Part 2

Applications

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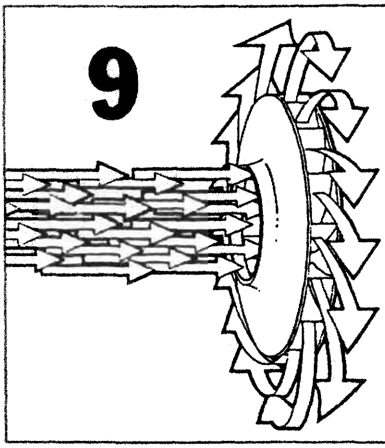

Vertical Pumps

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This chapter discusses radially split bowl pumps that are typically mounted vertically. In older literature these pumps are often, but improperly, referred to as vertical turbine pumps. This pump type is unique in that designs with optimum efficiency can be obtained over the full specific speed range, normally with values from 1,500 to 15,000. In the upper specific speed range, the pumps are referred to as axial flow or propeller pumps. The impeller profile changes with the specific speed as shown in Figure 9-1.

The hydraulic performance parameters, including efficiency, compare favorably with centrifugal pumps of the volute and diffuser type. However, except for highly specialized designs, vertical pumps are seldom used for high speed applications above 3,600 rpm.

Vertical pumps can be designed mechanically for virtually any application and are the only suitable configurations for certain applications such as well pumping. They are commonly used for handling cryogenic liquids in the minus 200°F to minus 300°F range as well as for pumping molten metals above 1000°F. The radially split bowl design lends itself to safe, confined gasketing. For high pressure applications, typically above 1,000 psi discharge pressure, an outer pressure casing can be employed, similar to that which is used for double-case, horizontal pumps. Except for conventional well pumps, the mechanical design for the majority of vertical pumps is customized in accordance with the application requirements. This requires close cooperation between the pump manufacturer and the architect/engineers responsible for the pump mounting structure and system piping.

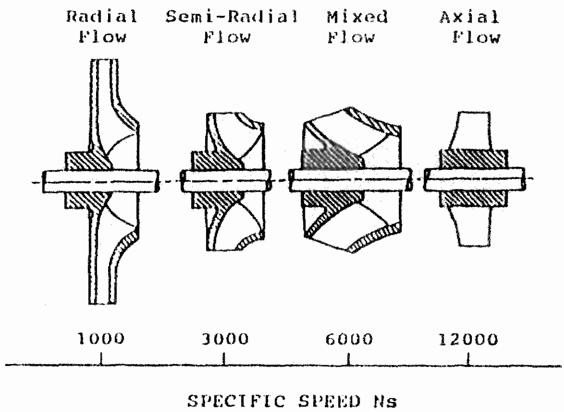

Figure 9-1. Specific speed and impeller profiles.

Configurations

There are three primary types of vertical pump configurations that are used for a broad range of applications.

Well Pumps

Designed to be installed in cased wells, these pumps consist of a multistage pumping element or bowl assembly installed at sufficient depth below the dynamic water level (the water level when the pump is operating) and with sufficient NPSH to preclude cavitation. The subject of NPSH is dealt with in detail Chapter 8. The bowl assembly, as illustrated in Figure 9-2, consists of a series of impellers mounted on a common shaft, and located inside diffuser bowls. The number of stages is determined by the height to which the liquid must be raised to the surface plus the design pressure required at the surface. The bowl assembly is suspended from a segmented column pipe that directs the flow to the surface where the column pipe is attached to a discharge head. The column also houses the lineshaft with bearings for transmitting the torque from the driver to the bowl assembly. The discharge head, in addition to providing the required connection to the customer's piping, also serves as the base for the driver. The driver can either be a direct electric motor drive, typically of hollow shaft construction, see Figure 9-3, or a right angle gear drive powered by a horizontal engine or turbine. The discharge head must be supported on a foundation adequate to carry the water-filled weight of the pumping unit plus the driver weight. However, the hydraulic thrust developed by the pump impellers is not transmitted to the foundation.

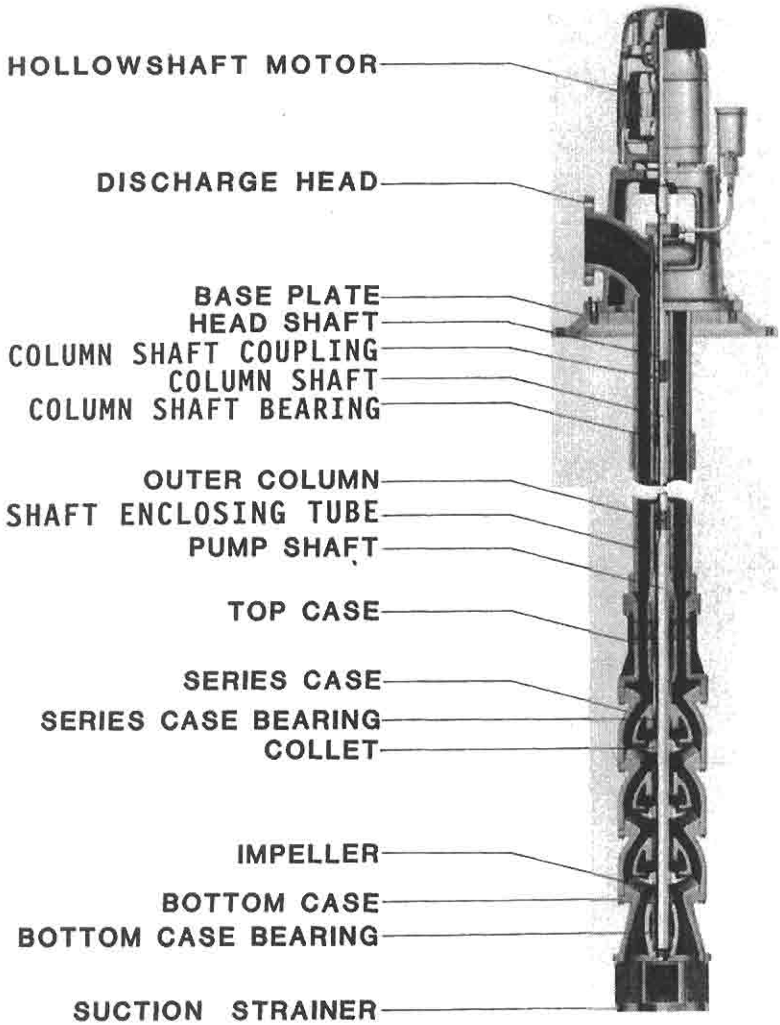

Figure 9-2. Well pump with hollow shaft, electric motor (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

An alternate well pump design uses a submersible electric motor drive, which is close coupled to the pump as shown in Figure 9-4. The motor can either be of the "wet winding" or "dry winding" design. For large motors, 250 HP and up, the preferred construction is the dry winding with the motor sealed and oil filled [2]. The motor is typically mounted below the pump, so there will be continuous flow of liquid around the outside of the motor for cooling. The submersible configuration eliminates the need for a lineshaft

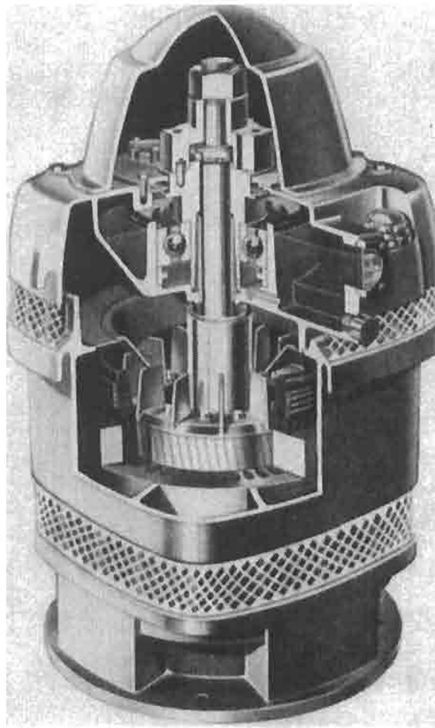

Figure 9-3. Vertical hollow shaft motor (courtesy U.S. Electrical Motors, Division of Emerson Electric Co.).

with bearings and its inherent, critical alignment requirements. Only a conventional, taper thread discharge pipe with the power cable attached leads to the surface. Here it is connected to a discharge elbow on which the electrical conduit box is also mounted. It should be noted that the well casing must be sized so that there is room alongside the bowl assembly for the power cable and a protective guard.

Wet Pit Pumps

This pump configuration, illustrated in Figure 9-5, can be either of the single-stage or multi-stage design, depending on the application requirements and covers the complete range of specific speeds. Installed in a pit or inlet structure, the water surface on the suction side of the pump is *free* and subject to atmospheric pressure. The available NPSH for a pump in an open system of this type is therefore equal to the atmospheric pres-

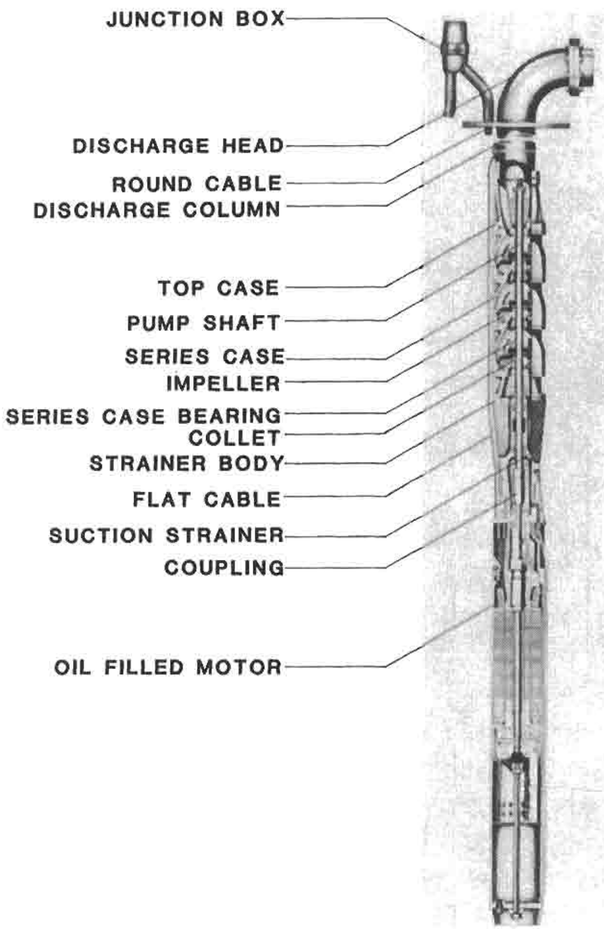

Figure 9-4. Submersible well pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

sure, plus the static liquid level above the first-stage impeller, less correction for the liquid vapor pressure at the pumping temperature.

Because the cost of a pit or intake structure is high and dependent on the depth of the structure, the submergence is typically kept to a minimum in line with sound design practices. As a result, the maximum pump speed is limited by the NPSH available and the required flow rate. Single-stage pumps can usually be furnished for pumping heads up to 200 feet, but multi-stage pumps are required for higher heads. The bowl diameters of well pumps and their corresponding flow rates are restricted. However, wet pit pumps can be furnished in any size and therefore for

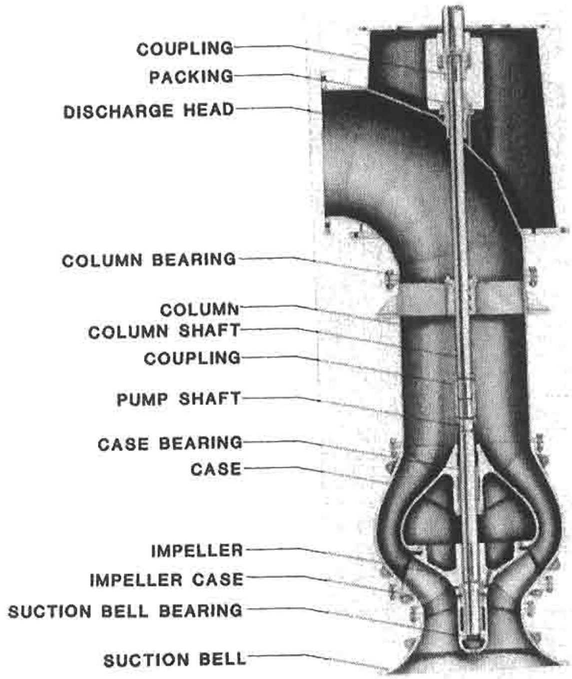

Figure 9-5. Wet pit pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

any desired flow rate (see Figure 9-6). Considerations, other than the pump itself, usually dictate that requirements for large flow rates be split between two or more pumps operating in parallel. The pump setting (the axial length of the bowl assembly plus the length of the discharge column from which the bowl assembly is suspended) is normally less than 100 feet. The column houses the lineshaft, which is connected to the driver shaft with a rigid coupling in the discharge head (see Figure 9-5). The discharge head also houses a shaft sealing device. The driver, which is supported on the top of the discharge head, is generally provided with a thrust bearing of adequate size to carry the weight of the motor rotor and pump rotating element plus the hydraulic axial thrust developed by the pump. When the driver is not designed to carry the total axial thrust from the pump, a thrust bearing assembly must be provided in the discharge head above the shaft sealing device. A flexible type coupling must then be provided between the pump and the driver.

