertical simplex direct-acting boiler feed pumps were installed in a new plant. The first pump seemed all right, but later developed an irregular twisting motion of the piston-rod crosshead. The second pump had to be run at least twice as fast as the other unit to carry the same load. What caused this trouble?

Answer: Lateral swelling of relatively hard piston packing was the cause. In the first pump, the packing was set against the cylinder liner. Because the pump was in use, gradual swelling locked the packing tightly in the piston groove (Fig. 20-4*A*). Further swelling caused abnormal friction between packing and cylinder liner. The second pump was not in use when its packing swelled laterally to jam in the piston groove (Fig. 20-4*B*). Consequently, there was clearance between the packing and the liner, permitting considerable leakage. Each liquid piston was remachined to provide packing clearance in its groove, thus allowing the packing to float and operate as it should.

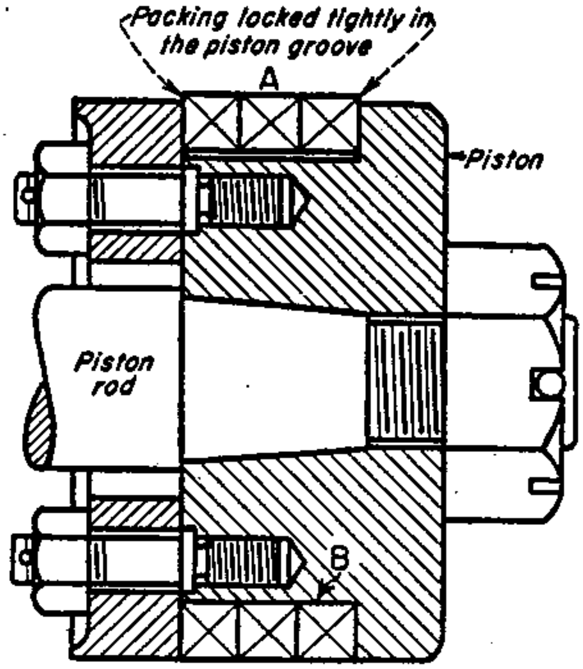


FIG. 20-4. The swelling of the packing can lock it in the piston groove causing abnormal friction in some cases, and excessive leakage by the piston in other cases.

Question 20-13: A small duplex direct-acting pump with piston steam valves short-stroked badly. Increasing the lost motion in the valve gear, in an effort to increase the stroke, required increased steam pressure to operate the pump. What was the cause?

Answer: It was found that the one-piece rings on the piston valves were slightly undersize, permitting them to open to an unusually large gap. Live-steam leakage through this gap, and through the steam port to the end of the cylinder, created a live-steam cushion that stopped the main piston as soon as it covered the exhaust port. After the piston valves were fitted with two-piece compressor-type rings, which provided a seal at the gap, the pump operated at full stroke and responded normally to adjustment of the valve setting.

Question 20-14: At the end of each stroke of a duplex direct-acting boiler feed pump, the discharge-line check valve slammed shut with a shock that shook the entire boiler feed line. How was this condition corrected?

Answer: The steam end of the pump had too much cushion for boiler feed service. Increasing the lost motion to make the unit run at rated stroke retarded its action so that there was a definite pause instead of a slight overlapping of stroke at each reversal. Slamming of the check valve was eliminated by (1) drilling cushion release holes between the steam and exhaust ports, as at *A* (Fig. 20-3), and (2) reducing the lost motion to a normal amount. Careful adjustment permitted satisfactory operation during boiler blowdown periods, when the pump had to operate at 50 per cent above its rated capacity.

Question 20-15: A vertical simplex direct-acting bilge pump with a discharge air chamber could not operate at more than one-half its rated speed without developing a bad water hammer. Why?

Answer: A combination of suction and discharge piping, together with the discharge air chamber, created a condition where pressure in the chamber dropped to less than one-half the rated discharge value while the pump paused at each end of the stroke. At the start of each stroke, lack of resistance on the discharge side permitted an unusually high piston velocity, which the suction column could not follow. As the pressure built up in the discharge air chamber to retard the piston, the suction column caught up with the liquid piston with a severe shock or water hammer, at about mid-stroke. Elimination of the air chamber made it possible to operate the pump at rated speed. It was recommended that the cushion chamber be transferred to the suction side.

Question 20-16: What caused a severe knock in the liquid end of a vertical simplex direct-acting boiler feed pump, occurring only at the start of the upward stroke?