While the discharge elbow is normally located in the head above the pump mounting floor, it may be advantageous for certain applications to

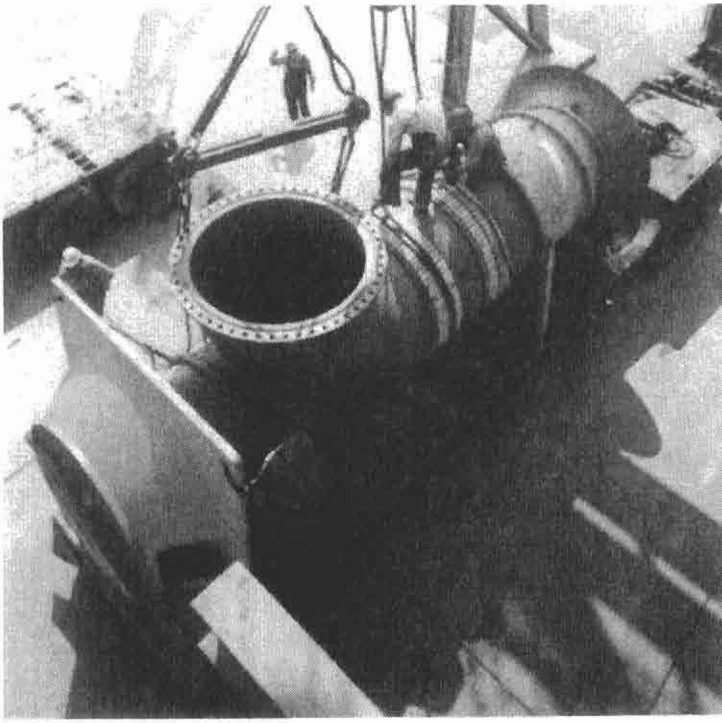

Figure 9-6. Installation of 100,000 GPM wet pit pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

locate the elbow on the discharge column below the mounting elevation. For this type of configuration, care must be taken by the piping designer to make sure that any horizontal expansion of the discharge pipe is contained and will not force the pump column with the lineshaft out of alignment. Furthermore, the column must be free to elongate axially at the discharge elbow when filled with water. This can be accomplished by locating a flexible coupling in the discharge pipe near the discharge elbow.

Wet pit pumps are usually driven by a vertical, solid shaft induction or synchronous motor as depicted in Figure 9-7. However, horizontal drivers that transmit torque through a right angle gear mounted on the discharge head may also be used. For applications where a wide variation in flow and/or head requirements exist, a variable speed driver may be used. Eddy current clutches or variable frequency electric drives are often used, the latter being the more efficient of the two.

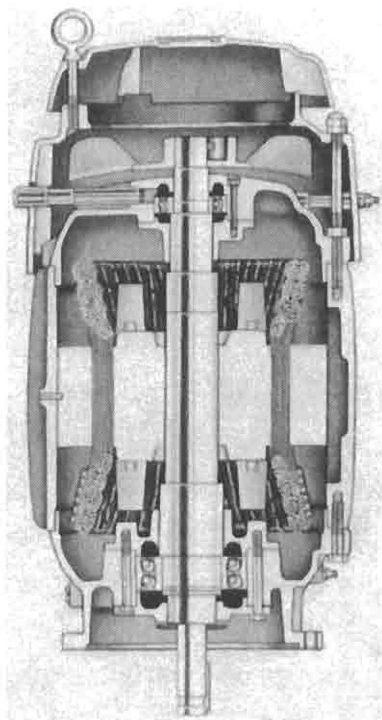

Figure 9-7. Vertical solid shaft motor (courtesy U.S. Electrical Motors, Division of Emerson Electric Co.).

Barrel-Mounted or Can-Mounted Pumps

This pump type, as depicted in Figure 9-8, is mounted in a suction barrel or can that is filled with liquid from the suction source. In this type of closed system, the NPSH available is not related to the atmospheric pressure. It is a function of the absolute pressure (above absolute vacuum) at the centerline of the suction flange, plus the liquid head from the center line of the suction to the first-stage impeller, less barrel losses, and less the vapor pressure of the pumped liquid. The available NPSH can therefore be increased by lowering the elevation of the first-stage impeller and extending the suction barrel until the available NPSH meets or exceeds the required NPSH. This feature is in fact frequently the reason for selecting a vertical, barrel-mounted pump. To achieve the same result with a horizontal pump would require lowering the entire unit at great ex-

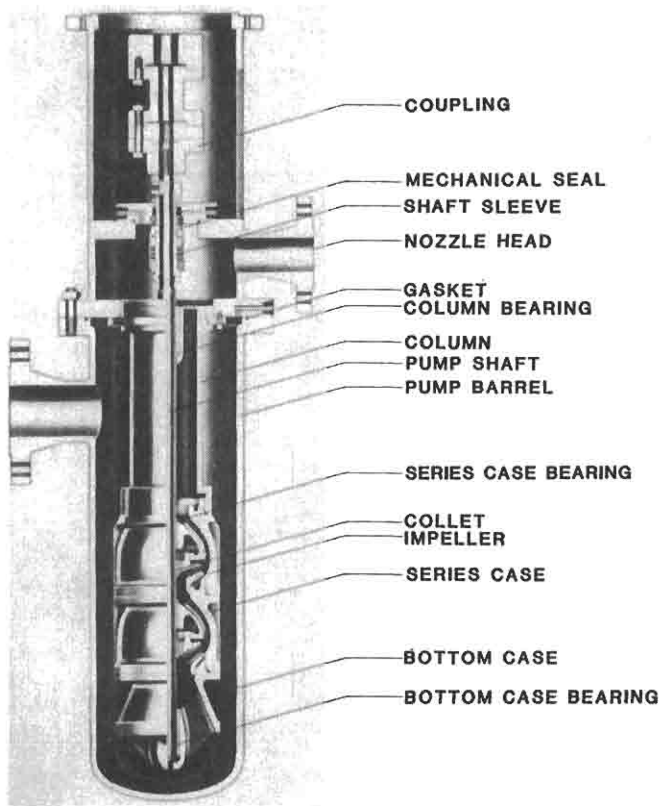

Figure 9-8. Barrel-mounted pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

pense. Often a special first-stage impeller with superior NPSH characteristics is furnished. Otherwise, the bowl assembly is of a multi-stage design with identical impellers of the radial or semi-radial flow type. The bowl assembly is either directly suspended from the discharge head or connected to the head with a discharge spool, the length of which is determined by the NPSH required.

The configuration shown in Figure 9-9 is of the pull-out type that permits removal of the pump without disturbing either the discharge or suction nozzle connections. Pump alignment is, within reason, not affected by any nozzle forces imposed. The suction nozzle can be located either in the suction barrel or in the discharge head, in line with the discharge nozzle. The latter "in-line" construction is commonly used for booster applications in pipelines. The shaft sealing device in the discharge head is usu-

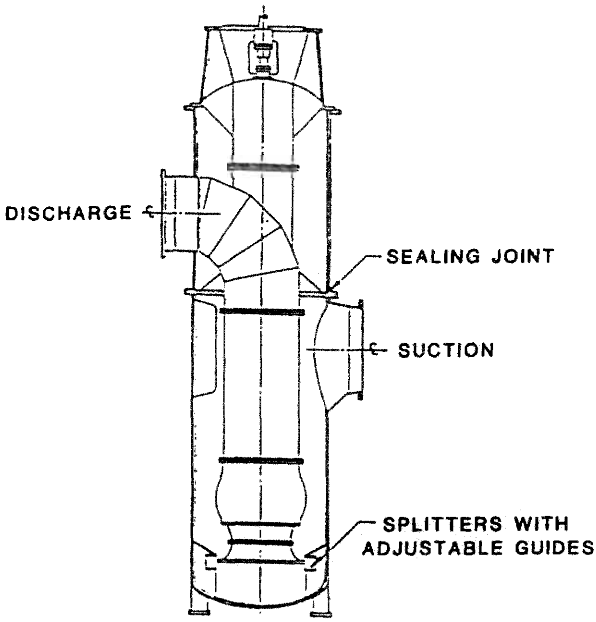

Figure 9-9. Barrel-mounted pull-out pump.

ally a face-type mechanical seal. The discharge head supports the driver, which should preferably be of the solid shaft design, either as a direct electric motor drive or a horizontal driver through a right angle gear. The unit is typically supported under the top flange of the suction barrel and bolted to an adequate foundation with a desired mass of five times the total unit weight. If desired, the entire barrel can be embedded in concrete or thermally insulated for high or low temperature applications.

Applications

Well Pumps

The most common applications for well pumps are:

- Water well, or bore hole installations, using either a surface-mounted driver or a close-coupled submersible motor.
- Incline-mounted water pumps installed on lake or river banks and driven by a conventional electric motor or a close-coupled submersible motor.

- Loading pumps in underground caverns used for storing petroleum products. The pump is mounted in a caisson.
- Dewatering pumps in mines. The pump is mounted in a mine shaft.

Water Well Pumps

This is the most common application and covers a broad range of services such as municipal water supply, irrigation service, and industrial service water. For settings down to 400 feet, a line shaft construction with a surface mounted driver is normally used. It should be noted that line shaft construction requires that the well be straight, so that the column shaft bearings can be kept in alignment. The well must be checked for this purpose prior to pump installation. For settings beyond 400 feet, and where electric power is available, the close-coupled submersible unit is usually the most cost-effective and also the most reliable. The close-coupled design often permits running at higher rotative speeds. With elimination of line shafting and corresponding bearings, well straightness is not as important. However, the well should be "caged" (checked with a dummy pump/motor assembly) prior to pump installation to make sure the unit will not bind in "dog legs."

Incline-Mounted Pumps

The cost of excavating and providing an adequate intake and mounting structure for vertical pumps on lakes and river banks can be substantial, particularly where large fluctuations in water level require high structures. A less costly installation may be achieved by mounting vertical pumps on an incline on a lake or river bank, as shown in Figure 9-10. The pump is mounted inside a pipe or in a trough, permanently anchored on piers along the bank with the bowl assembly at a sufficient depth to provide adequate submergence. The pump can either be of the line shaft type with an electric motor drive or close coupled to a submersible motor, in which case both the pump and motor are mounted in the pipe or on the trough.

Cavern Pumps

For ecological as well as safety and economic reasons, petroleum products, ranging from propane to crude oil, are often stored in natural or man-made caverns rather than in large surface tanks [7]. Well type pumps, most commonly driven by submersible motors, are used for unloading the cavern before the product is further transported by pipeline or ship. The pumps are usually mounted in a caisson, which is sealed at the top and terminates near

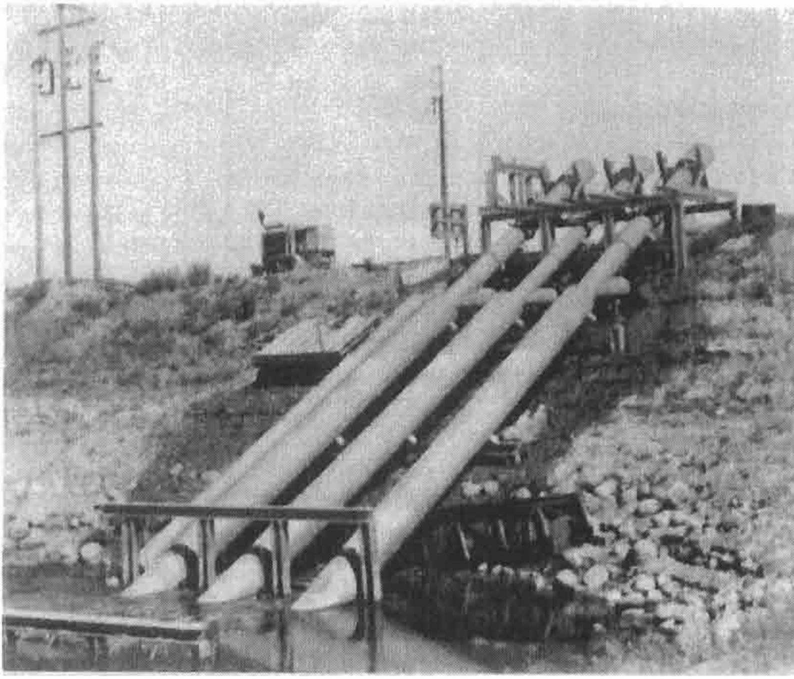

Figure 9-10. Incline-mounted line shaft pump.

the bottom of the cavern, where the pumps take suction. During maintenance, water is let into the cavern, and with the petroleum floating on top of the water, the bottom of the caisson is sealed off with a water "plug," preventing undesirable gases from escaping. The water level in the cavern can be maintained with separate pumps.

Mine Dewatering Pumps

Vertical well pumps are often preferred for mine dewatering. All sensitive electrical equipment, including control panels, can be located well above levels where accidental flooding might occur or where explosive gasses may be present. This includes both conventional motor driven pumps and submersible motor pumps. The pump may be installed in an open mine shaft, or a separate well may be sunk for the purpose of dewatering. The mechanical construction of the pump is similar to a water well pump.

Wet Pit Pumps

The most common applications for wet pit pumps are:

- Water supply pumps for municipalities and industry. The pumps are mounted in intake structures on lakes or rivers.
- Condenser cooling water pumps for central power plants. The pumps take suction from a natural body of either fresh or salt water.
- Cooling tower pumps. Take suction from a cooling tower basin and circulate water through a closed system.
- Flood control pumps mounted at dams and in collection basins, often as part of large flood control systems.
- Transfer pumps for central irrigation districts and water treatment facilities.

Water Supply Pumps

This pump type is normally installed as multiple, parallel operating units in a simple intake structure or as a stand-alone pumping plant located on a reservoir, lake, or river and discharging into a pipeline or an open canal. Depending on the system requirements, multi-stage pumps or single-stage pumps of the desired specific speed are used. A combination of fixed and variable speed drivers may be desirable to obtain optimum system efficiency. While structural integrity and cost are critical items in design of intake structures, hydraulic considerations and protection of the pumping equipment are equally important. The structure should be physically located so that a minimum of debris and silt will be diverted toward it. Trash racks and rotating screens, which can routinely be cleaned, must be provided to keep foreign objects from entering the pumps. The intake structure must be designed for low approach velocity and with dividing walls forming individual bays as required [5]. The pumps should be located within the structure in such a fashion that uniform velocity distribution is provided at each pump suction bell. Obstructions, changes in flow direction, or velocity changes that may cause formation of vortices and air entrainment must be avoided. Flow patterns within the structure, when one or more pumps are idle, must also be considered. Pump settings, the distance from the mounting floor to the suction intake, typically vary from 15 feet to 80 feet. The discharge may be located above or below the mounting floor, depending on the system requirement. In either case, the discharge pipe should be anchored downstream from the pump discharge flange to prevent pump misalignment from pipe reaction forces. When a flexible discharge piping connector is used, tie bars must be provided across the connector to restrain the hydraulic separating forces and prevent pump misalignment. Figure 9-11

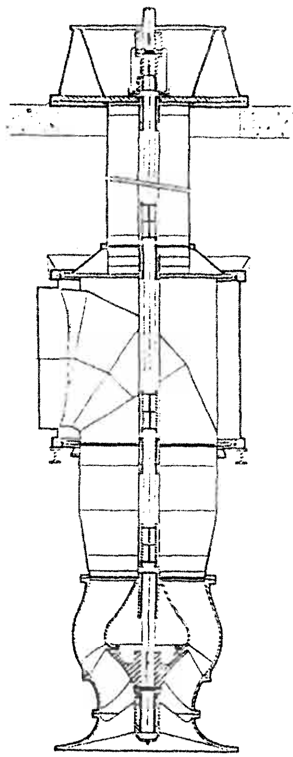

Figure 9-11. Water supply pump of pull-out design with below ground discharge.

shows a below ground discharge application with pull-out construction. This construction is particularly well suited for locations where the discharge is located below the water level because the nozzle can be permanently welded to the discharge piping system. Solid shaft electric motors are most commonly used as drivers.

Condenser Cooling Water Pumps

Installation considerations, configuration, and driver requirements for these pumps are in general the same as for water supply pumps. However, in addition, the following must be considered:

- When selecting lubrication method for the column bearings, neither oil nor grease can be used, because small amounts of hydrocarbon entering the condenser will impair the heat-transfer properties.

- Fresh water bearing lubrication is preferred, but salt water injection is acceptable as long as it is filtered and continuous. If a pump sits idle for extended periods without injection, accelerated corrosion may take place as well as the build-up of harmful crystals and marine life.
- Care must be taken in the selection of base materials for salt water. Consideration should be given to the average temperature of the water and the potential absence of oxygen.

Cooling Tower Pumps

These pumps typically have short settings, less than 20 feet, operate against a fixed head, and are connected to fixed speed drivers. The cooling tower basin and associated sump for pump installation are usually very limited in depth and area, requiring specific precautions to avoid vortices and ensure uniform velocity distribution [5]. Model structure testing is recommended where design margins are small. Water quality tends to become questionable in this type of closed system, warranting caution in material selection.

Flood Control Pumps

These are typically low lift, short setting pumps, but vary greatly in size depending on historic demands. A 2,000 to 3,000 gpm sump pump may be adequate for protecting a small area, while a large flood control district may require multiple pumps of several hundred-thousand gpm capacity each. Such large pumps can partly be formed in concrete at the site. This can be a major cost advantage, particularly if the water quality demands high alloy metals. Figure 9-12 shows a propeller pump where the suction bell, column, and discharge elbow have been formed in concrete.

Pump efficiency is not the primary consideration for flood control pumps because operating time and therefore power consumption is limited. Reliability is the main concern, and the pumps must be capable of handling large amounts of silt and sand. Serious flood conditions may be connected with loss of electric power, and flood control pumps are therefore often driven by diesel engines through right angle gear drives.

Transfer Pumps

This general category of pumps covers a wide variety of applications, from highly efficient central irrigation pumps and canal lift pumps, to simple, non-clog pumps in sewage and water treatment plants. Commonality in design is therefore minimal. When efficiency is one of the

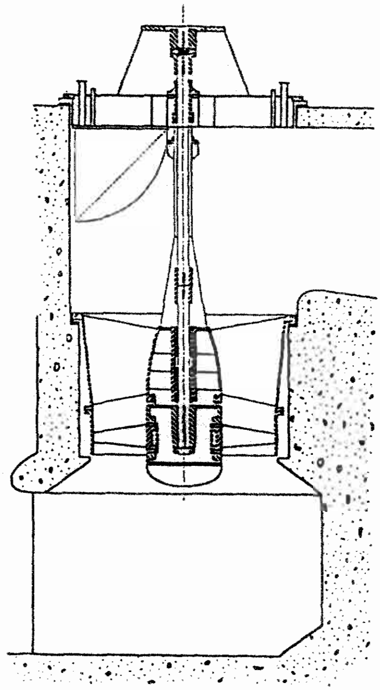

Figure 9-12. Propeller pump partly formed in concrete.

primary design criteria, the design is generally the same as for cooling tower pumps. On the other hand, non-clog pumps are designed for maximum reliability and availability. The impeller is typically of semi-open construction with two or three vanes and contoured to prevent adherence of stringy material. Where suspended solids are a problem, provision for clean water injection to the bearings can be provided. Handhole covers are provided at locations where buildup of solids will require removal.

Barrel-Mounted or Can-Mounted Pumps

The most common applications for barrel-mounted pumps are:

- Condensate and heater drain pumps for power plant service.
- Process pumps for products with limited NPSH available.
- Small boiler feed pumps for industrial applications.
- Cryogenic process and transfer pumps.
- Loading pumps on tank farms.
- Booster pumps for pipelines handling either water or petroleum products.

Condensate and Heater Drain Pumps

This pump type is typically installed as two 50% capacity pumps taking suction from a header connected to a condenser or heater for boiler feed water. The available NPSH is normally only two to four feet at the mounting floor, requiring additional NPSH to be built into the barrel. The multi-stage bowl assembly, typically in the 1,500 to 2,500 specific speed range, can be fitted with a special first-stage impeller to meet the required NPSH. Figure 9-13 shows a unit with a double suction first-stage impeller. The suction nozzle may be located either in the barrel or the discharge head, whichever the user prefers. The shaft seal in the discharge head is typically of the mechanical face type and must be water quenched because the seal is under vacuum when on standby. A continuous vent line must be provided from the top of suction side in the pump to the vapor phase in the suction tank (condenser). A minimum flow bypass line may be required at the discharge control valve if extended low flow operation cannot be avoided. Induction motor drive is the most common, but a variable speed drive offers advantages for peak loaded plants. Condensate pumps normally operate in the 130°F range, and cast iron bowls with bronze impellers and bearings are usually adequate. For applications where the peripheral vane velocity in the suction eye exceeds 80 feet per second, a stainless steel impeller should be used. Some users will not permit bronze materials in the system because it may contribute to corrosive attack on condenser tube welds. In these cases, all impellers should be furnished in martensitic steel. Heater drain pumps may operate up to 350°F and require impellers of martensitic steel and bearings of a carbon-graphite composite. Because flashing in the first stage cannot always be avoided in this service, injecting second-stage pump pressure into the suction case bearing is recommended.

Process Pumps

These pumps are of multi-stage construction, with a special first-stage impeller to meet the limited available NPSH. Handling liquids near their boiling point requires a continuous vent line from the pump suction side back to the suction source. The mechanical shaft seal can either be mounted internally in the discharge stream, flushed and cooled by the pumped liquid, or mounted in an external, water jacketed stuffing box for high temperature applications. Materials for the bowl assembly and fabricated components are selected to suit the liquids handled, including cavitation resistant material for the first-stage impeller, when applicable.

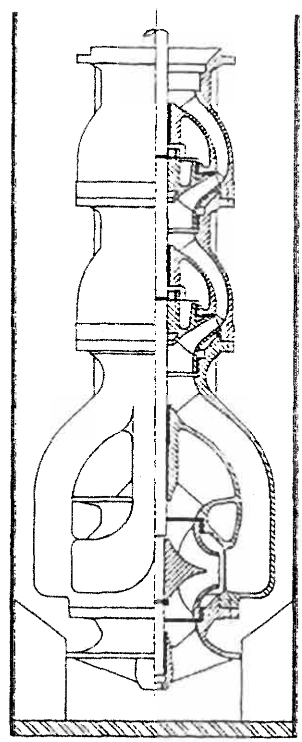

Figure 9-13. Bowl assembly with double suction first stage.

Small Boiler Feed Pumps

In design, these pumps are quite similar to heater drain pumps, although the NPSH margin is usually sufficient not to require injection to the suction case bearing. The mechanical seal should be located in a stuffing box, mounted externally on the discharge head. The minimum recommended material selection is cast iron bowls, martensitic steel impellers, and carbon/graphite bearings.

Cryogenic Pumps

Vertical barrel pumps are particularly well suited for cryogenic applications. Being vertically suspended, thermal contraction and expansion will not cause pump misalignment as long as reasonable precaution is taken in dealing with nozzle forces at the suction and discharge flanges. Because the motor is supported on top of the discharge head, it is automatically aligned to the pump. The external pump configuration is simple and easy to thermally insulate with jackets or in a "cold box." A thermal barrier, also known as a "warming box," with a throttle bushing and a double mechanical seal or a gas shaft seal, is located in the discharge head, just above the discharge nozzle. Here the cryogenic liquid is flashed and bled back to the suction source, while an inert gas blanket under the shaft seal prevents leakage to atmosphere. Depending on the liquid pumped, pump materials with adequate impact strength are bronzes, aluminum, and austenitic stainless steels. Materials with similar thermal coefficient of expansion must be used where tolerances are critical.

Loading Pumps

Normally installed immediately adjacent to large storage tanks used for loading product into pipelines or transport vessels, these pumps are mounted so that the storage tank can be emptied, even when the available NPSH becomes zero at the bottom of the tank. Adequate provisions must be made for venting the suction barrel and providing a minimum flow bypass when applicable.

Pipeline Booster Pumps

These pumps typically operate unattended and must be designed for reliability. The discharge and suction nozzle should both be located in the discharge head for simplicity in piping and valve placement. When handling liquids where leakage to the atmosphere is hazardous, a tandem or double mechanical shaft seal with a buffer fluid should be provided for the stuffing box. Pumps can be arranged to operate singly, in parallel, or in series.

Design Features

For comparison and evaluation of design features, the three basic assemblies of vertical pumps should be addressed, namely the bowl assembly, the column assembly, and the head assembly. (See Figure 9-14).