Answer: Liquid pistons in such pumps are usually fitted on a taper on the piston rod, and held by a nut. When slightly loose, the piston seats quietly on the taper under the load of the down-stroke. If often sticks on the taper, however, until almost full pressure is developed at the start of the upward stroke. If load is sufficient to start a loose piston off the taper, a severe knock occurs when the piston strikes the piston-rod nut. For present operating pressures, piston-rod nuts must be drawn up very tightly, and often retightened after the pump has been in service a short time.

Question 20-17: Why do power pumps fail to discharge?

Answer: They fail from the same causes as direct-acting steam pumps. See causes and remedies for "Pump fails to discharge," in Check Chart, page 331.

Question 20-18: Why do power pumps fail to discharge full capacity?

Answer: From the same causes as direct-acting steam pumps. See causes and remedies for "Pump not up to capacity," in Check Chart, page 332.

Question 20-19: What causes the suction or discharge line, or both, to vibrate on a power pump?

Answer: Vibration is generally caused by either the suction or the discharge line, or both, being too small. (See remedies in Check Chart, page 335.)

Question 20-20: What causes the liquid end of a power pump to pound or vibrate?

Answer: Among the causes are:

1. Air or gas in the liquid.
2. Suction lift too high or insufficient suction head.
3. Pump speed too high for existing hydraulic conditions.

4. Pounding of discharge or suction valves, or both.
5. Pump cylinders' failure to fill completely.
6. Viscosity of liquid too high.

(See remedies in Check Chart, page 335.)

Question 20-21: What causes the power end of a pump to pound or vibrate?

Answer: Causes are:

1. Excessive speed.
2. Loose or worn bearings.
3. Loose or worn crosshead or guides.
4. Loose crosshead pin or crank pin.
5. Gears out of line or improperly adjusted.
6. Worn or noisy gears.

(See remedies in Check Chart, page 336.)

Question 20-22: What causes a power pump to take excessive power?

Answer: The packing may be too tight, the discharge pressure too high, or the liquid's viscosity so high as to cause excessive friction losses in the suction and discharge lines. (See remedies in Check Chart, page 336.)

Question 20-23: How can the liquid end of a power pump be checked for leakage of valves or leakage between chambers?

Answer: A slip test such as is used on a direct-acting steam pump cannot be applied to a power pump, unless it can be turned by hand against a closed, or practically closed, discharge valve. Unusual fluctuation of suction or discharge pressure is a good indication of internal leakage, as is the speed or operating time the pump requires to deliver a known volume of liquid. A final check can be made only by applying water pressure to the suction or discharge chambers and looking for leakage past the valves or into adjacent chambers.

Question 20-24: Why is it important to drain and replenish the oil supply regularly in any enclosed-crankcase power pump?

Answer: The oil becomes contaminated, particularly by condensation, and must be renewed as often as existing conditions require. Whenever a pump is shut down, especially in a humid atmosphere, condensation occurs as the pump cools. After settling, most of this water and some of the dirt and grit that accumulate in any oil sump may be drawn off from its bottom. At regular intervals, however, draw out all oil and thoroughly clean the sump.

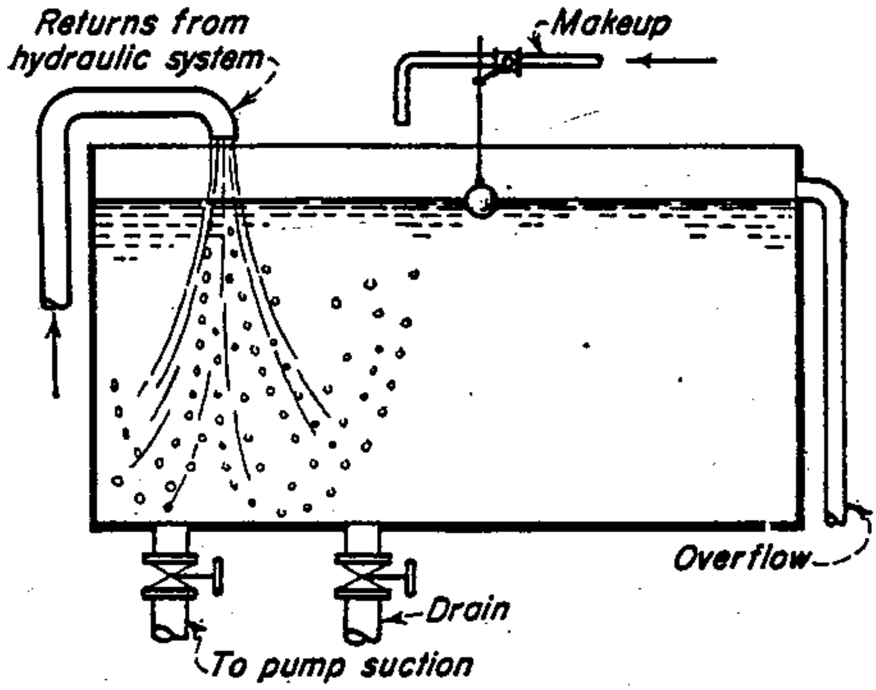


FIG. 20-5. Incorrect way to bring returns and make-up into tank. Air in the returns plus that carried below the water surface by the jet enters suction in small bubbles, causing rapid corrosion of pump cylinder.