The Bowl Assembly

The simplest bowl assembly configuration consists of a straight shaft with taper collet mounted impellers, bowls that are joined together with straight threads and furnished with a shrink fitted bearing (Figure 9-2). The impellers can be either of the enclosed design, with both a front and a back shroud, or the semi-open design without a front shroud (see Figure 9-15). The bottom case bearing is normally permanently grease lu-

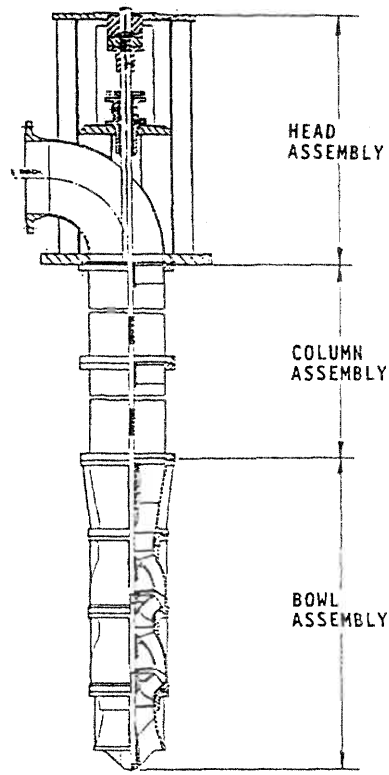

Figure 9-14. Vertical pump assemblies (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

bricated and the other bearings lubricated by the pumped liquid. This design lends itself well to smaller pumps with up to 18-inch bowl diameter and 2-inch shaft diameter. However, the following limitations must be noted:

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

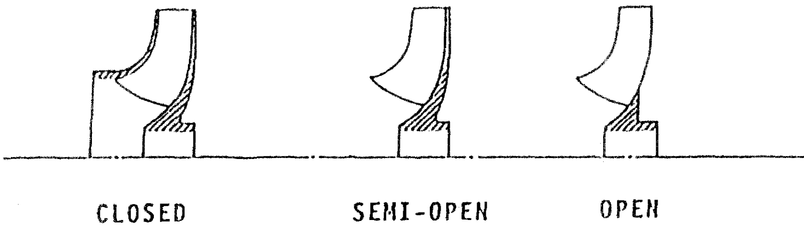

Figure 9-15. Impeller configurations.

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

- The bowl joints should be flanged and bolted. Gaskets, when required, should be of the "O"-ring type, so that joints are made up metal to metal, and pump alignment therefore maintained.
- The impellers should be mounted on the shaft with key drives and secured axially with split rings and thrust collars.
- When selecting closed vs. semi-open impellers, the following must be noted:
  - Closed impellers should always be used for handling hot or cryogenic liquids.
  - Closed impellers exhibit lower downthrust when in the axially unbalanced configuration.
  - Closed impellers are easier to assemble for large pumps with more than three stages.
  - Semi-open impellers are more efficient due to elimination of disc friction from the front shroud.
  - Efficiency loss due to wear on the semi-open impeller vanes can be regained by adjusting the impeller setting at the adjustable pump to driver coupling, typically in increments of 0.016 inch.
  - When semi-open impellers are axially balanced to reduce axial thrust, downthrust can be maintained over the full operating range. This prevents shaft whip from upthrust with associated bearing wear.
  - Semi-open impellers can readily be hardsurfaced for erosion protection.
  - Semi-open impellers are less likely to seize when handling sand or foreign material.

The Column Assembly

The column assembly consists of three primary components:

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

Outer Column

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

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

Column Shaft

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

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

Shaft Enclosing Tube

When abrasives or corrosive properties prohibit the pumped liquid from being used for flushing and lubricating the column bearings, the bearings should be fitted inside a shaft enclosing tube. The bearings, typically of bronze material, are threaded on the O.D. and serve as joiners for the five-feet long enclosing tube sections. Bearing alignment is provided by placing the enclosing tube assembly in tension through a threaded tensioning device located in the discharge head. The enclosing tube is stabilized within the outer column by random placement of hard rubber spiders, the hub of which fits tightly around the enclosing tube and the three legs fit tightly against the inside of the outer column. The desired bearing lubrication, which can be oil, grease, clean water, or any fluid compatible with the pumped liquid, is injected at the top end of the enclosing tube assembly.

To overcome the mounting number of assembly and handling problems that can occur with increase in column size, an enclosing tube integrally welded with ribs at the top and bottom of the outer column is a preferred design. Alignment is assured with simultaneous machining of the registered column fits, the inner column joints, with slip fit and "O"-rings, and the bearing seats. Furthermore, the need for a tensioning device in the discharge head is eliminated. The result is that both assembly and disassembly time for the pump is significantly reduced.

The Head Assembly

The discharge head is designed to serve the following functions:

- Support the suspended, liquid-filled weight of the pumping unit.
- Provide support for the driver.

- Incorporate a discharge nozzle to guide the flow from the outer column to the system pipe. For barrel-mounted pumps, a suction nozzle may also be located in the discharge head.

The discharge head must house a shaft sealing device suitable for the maximum pressure the pump can be subjected to. The sealing device is located in a stuffing box that can be placed either in the discharge stream for flushing or mounted externally for cooling and flushing. The actual sealing can be done with packing or a mechanical face seal. A pressure breakdown bushing, with bleed-back to pump suction, can also be included in the sealing device for high pressure applications.

The standard drive coupling for vertical pumps with solid shaft drivers is of a rigid design, capable of transmitting the maximum torque from the driver and the combined axial force from hydraulic thrust plus rotating element weight. The coupling typically incorporates a disc threaded on to the top end of the column shaft, clamped between the two coupling halves, that permits adjustment of the impeller setting within the bowls (see Figure 9-14). For drivers with limited thrust carrying capability, a thrust bearing must be incorporated into the discharge head design.

For pumps using hollow shaft drivers, torque is transmitted to the top column shaft or head shaft through a keyed clutch at the top of the motor, and impeller adjustment is made by a nut seated on top of the clutch (see Figure 9-3).

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

Pump Vibration

The vibration pattern of a vertical pump is an inherent characteristic of its configuration, manufacture, and physical condition. Vibration results from factors such as rotating element unbalance, misalignment, looseness in the assembly, bent shafting, or bad driver bearings. Also, the operating parameters and the installation, including the rigidity of the supporting structure and attached piping, have a strong influence. The latter may cause vibration from sources such as hydraulic resonance in piping, turbulence at the pump intake, cavitation problems, low flow recirculation, and structural resonance in the pump/driver assembly.

The availability of data collectors and matching computer hardware and software has greatly facilitated collecting and analyzing vibration data. The establishment of vibration signatures is not only a means of verifying the satisfactory condition of a new installation, but can also serve as the basis for scheduling pump maintenance.

Measurement of axial and lateral vibration on the pump and driver is measured either as absolute movement on driver bearing housings or as relative movement between the shaft and the bearing housing or pump structure. Torsional vibration is seldom a problem in vertical pumps because the exciting force generated by the rotating impeller vanes passing the stationary bowl vanes is small. However, for applications where right angle gears and engines are used, exciting forces can be generated that may cause damaging torsional vibration. When these types of drivers are used, an analysis for torsional critical frequencies should be performed at the design stage. The computer models for performing these analyses are quite accurate and give good results. This subject is discussed in more detail in Chapter 18.

Figure 9-16 shows the desired locations for taking vibration measurements. The axial reading is taken as an absolute measurement directly on the motor thrust bearing housing. The lateral readings on the motor and discharge head are also taken as absolute measurements and should be taken in line with, and at right angle to, the discharge nozzle. Measurements on the shaft should be taken as relative measurements, 90° apart, just above the stuffing box.

Absolute vibration measurements are taken with velocity transducers or accelerometers. Accelerometers should either be permanently attached or attached with a magnetic base, while velocity transducers can be handheld. Velocity transducers and accelerometers are directional and must be installed with the base perpendicular to the desired direction of measurement. Relative vibration measurements are taken with proximity transducers. It should be noted that proximity transducers, due to their working principle, are sensitive to shaft material properties as well as surface finishes.

Because of the wide varieties and sizes of vertical pumps in use, the issue of acceptable vibration levels becomes rather complicated. However, both the Hydraulic Institute and the American Petroleum Institute have published acceptance criteria, specifically applicable to pumps, covering both overall vibration levels and filtered vibration, i.e., read-

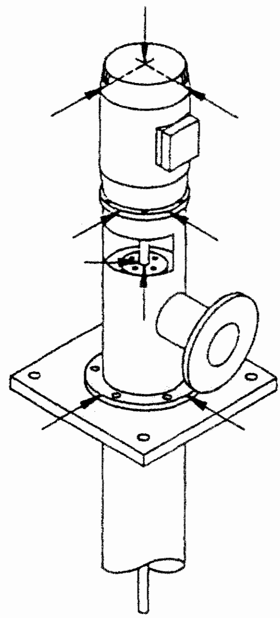

Figure 9-16. Locations for taking vibration measurements.

ings taken at discreet frequencies. Caution should be used in applying severity charts published for general machinery.

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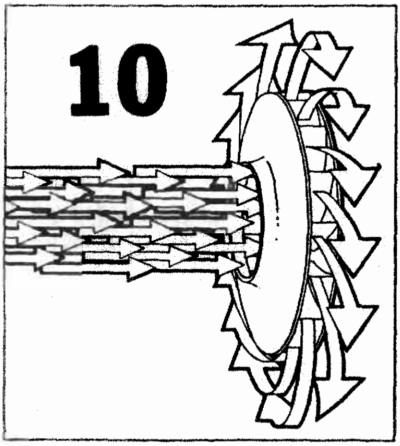


10 Pipeline, Waterflood, and CO_2 Pumps

Pipeline Pumps

Unlike most other pump applications, pipelines constantly have changes in throughput and product. This is particularly true in the transportation of crude oil and hydrocarbons. This variation in liquid characteristics, throughput, and pressure can result in a wide range of system head curves, requiring extreme flexibility in pump operation. Selecting pumps can be complicated, usually requiring multiple pumps installed in series at each station. Selection may involve parallel operation, variable speed, and/or modifications to meet future requirements.

As pumping requirements must match pipeline characteristics, the first step in pump selection is analysis of the hydraulic gradient, and profile. This defines the length and elevation change of the pipeline and is used to establish pipeline pressure, pipeline horsepower, number of stations, number of pumps, and appropriate mode of operation. Pipe size is determined from throughput requirements and optimum investment, plus operating cost economics. Calculation of friction loss and static head establishes the pressure required to move throughput. With pipeline throughput and pressure known, pump efficiency can be estimated and pipeline horsepower calculated. The number of pumps required is then estimated by selecting the preferred driver size. Number of stations is estimated by the safe working pressure (S.W.P.) of the pipeline and the pressure required to move throughput.

$$\text{Pipeline HP} = \frac{\text{Pressure req'd (PSI)} \times \text{Throughput (BPD)}}{58{,}700 \times \text{Estimated pump efficiency}}$$

$$\text{No. of pumps} = \frac{\text{Pipeline HP}}{\text{Driver HP}}$$

$$\text{Min. no. of stations} = \frac{\text{Throughput pressure req'd}}{\text{S.W.P. of pipeline}}$$

Variable capacity requirements or horsepower limitations may dictate a need for multiple pumps. In this event it must be decided if the pump should operate in series or in parallel. With series operation, each pump delivers full throughput and generates part of the total station pressure.

Once pump conditions and mode of operation are clearly determined, pumps can be selected. With pumps currently evaluated competitively in excess of $1,000 per horsepower per year and pipelines operating up to 40,000 horsepower per station, it is essential that pumps be selected for optimum efficiency. This requires an understanding of the losses that occur inside a pump. These are:

- Friction losses.
- Shock losses at inlet to the impeller.
- Shock losses leaving the impeller.
- Shock losses during the conversion of mechanical power to velocity energy then to potential energy.
- Mechanical losses.
- Leakage losses at impeller rings and interstage bushings.
- Disc friction losses at the impeller shrouds.

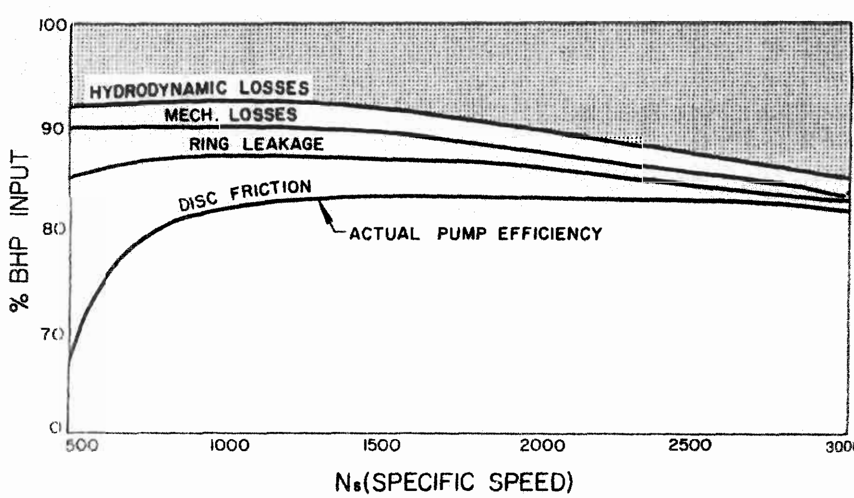
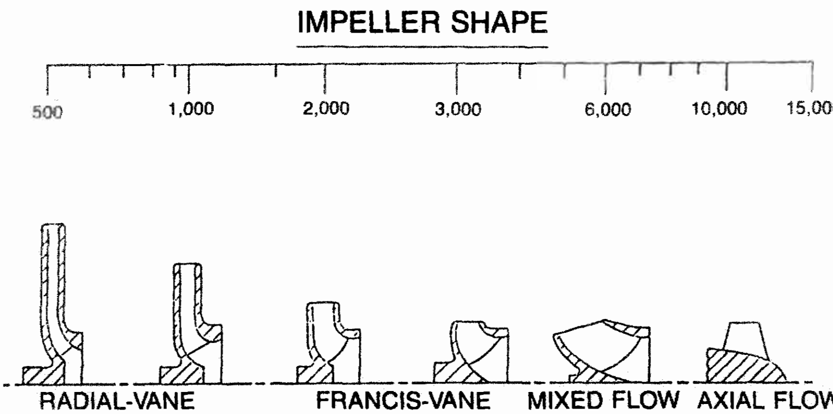


Figure 10-1. Typical analysis of pump losses.



$$\text{SPECIFIC SPEED } N_s = \frac{\text{RPM}\sqrt{\text{GPM}}}{H^{3/4}}$$

Figure 10-2. Specific speed describes impeller shape.