If a centrifuge is available, oil which is not too badly contaminated can be cleaned for further use. When there is doubt about the condition of the oil, it is better to replace it than run the risk of damaging costly machinery.

Question 20-25: Why is it important to keep connecting-rod bolts pulled up tightly at all times?

Answer: A machine part subjected to widely fluctuating loads fails much more quickly than a similar part carrying a constant load of the same or even greater magnitude. A failure caused by repeated application and removal of load is known as "fatigue failure." Bolts or studs joining two parts that are alternately loaded and unloaded should be given a greater initial stress than that occurring under operating conditions. This has the effect of putting a steady load on the bolts and prevents fatigue failure, although the parts joined are not under a constant load. Connecting-rod bolts, subjected to additional stress due to shock when bearings wear loose, need attention to keep them tight.

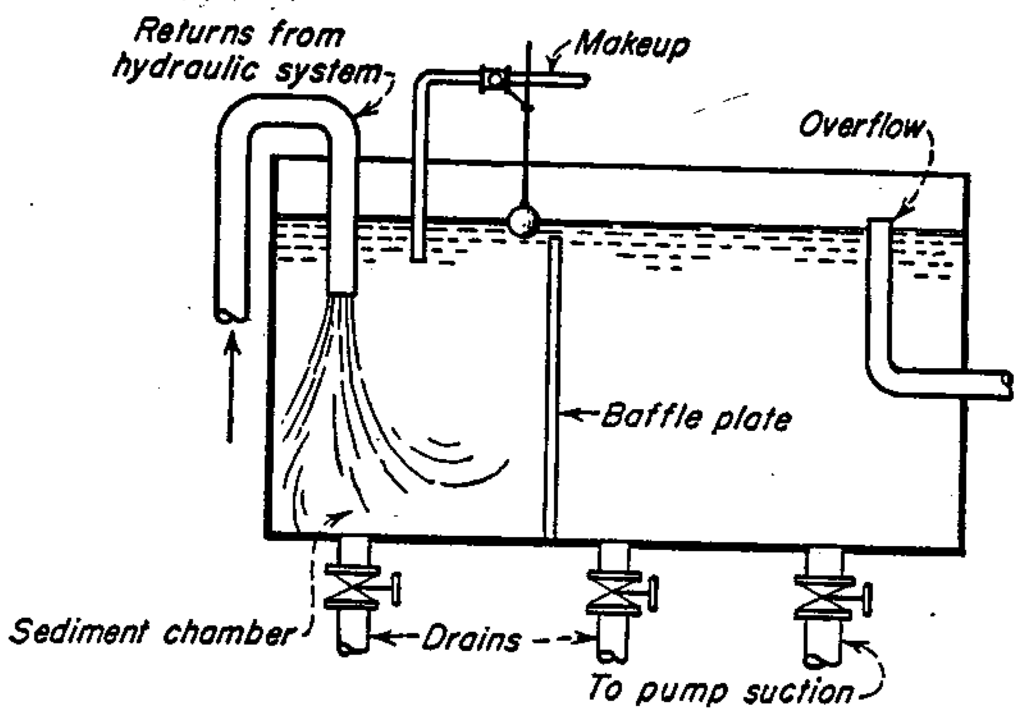


FIG. 20-6. Correct way to bring returns and make-up into a tank. The ends of return and make-up lines are submerged behind a baffle. All water flows slowly over the baffle to release air. Top of baffle should be sufficiently below low-water level to avoid disturbing the flow.

Question 20-26: What is the common cause of abnormal corrosion in the fluid ends of hydraulic pumps?

Answer: Excessive air entrained or dissolved in the water going to the pump. Returns and make-up to pump-suction supply tanks are too frequently arranged so that air bubbles are carried down into the water and into the suction outlet (Fig. 20-5). This air collects on the suction stroke as small bubbles above the suction valves and in the plunger chamber. The violent collapse of these bubbles, as pressure increases at the start of each discharge stroke, creates a shock condition not unlike cavitation.

This action destroys the iron-oxide coating on the cylinder and other surfaces that the bubbles contact, exposing fresh metal to the corrosive attack of air and water. Submerge all return and make-up lines in a section of the supply tank separated from the suction end by a submerged baffle (Fig. 20-6). By making all the

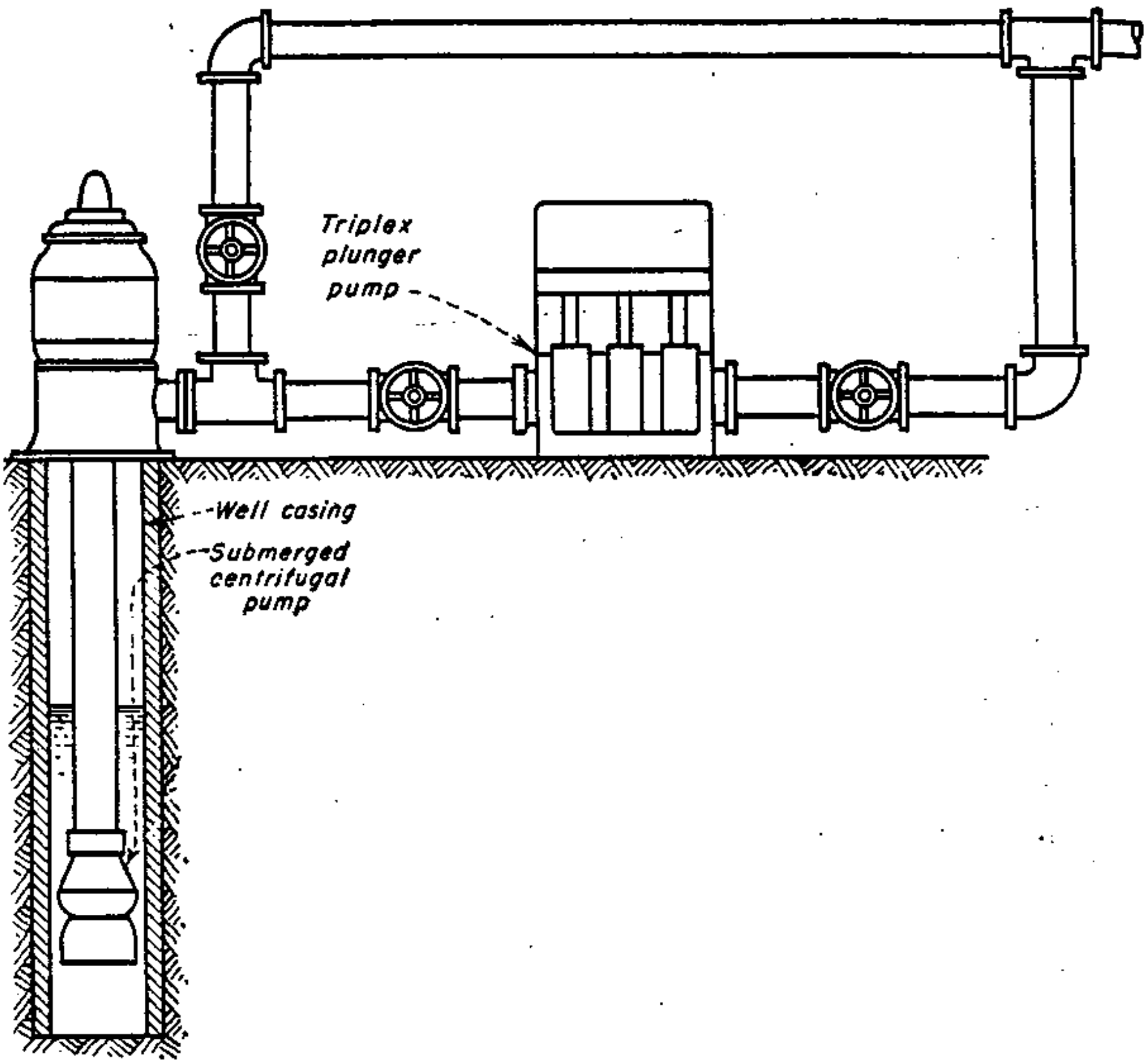


FIG. 20-7. To avoid disturbing a centrifugal well pump, a plunger pump was connected to take its suction through the barrel and outlet of the former pump.

water flow over this baffle, submerged a few inches below minimum water level, a large part of the dissolved air can escape before the water enters the suction line.

Question 20-27: A motor-driven triplex shallow well pump in an installation requiring a relatively high suction lift was difficult to prime when first started. Soon it lost its prime on one and

sometimes two plungers, even though pumping continued at proportionately reduced capacity. What caused this action?