These losses can be generally classified as hydrodynamic, mechanical, ring leakage, and disc friction. Analysis of these losses for one specific performance at various speeds is shown in Figure 10-1. Pump efficiency is a result of the sum of these losses and is influenced by specific speed, which is basically a non-dimensional number. As discussed in Chapter 2, the physical meaning of specific speed has no practical value; however, it is an excellent means of modeling similar pumps and describes the shape of the impeller under discussion (Figure 10-2).

Pump efficiency is also influenced by hydrodynamic size (Figure 10-3). For any given speed, pump efficiency increases with size of pump or with hydrodynamic size (Figure 10-4). Through careful selection of pump speed and stage number, optimum specific speed and hydrodynamic size can be determined for maximum efficiency.

Condition Changes

Many pipeline conditions require low-capacity, low-pressure start-up with ultimate change over to high-capacity, high-pressure. By considering this requirement at the design stage, pumps can be built to accommodate the initial and ultimate conditions through field modifications. One method is to adjust the ratio of liquid velocity leaving the impeller to liq-

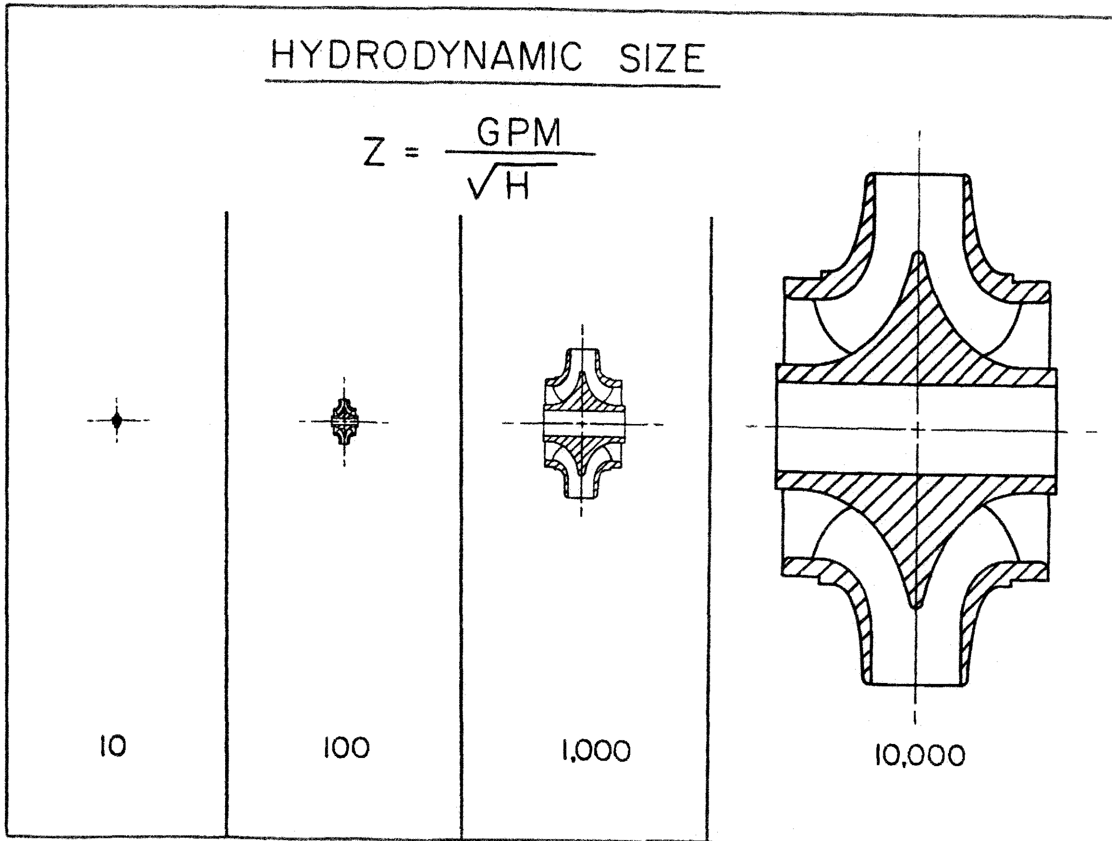


Figure 10-3. Pump size increases with hydrodynamic size.

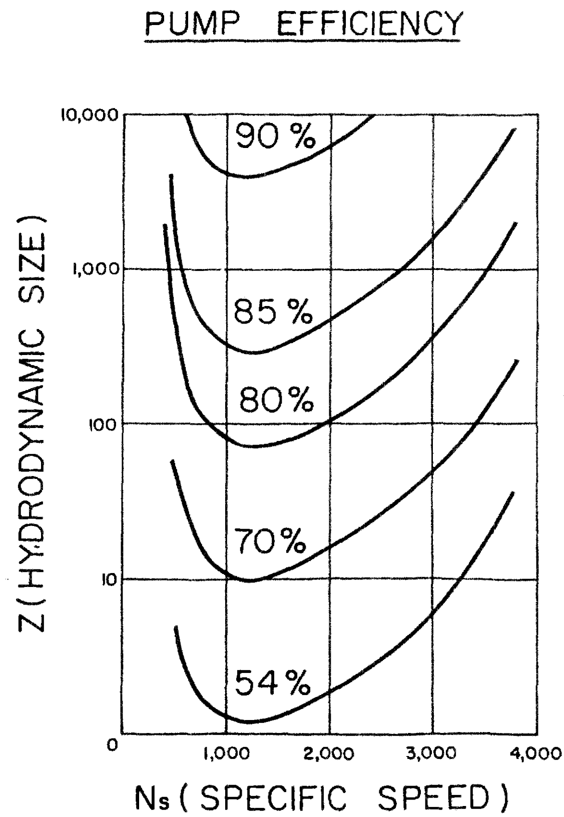


Figure 10-4. Efficiency increases with hydrodynamic size.

uid velocity entering the volute throat. By careful selection of this velocity ratio, pump throughput and optimum efficiency can be moved from the initial to the ultimate condition (Figure 10-5).

The volute throat velocity can be adjusted by cutting back (chipping) the stationary volute lip or lips to a predetermined dimension (Figure 10-6). This technique can be used on single- or double-volute pumps. Similarly, the velocity leaving the impeller can be adjusted by removing metal (underfiling) from the non-working side of the impeller blade (Figure 10-7). Through a combination of volute chipping, impeller underfiling, and ultimate installation of a high-capacity impeller, a wide variety of pump conditions all at optimum efficiency becomes possible (Figure 10-8).

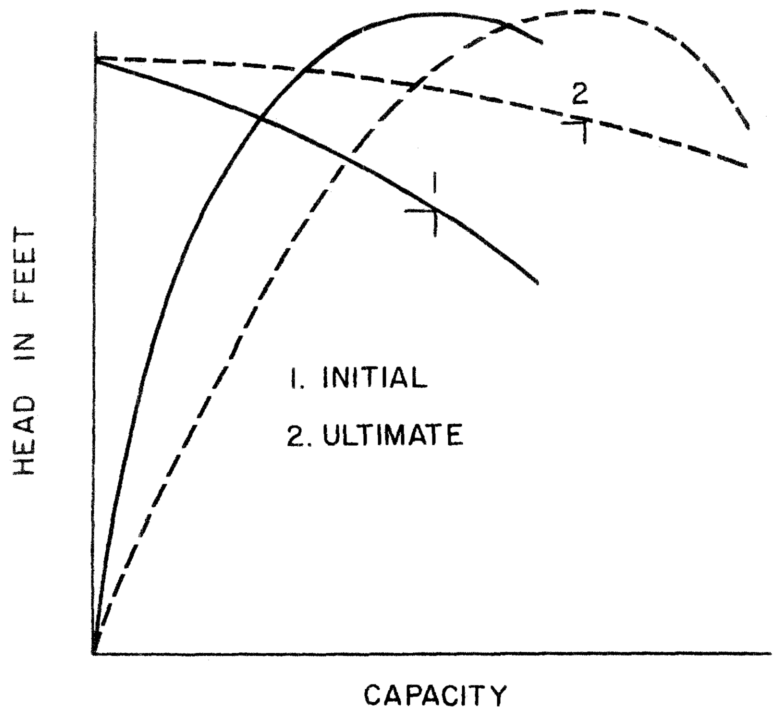
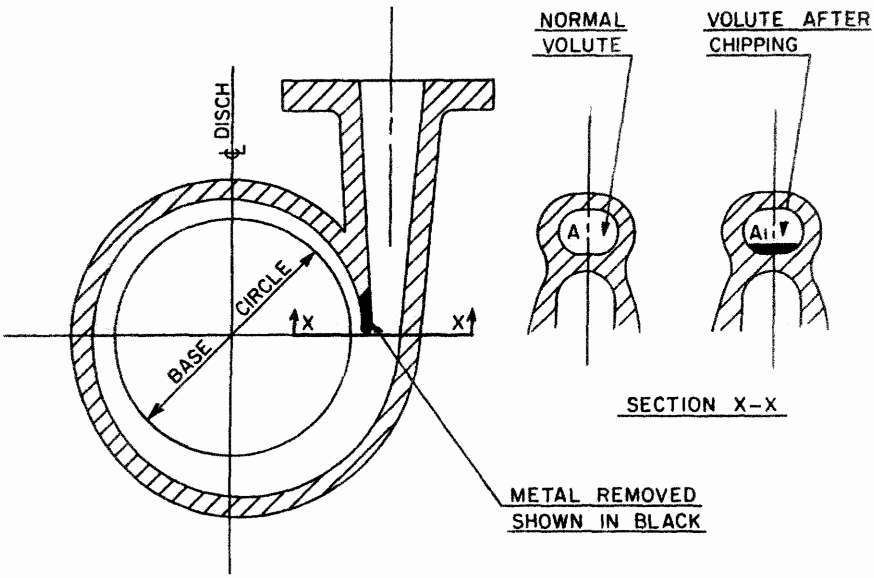


Figure 10-5. Pump performance for initial and ultimate condition.

Another method of satisfying two capacities at optimum efficiency is the installation of volute inserts. These removable pieces reduce the internal fluid passages and physically convert a high-capacity pump to low capacity (Figures 10-9 and 10-10). Inserts of this type are ideal for low-capacity field testing or reduced throughput operation with improved efficiency (Figures 10-11 and 10-12).

Destaging

It is not unusual for pipelines to start up at low pressure then later change to high pressure at constant throughput. This change in pump head requirements can be suitably handled by selecting a multi-stage pump for the ultimate high-pressure condition. For the initial low-pressure condition, appropriate impellers are removed and the interstage chambers isolated by destaging tubes (Figure 10-13). When changing to high pressure the destaging tubes are removed and additional impellers installed (Figure 10-14).



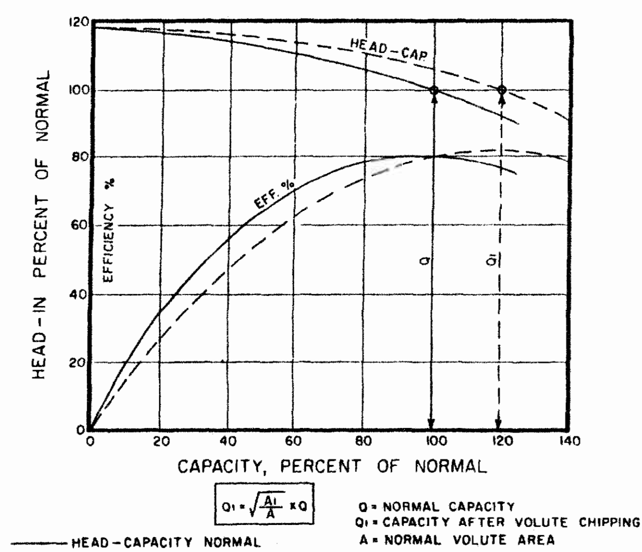
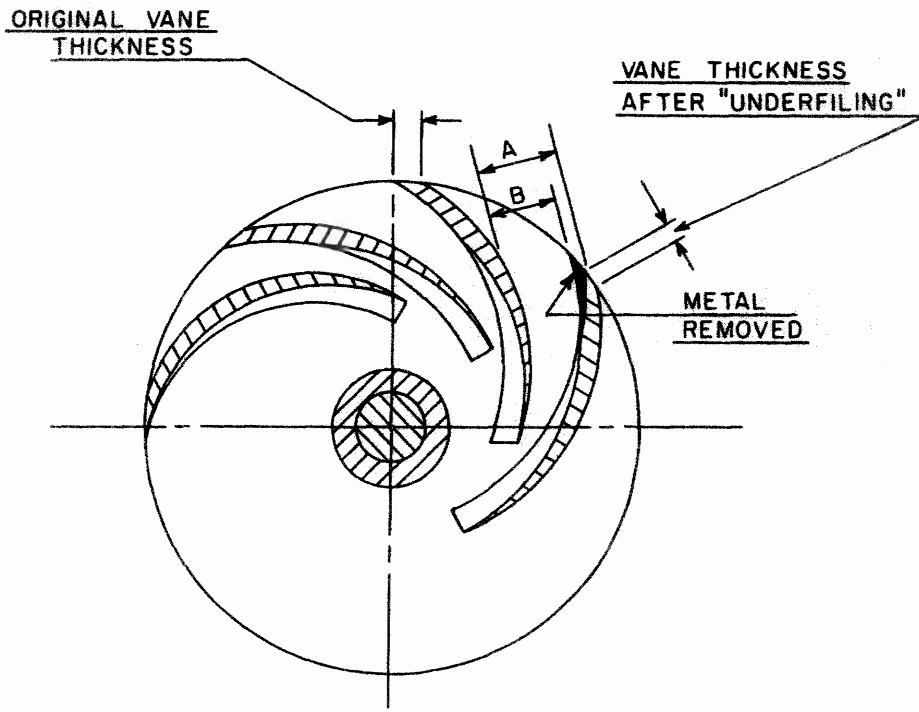


Figure 10-6. Changing pump performance by volute chipping.