Answer: Well draw-down caused by overpumping was partly responsible, but an unusual suction line with an excessive friction loss was the primary cause. Suction was taken through the barrel of a centrifugal well pump which filled the well casing (Fig. 20-7). Reducing the pump's speed decreased the discharge rate, which in turn reduced the friction loss and well draw-down to where the pump could work at a lift of 22 to 25 ft of water instead of failing at a more than 26-ft lift. Later an ordinary suction line of proper size was installed, which permitted the pump to work at a somewhat increased capacity.

Question 20-28: A motor-driven triplex boiler feed pump operated satisfactorily at minimum speed but overloaded its motor badly at increased speed. This overload reached a peak once each revolution of the pump, causing a violent swing in power input. What was the cause?

Answer: Satisfactory operation at 50 per cent rated speed makes it seem unlikely that an obstruction in the pump's liquid end caused the overload. It was found, however, that on two of the three plungers the valve stops limited discharge-valve lift to about $\frac{1}{64}$ in., which imposed a severe throttling action as water velocity increased. Apparently these valves initially had just enough lift to the stops to permit full-speed operation, but compressing the gaskets under the valve plugs caused the difficulty. Cutting back the stops to allow valve lift of $\frac{1}{4}$ in. cured the trouble.

Question 20-29: A motor-driven triplex boiler feed pump was installed in parallel with a turbine-driven centrifugal unit (Fig. 20-8). Following replacement of suction and discharge valve springs, the triplex pump refused to discharge or even build up pressure in the discharge line. Why?

Answer: To test this unit, the centrifugal was stopped and the triplex pump started. Between each trial the centrifugal pump was run long enough to keep water in the boiler. Since the triplex pump had worked before putting in the new springs, it was assumed that the pump was airbound and that for some reason it would not expel air and prime itself even though there was a reasonable suction head. Finally, instead of depending on the discharge check valve to prevent backflow through the centrifugal pump, its discharge stop valve was closed. Pressure and feed to the boiler

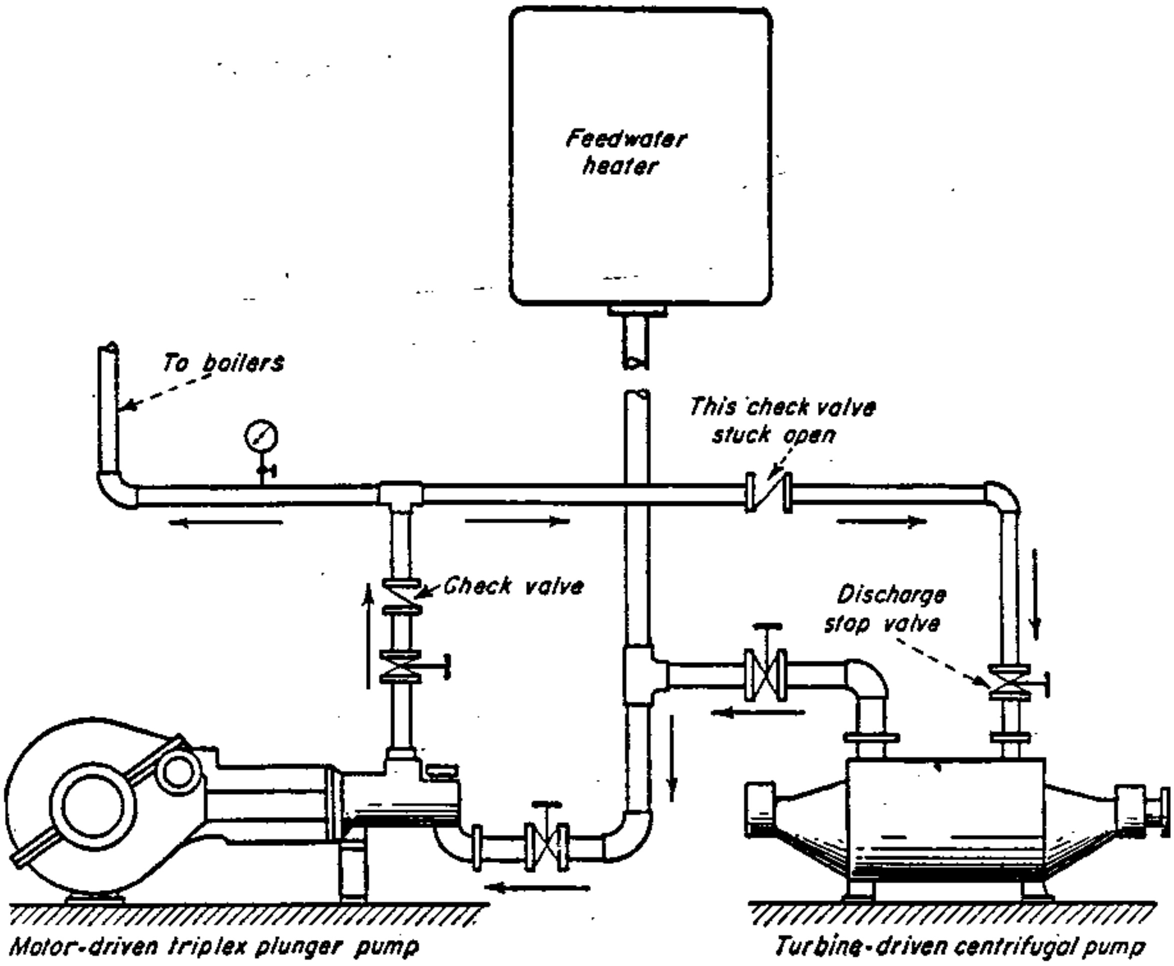


FIG. 20-8. A motor-driven triplex boiler-feed pump, in parallel with a turbine-driven centrifugal pump, failed to discharge into the boilers because, as it was found later, the check valve of the centrifugal pump was stuck wide open.

were immediately established, showing that the check valve in the centrifugal discharge was stuck open during the triplex-pump trials. (Figure 20-8 shows the check valves on the discharge side of the discharge gate valves as in the actual installation. It is better to install a check valve between the discharge gate valve and the pump so that the check valve can be isolated for repairs by simply closing the gate valves on the pump.)

CHECK CHART FOR DIRECT-ACTING STEAM PUMPS
CAUSES OF TROUBLE AND SUGGESTED REMEDIES

I: Pump Fails to Discharge

A. Not Properly Primed

Reprime pump. Keep vents open on force chamber (above discharge valves) until pump is free of air. Use screw-down check valves for venting individual cylinders to prevent reentry of air on suction stroke.

B. Suction Lift Too High

For cold water at sea level, suction lift should not exceed 22 ft. Check with vacuum gauge.

C. Air Leaks in Suction Line

Blank off suction line and test with water pressure. Seal joints with suitable compounds.

D. Airbound

Prime as under *A*. Pump should have discharge check valve so that air may be vented without compressing it to discharge pressure.

E. Steambound

Not enough suction head for hot water. Provide at least 15 ft of static suction head for 212°F and hotter water. In changing from heated suction to cold suction from a lower level, pump must be cooled before it can pick up with a suction lift.

F. Suction Line Strainer Clogged

Examine strainer and inside of suction line to make sure that they are free of all obstructions.

G. Foot Valve Stuck

Clean valve and make sure that it can open freely. Pipe may screw into valve far enough to hold it closed.

H. Suction Valves Stuck

See that valves are free and working properly.

I. Suction Valves Worn

Reseat valves. Renew valves and seats if necessary.

J. Piston Packing Badly Worn

Replace with properly fitted packing. If necessary, replace or rebore liners.

K. Rod Packing Worn

Repack rod stuffing boxes.

II: Pump Not Up to Capacity: Discharge Pressure Low

1. If Pump Speed is Slow

A. Low Steam Pressure

Under average conditions, steam pumps have about 75 per cent efficiency. That is, at rated speed, the ratio of discharge pressure to steam pressure will be about 75 per cent of the ratio of piston areas.

B. High Back Pressure.

The effective pressure is the steam pressure minus the back pressure. Check back pressure in calculating performance under *A*.

C. Tight Packing

Unduly tight piston or rod packing will limit speed of pump.

D. Worn Steam End

Excessive leakage of steam-piston rings and steam valves will lower efficiency. Reface and renew as required.

2. If Pump Speed Is Fast

A. Worn Piston Packing

Repack liquid piston. Rebore or replace liner if necessary.

B. Blocked or Leaky Valves

Remove any material blocking valves open. Reseat valves, or renew valves and seats as required.

C. Air Leaks in Suction

Check suction line for leaks. A high spot in a suction line will permit air to collect until it passes into pump all at once, airbinding one or more cylinders. Rearrange line to slope continuously up to pump suction.

D. Excess Air or Gas in Water

If water contains excess air or gas that separates out at high suction lift, provide a tank near the pump suction with a float-controlled vacuum pump to remove air from top of tank as it separates from the water.

E. Suction Lift Too High

For cold water, do not exceed 22 ft with average pump.

F. Foot Valve or Strainer Partly Clogged

Excessive friction and a high suction gauge reading shows when foot valves or strainers are partly clogged. Clean as required.

G. Suction Pipe Not Sufficiently Submerged

Extend intake end of suction line far enough into suction well to prevent vortexes forming at the surface of the liquid, through which air is drawn into the suction line. Deepen suction well if necessary. For an emergency condition, planks floating on the surface of the water usually prevent vortexes forming.