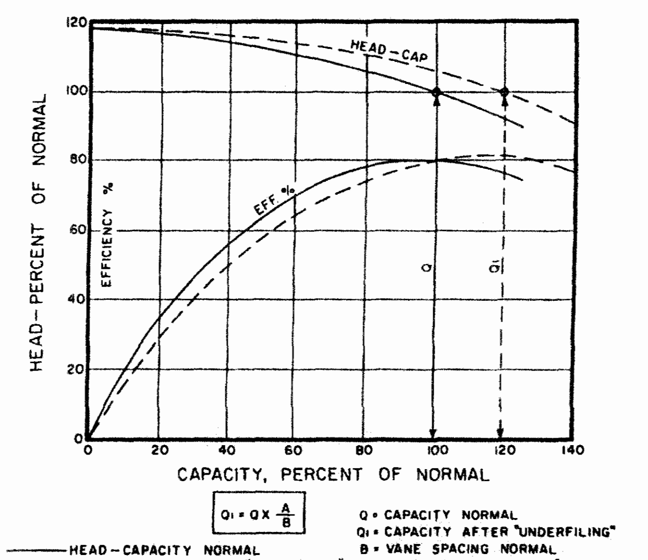


Figure 10-7. Changing pump performance by impeller underfiling.

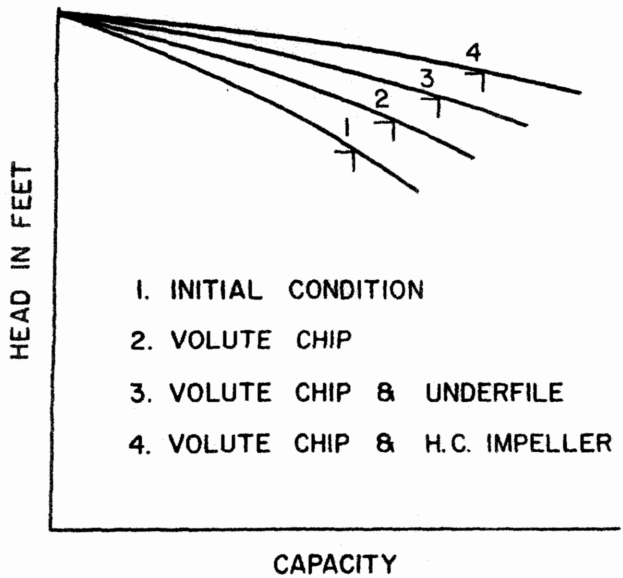


Figure 10-8. Performance range by volute chipping, impeller underfiling, and installation of high-capacity impeller.

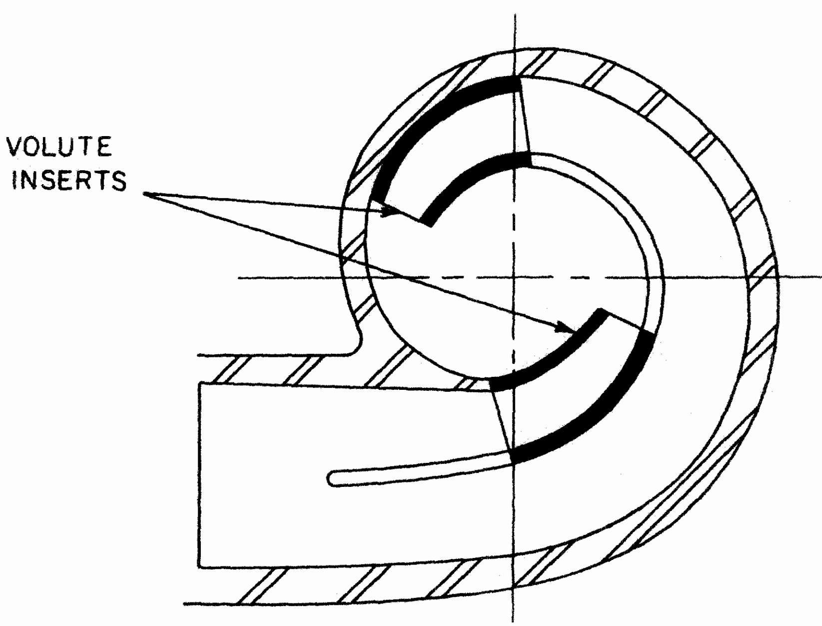


Figure 10-9. Volute inserts convert high-capacity pump to low capacity.

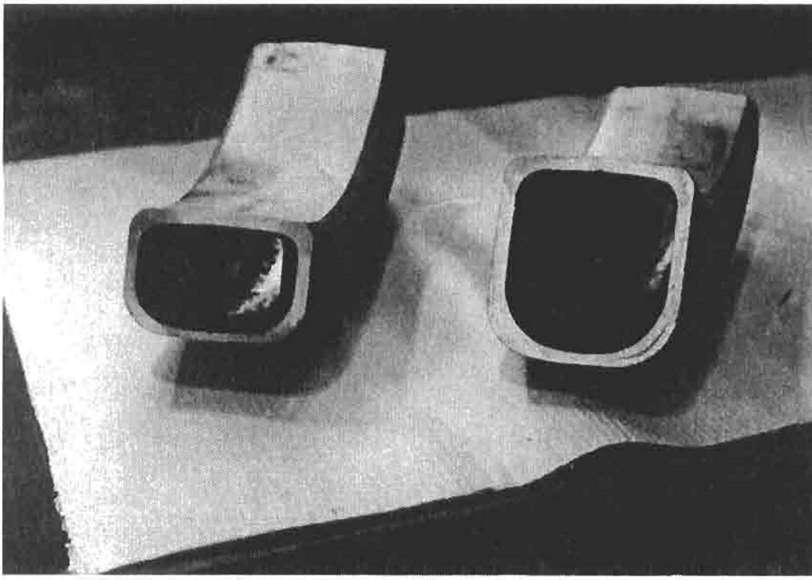


Figure 10-10. Volute inserts.

Bi-rotors

Unusual pipeline requirements can often be satisfied using bi-rotor pumps. These are basically two single-stage pumps on the same shaft, with two conventional incoming and outgoing nozzles at each body cavity. Originally developed with double-suction impellers to reduce NPSHR at high flow rates, this design is extremely flexible. Two applications are now described.

Bi-rotor pipeline pump. The pipeline conditions illustrated in Figure 10-15 have an ultimate pipeline capacity of 60,000 GPM requiring three pumps operating in parallel with each pump delivering 20,000 GPM at maximum pipeline pressure. During the early life of the pipeline, initial throughput would be 40,000 GPM. This would normally mean operating two pumps in parallel and throttling out excess pump pressure or reducing pump speed. A more economic approach would be to install a bi-rotor pump with case construction that permits operation of the impellers either in series or in parallel. The conventional, integral cast cross-over

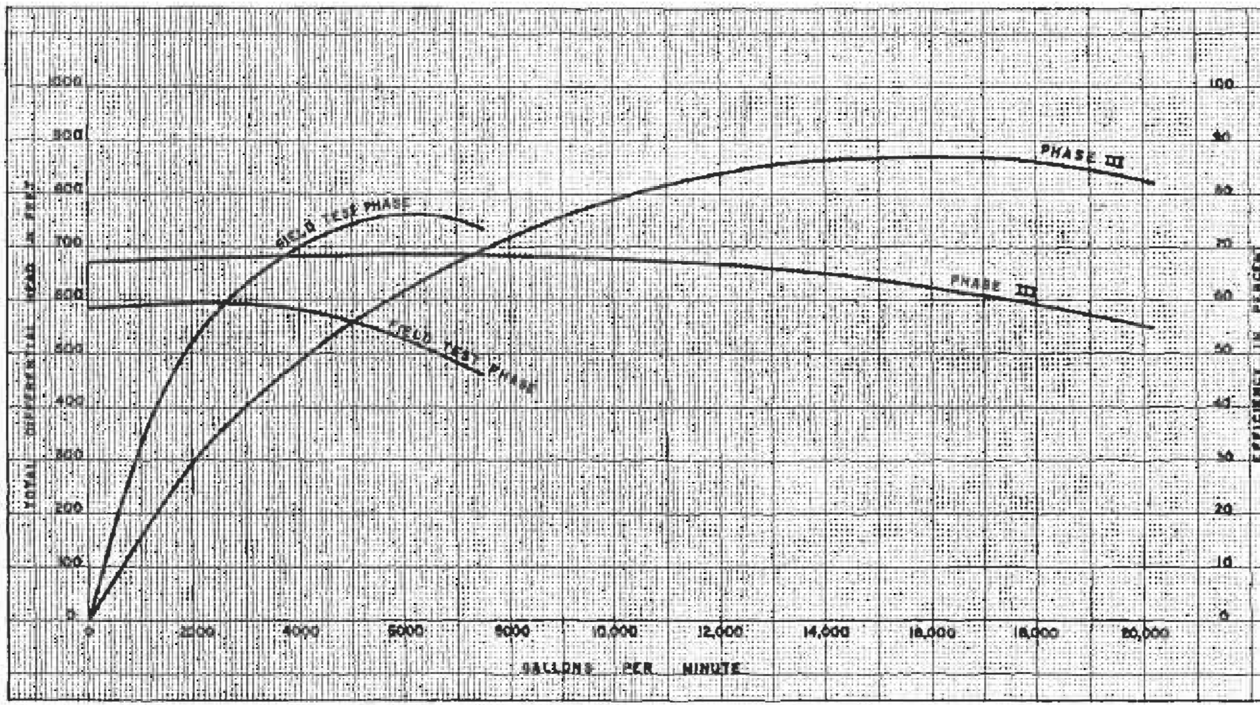


Figure 10-11. Performance change with volute inserts and low-capacity impeller.

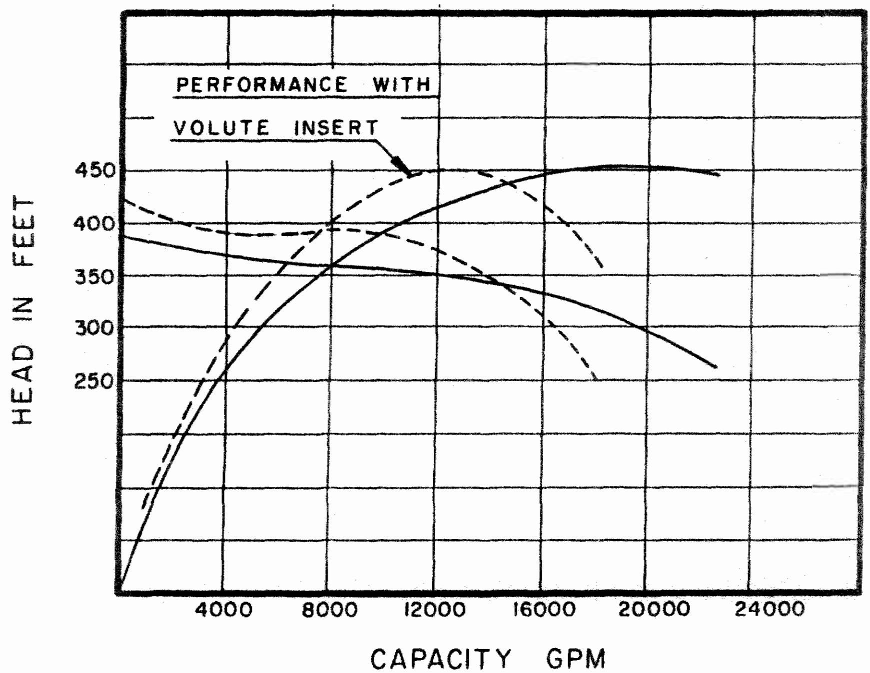


Figure 10-12. Performance change with volute inserts.