III: Short Stroking

A. Tight Packing

Packing in the steam and liquid ends of a pump should be only tight enough to reduce leakage to a reasonable amount. Allow for swelling of packing, particularly in pistons, under service conditions.

B. Excessive Cushion in Steam Cylinders

If provided, adjust cushion valves to get rated stroke. If a pump consistently short-strokes, and cushion valves are not provided, cushion release holes between steam and exhaust ports may be necessary.

C. Steam-valve Leakage

Leakage past the steam side of a pump valve provides live steam in end of cylinder, which stops the piston as soon as it cuts off the exhaust port. Replace valve and seat to stop leakage.

D. Air or Gas in Liquid Pumped

Excessive air or gas in liquid pumped will cause severe short-stroking of a duplex pump. Correct suction conditions.

E. Piston Rings Striking Shoulders Worn in Steam-cylinder Bore

Remove shoulders by reboring cylinder. Fit oversized pistons and rings to suit bore. If wear is not too great, shoulders may be removed with portable grinder. Adjust pump to full stroke to prevent wearing shoulders at end of stroke.

F. Not Enough Lost Motion

If adjusting nuts are provided, increase lost motion to lengthen stroke. This applies particularly to simplex pumps. If other remedies fail, reduce thickness of square nut in fixed lost-motion duplex pumps.

G. Valves Set Wrong

Set valves according to instructions supplied by pump manufacturer.

H. Pumps Not Properly Lubricated

A small but steady supply of lubrication to steam cylinder is essential to smooth and trouble-free operation.

IV: Long Stroking, Piston Strikes Head

A. Cushion Valves Not Properly Adjusted

Cushion valves should be closed enough to keep steam pistons from striking the cylinder head at maximum operating speed. Where pumps are run for fairly long periods at reduced speed, cushion valves may be opened to prevent short-stroking, and closed again before operating at increased speed. Adjust cushion valves under load to suit maximum conditions.

B. Too Much Lost Motion

Reduce lost motion to get rated stroke at maximum speed with cushion valves nearly closed. Too much lost motion in a duplex pump will make the pump sluggish and prevent any overlapping of the strokes of the two sides of the pump.

C. Worn Steam-piston Rings

Badly leaking steam-piston rings make the cushion ineffective. Replace rings, reboring cylinder if necessary.

D. One Piston Strikes Head, the Other Short-strokes

This condition is most likely to develop in an outside-packed plunger pump, and is usually caused by excessive packing friction on one plunger. Loosen gland or remove one or more packing rings on the short-stroke side, as required to balance friction. Look for leaky valves in the liquid end on the side that overstrokes.

V: Vibration and Pressure Fluctuation

A. Pump out of Line

Check foundation and hold-down bolts to be sure that pump is not pulled out of shape when at operating temperature. Be sure that piping strains do not pull pump out of line.

B. Packing Too Tight

Overly tight piston packing will cause chatter and vibration, particularly at low speeds. Allow for expansion of piston packing under service conditions. When unfamiliar with the packing used, check for swelling after it has been in service for a week or two.

VI: Pressure Fluctuates, Jumps at Start of the Stroke

A. Air in Liquid End of Pump

If suction lift is high, operate pump at slow speed with open vents until it is certain that all air is out of pump and suction line.

B. Speed Too High

If liquid is viscous, heat to reduce viscosity and increase the suction head to make sure that pump fills. The service conditions, as much as the pump's design, determine the maximum operating speed. Extra-long suction and discharge lines require reduced speed, unless effective cushion chambers are kept properly charged.

C. Not Fully Primed

See *A* in Section I, "Pump Fails to Discharge."

D. Not Enough Suction Head for Liquid Temperature

Reduce speed of pump, increase suction head, or reduce temperature of liquid. See *E* in Section I, "Pump Fails to Discharge."

VII: Pump Stalls

A. Valves Improperly Set

Reset steam valves in accordance with manufacturer's instructions.

B. Steam Valves Worn

A duplex steam pump may stall or hesitate when just past midstroke, because of steam leakage past the steam valve on the opposite side of the pump. At five-eighths' stroke, the valve it actuates is at midposition, where severe leakage robs the operating piston of its motive steam. Leakage of the pilot valve or valve-operating piston of a simplex pump causes it to stall at the end of its stroke.

VIII: Excessive Stuffing-box Leakage

A. Shoulder Worn in Cylinder at Ends of Stroke

When the steam pistons hit the shoulders, they lift the rod in the stuffing box, causing blow-by at the ends of the stroke. Correct as under *E* in Section IV, "Short-stroking." Run pump at not less than rated stroke, so that rings may override the counterbore provided to prevent the formation of such shoulders.