(Figure 10-16), which permits transfer of liquid from the first- to the second-stage impeller, is replaced by a bolted-on cross-over (Figure 10-17). With the cross-over bolted to the pump case, both impellers operate in series. With the cross-over removed, both impellers operate in parallel (Figures 10-18 and 10-19). With both impellers operating in parallel, the pump will deliver twice-normal pump capacity at half-normal pump pressure. In this configuration, initial throughput can be handled by one pump instead of two. For ultimate throughput, the cross-over is bolted in place and three pumps operate in parallel. Driver size is not affected, as the required horsepower is identical for either configuration.

Pulsation. The bolted-on cross-over has the additional benefit of permitting corrective action in the event of sympathetic acoustical frequency. All centrifugal pumps have a source of energy at blade passing frequency (Figure 10-20). Normal pressure pulsations generated by the pump can be magnified by system resonance when they are coincidental with fluid or mechanical natural frequencies within the system. These can be in the suction piping, discharge piping, or within the pump itself when the pump has more than one stage. Corrective action involves either relocat-

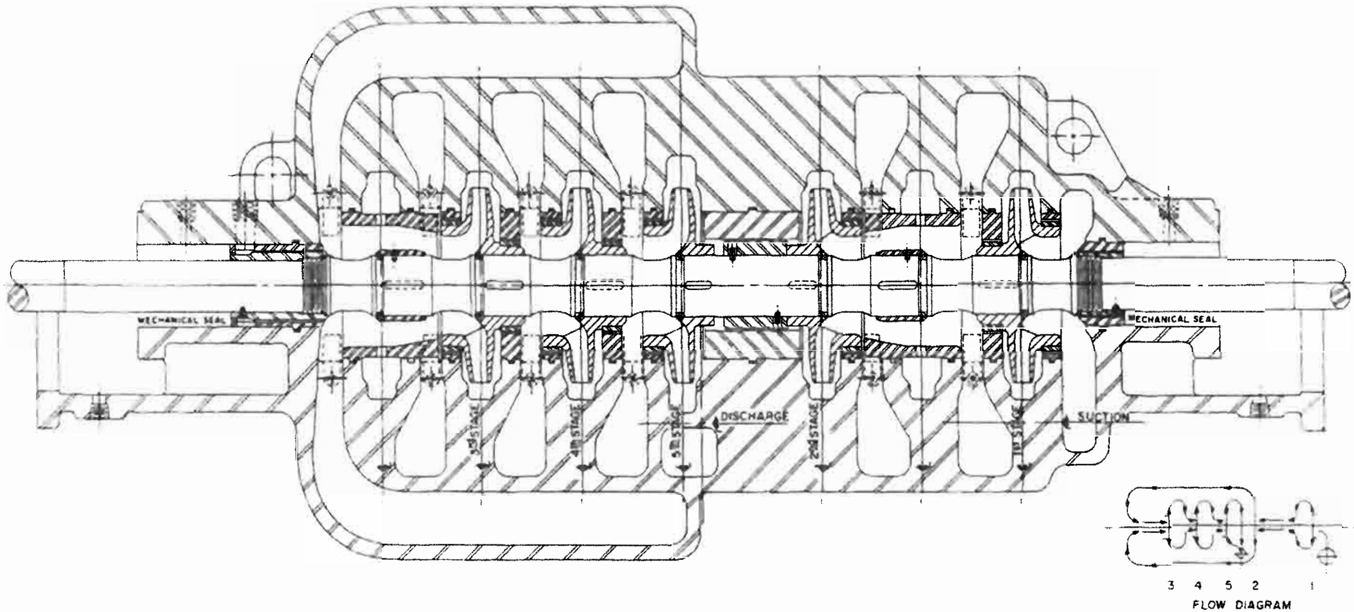


Figure 10-13. Seven-stage pump destaged to five stages.

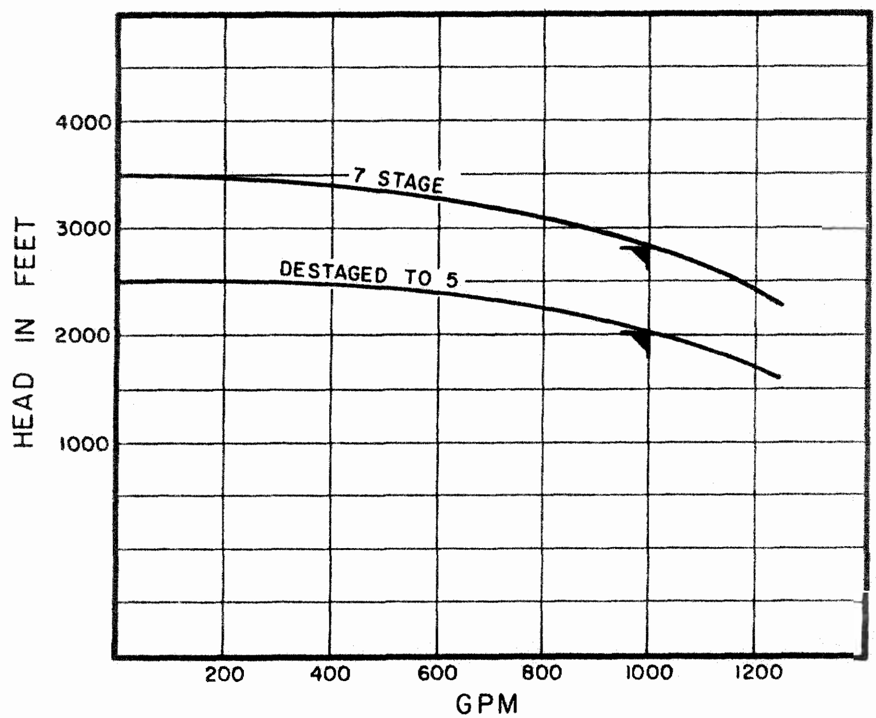


Figure 10-14. Pump performance before and after destaging.

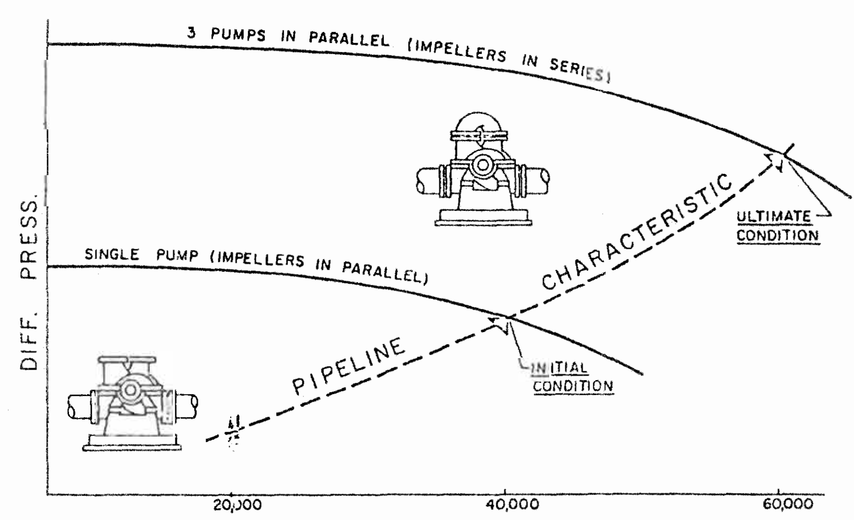


Figure 10-15. Pump configuration to satisfy initial and ultimate pipeline requirement.

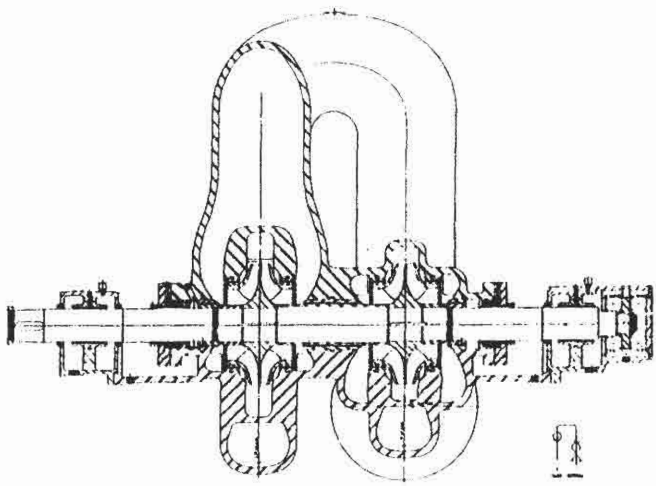


Figure 10-16. Two-stage pipeline pump with integral cast crossover.

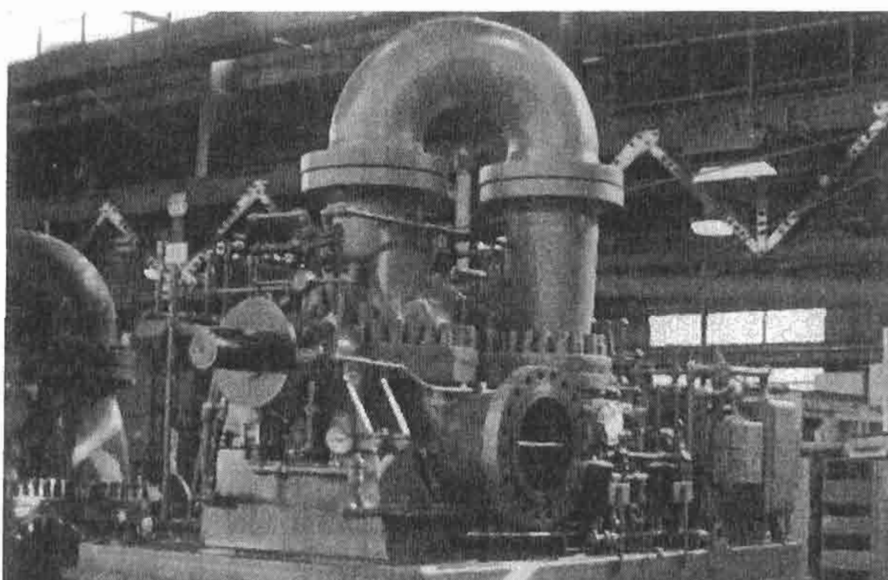


Figure 10-17. Two stage pipeline pump with bolted on crossover.

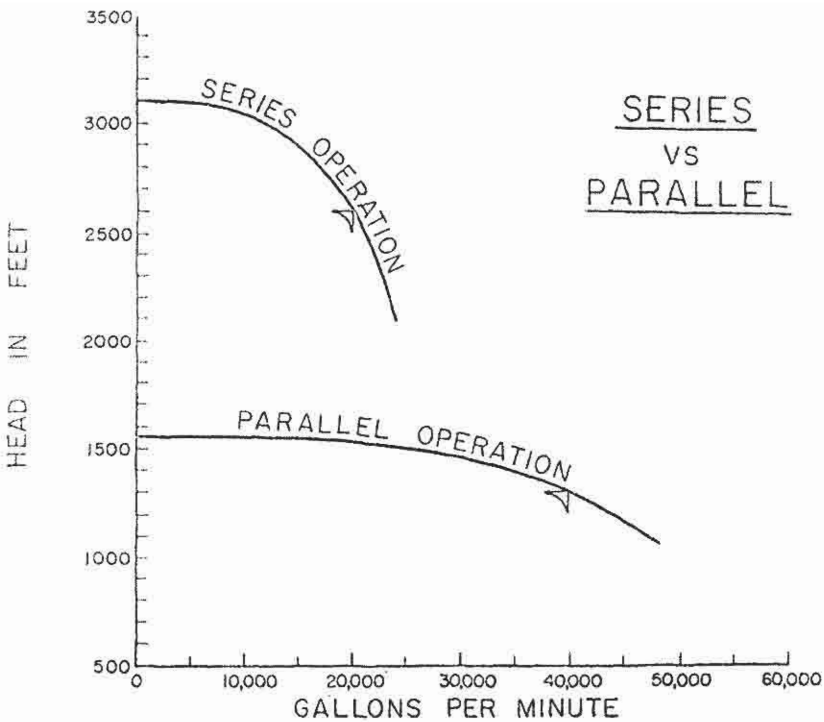


Figure 10-18. Performance of two-stage pipeline pump with bolted on crossover (series operation) and without crossover (parallel operation).