B. Rods Worn

Since a piston rod cannot run clear through the packing, it wears less at each end than in the middle. This causes leakage throughout the center part of the stroke. Replace worn rods, or restore to original diameter by building up with spraying and refinishing.

C. Pump out of Line

See *A* under Section VI, "Vibration and Pressure Fluctuation."

D. Bent Piston Rod

Repair or replace piston rod. Piston rod must be true and smooth throughout the length that runs in the packing.

CHECK CHART FOR POWER PUMPS*
CAUSES OF TROUBLE AND SUGGESTED REMEDIES

I: Pump Fails to Discharge

A. Same as for Direct-acting Pumps

Same as for direct-acting pumps; (see Check Chart, page 331).

II: Pump Not up to Capacity

A. Same as for Direct-acting Pumps

Same as for direct-acting pumps; (see Check Chart, page 332).

III: Suction and or Discharge Line Vibrates or Pounds

A. Suction Line Is Too Small

Increase size of line, decrease its length. Use a centrifugal booster pump to keep minimum of fluctuating pressure above the flashing point of the liquid pumped. Use a suction air chamber at pump suction to smooth out the flow in the suction line.

B. Discharge Line Is Too Small

Increase size of line, shorten it if possible. Use a discharge air chamber, if practicable, to smooth out flow in line. If two or more pumps discharge to a common manifold some distance away, install equalizing lines between pump discharge connections.

IV: Liquid End of Pump Pounds or Vibrates

A. Air or Gas in Liquid

If air or gas is released by suction lift, install separation chamber near pump with automatic air-removal apparatus. Check for and correct air leaks in suction line and around stems of valves. Low liquid level in suction well or turbulence due to the inflow falling into the well close to the suction pipe may cause entrainment of air. Submerge inflow lines at a maximum distance from suction pipe.

B. Suction Lift Is Too High, or Suction Head Is Insufficient

Older low-speed pumps may be capable of operating at a suction lift of 22 ft on cold water, but higher speed pumps may require a flooded suction or even a positive-suction head. Consult manufacturer for suction requirements, and provide ample suction head at all times.

C. *Excessive Speed*

Operate at rated speed for average conditions. If lines on suction and discharge are long, reduced speed may be necessary. If suction head is ample, increase strength of valve springs in proportion to square of increase in speed.

D. *Valves Are Pounding*

Look for broken or weak valve springs. Replace as required. Increase strength of springs if suction head permits, or reduce pump speed.

E. *Pump Does Not Fill*

Increase suction head or reduce speed of pump. Reduce strength of suction valve springs in proportion to square of speed reduction.

F. *Viscosity Too High*

Heat liquid to reduce viscosity or reduce pump speed and possibly valve-spring loading.

G. *The Pump Vibrates*

Check for alignment. Be sure that pump is level and properly supported.

V: Power End of Pump Pounds or Vibrates

A. *Speed Is Excessive*

Reduce speed to rated value or to meet operating conditions.

B. *Loose or Worn Bearings*

Tighten or replace bearings. Bearing bolts, particularly at crankpin end of connecting rod, must be kept tight at all times.

C. *Loose or Worn Crosshead or Guides*

If crosshead has separate shoe, adjust or shim out the shoe to restore original running clearance. Rebore guides as required and install oversized crossheads to suit.

D. *Loose Crosshead Pin or Crankpin*

Replace with properly fitted pins and adjust bearings to suit new pins.

E. *Gears Out of Line or Improperly Adjusted*

Realign gears for even tooth bearing and adjust to proper backlash.

F. *Worn and Noisy Gears*

Replace gears, adjusting to give proper backlash.

VI: The Pump Takes Too Much Power

A. *Packing Is Too Tight*

Check packing; if too tight make necessary adjustments or replace with correct type of packing, properly installed. Do not fill stuffing boxes too full; allow for expansion.

B. *Discharge Pressure Is Too High*

Check piping system for abnormal obstructions, such as a partly closed valve. If installation is new or changed, recalculate friction loss through fittings and pipe. Check discharge head against rating of pump. Reduce size of piston or plunger to get no more than manufacturer's rated pressure for size of piston or plunger used. Power pumps are built for a certain maximum cylinder pressure with a minimum-size piston or plunger. Use of larger size pistons or plungers requires a proportionate reduction in discharge pressure to avoid overloading the power end of the pump. If motor is overloaded, reduce speed of pump if pressure is within the pump's rating; otherwise, reduce load on motor and pump by changing to smaller pistons or plungers. At rated speed, install larger motor if this will not overload the pump.

C. *Viscosity of Liquid Causes Excessive Friction Loss*

Increase pipe size, reduce pump speed, or reduce viscosity of liquid by heating.

INDEX