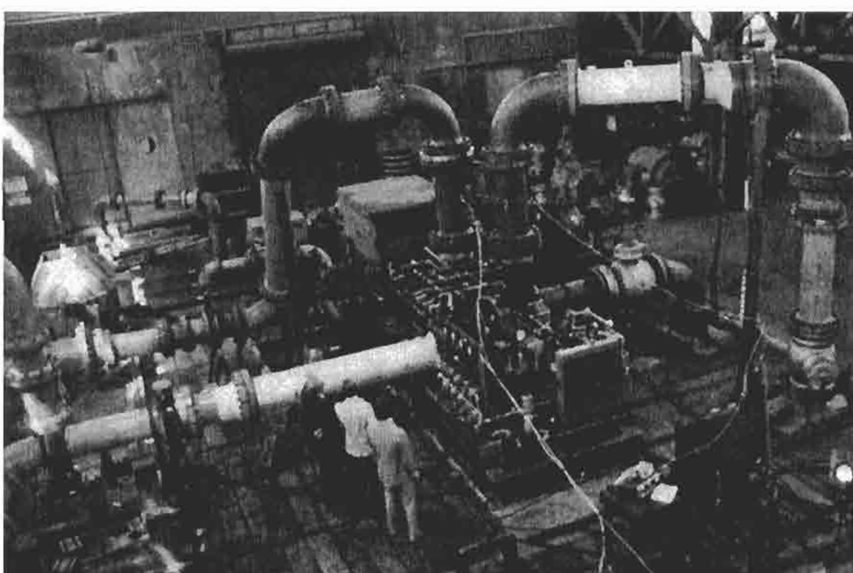


Figure 10-19. Two-stage pipeline pump being tested with crossover removed and both impellers operating in parallel.

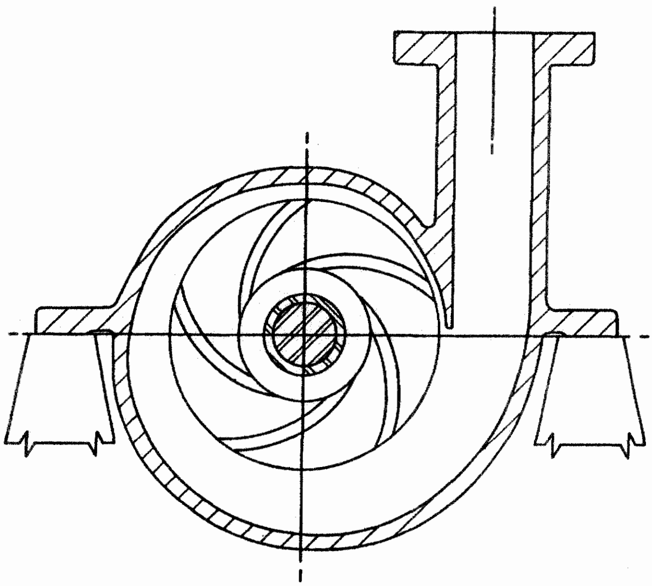


Figure 10-20. Blade passing frequency.

ing one or more of the sympathetic frequencies, or changing the pump generating frequency by a change of speed or blade number. With a bolted-on crossover a simpler solution is to add calculated pipe length between pump case and cross-over, which will change the wave length and relocate the acoustic frequency away from the generating frequency (Figure 10-21).

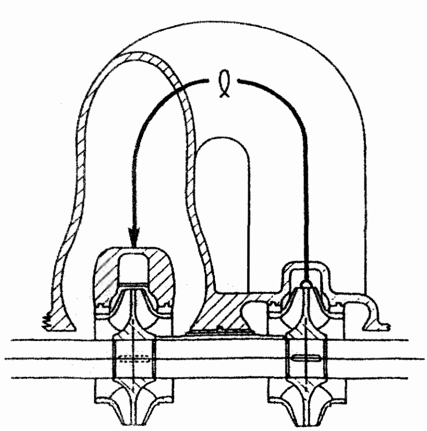


Figure 10-21. Acoustic frequency.

Slurry Pipelines

It would seem appropriate when discussing pipeline pumps to comment on future slurry pipeline operations. A review of existing slurry pipelines shows the most common pumps are positive displacement (PD). These are limited in capacity and normally operate up to 2,000 GPM with larger pumps currently being developed to achieve 4,000 GPM. To accommodate the high-capacity pipelines being planned with pipe size up to 42 in. and throughput rates of 500,000 BPD to 1,000,000 BPD, high-capacity centrifugal pumps will be a viable alternative. As installation costs and problems are directly related to the number of pumps required for the service and as centrifugal pump efficiency increases with specific speed, it is expected that centrifugals will become economically competitive at capacities starting at 20,000 GPM (Figure 10-22). As slurry pipeline capacity increases, the benefits of large-capacity centrifugal pumps operating in series as an alternative to multiple PD pumps operating in parallel become obvious. It is reasonable to assume that the principles and guidelines outlined in this chapter will be used on future slurry pipelines.

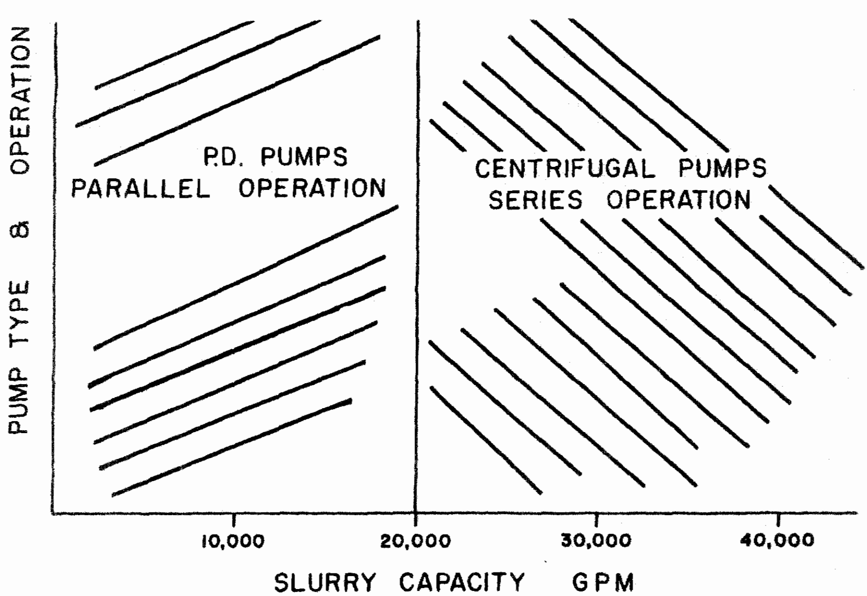


Figure 10-22. Suggested economic range for centrifugal pumps in slurry pipeline applications.

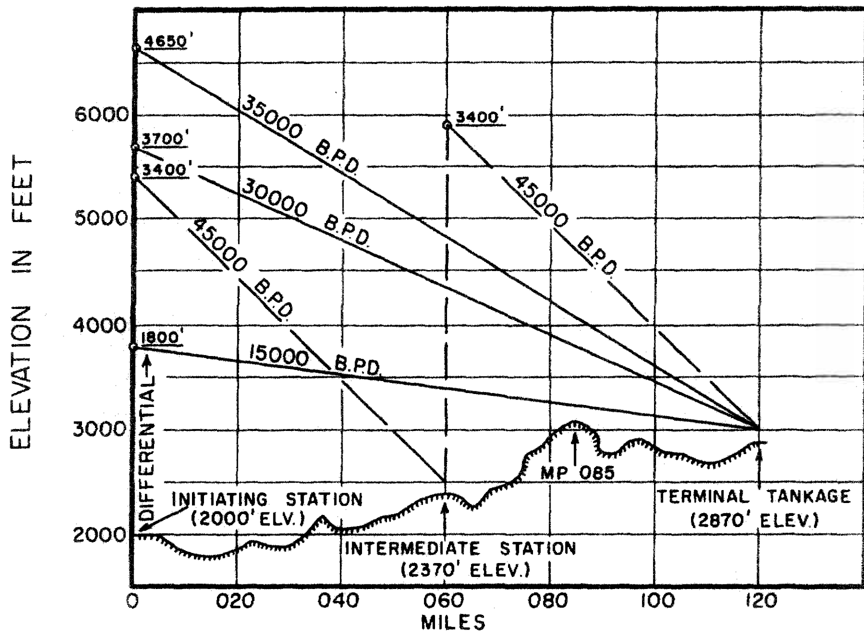


Figure 10-23. Hydraulic gradient and profile of 120-mile-long pipeline.

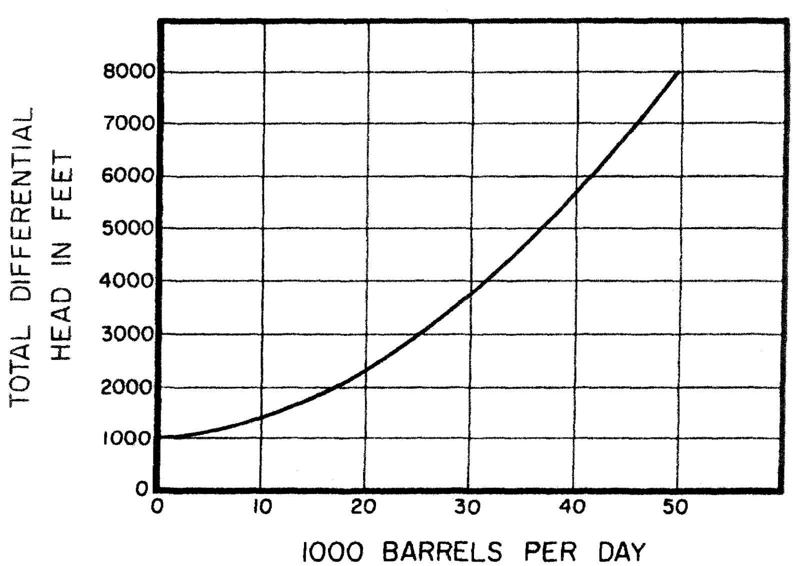


Figure 10-24. System head curve for 15,000 to 30,000 BPD production.

Example of Pipeline Pump Selection

As described earlier, pipeline pumps must be capable of adapting to change in pipeline throughput. The following exercise illustrates the pump selection process for a crude oil pipeline, where condition changes necessitate impeller changeouts, destaging, and volute chipping.

Pumps are required to transport crude oil from a developing oil field through a trunk line to a tank farm 120 miles away. Routing, pipe size, and pipe rating have been determined. A profile with hydraulic gradients and typical system curves has been developed, and the field is expected to produce 15,000 to 30,000 BPD (Figures 10-23 and 10-24). Note Milepost 085 (Figure 10-23) becomes a "control point." At least 1,200-ft station head is needed to overcome the elevation and to insure 50 psi positive pressure at this high point. The projected future rate is 35,000 BPD, resulting in the following pipeline design conditions:

	Initial	Future
Capacity (BPD)	15,000–30,000	35,000
Capacity (GPM)	437–875	1,021
Differential head (ft)	1,800–3,700	4,650
Differential head (psi)	694–1,246	1,792

A booster pump has been sized to provide adequate NPSH to the mainline pumps. In this example, the head developed by the booster pump will be disregarded.

First consideration is the "future" condition. With 0.89 specific gravity and an estimated 76% pump efficiency, the total brake horsepower required is approximately 1,400. The most economical energy source is a local electric company. Starting current restrictions at the station site limit motor horsepower size. To satisfy this restriction and to gain flexibility of operation as well as partial capacity with one pump out of service, two pumps driven by 700 HP motors operating in series are preferred. The ratings for each pump to meet future conditions, will then be 1,021 GPM at 2,325 ft.

A good selection would be a 4-in. pump with 5 stages and 10³/₈-in. diameter impellers (pattern 2008-H), for 465 ft per stage (Figure 10-25). With 80% efficiency, the brake horsepower is 667. Performance curves for single-pump operation and two pumps in series are plotted against the system head curves (Figure 10-26). Operating points will be at intersections of pump curves and system curves. Lower capacities will require throttling at the station discharge control valve, which, in effect, produces a steeper system head curve. In this example, when flow is reduced to 885 GPM, the differential head developed by two pumps is

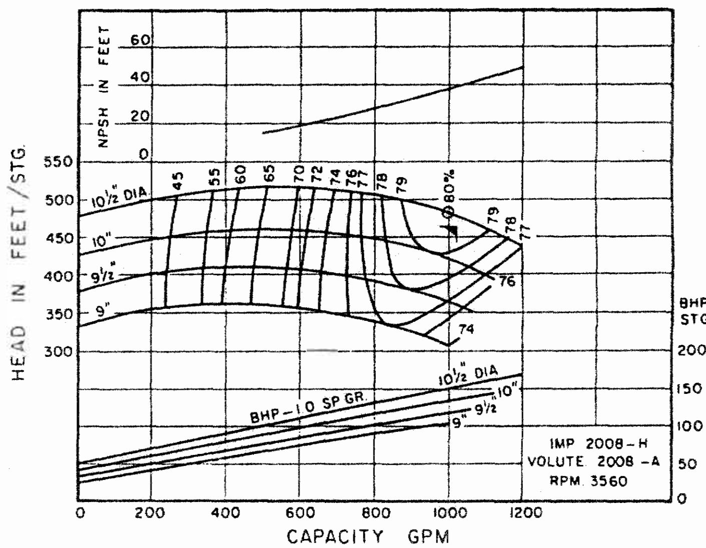


Figure 10-25. Performance for one stage of multi-stage pipeline pump selected for "future" condition.

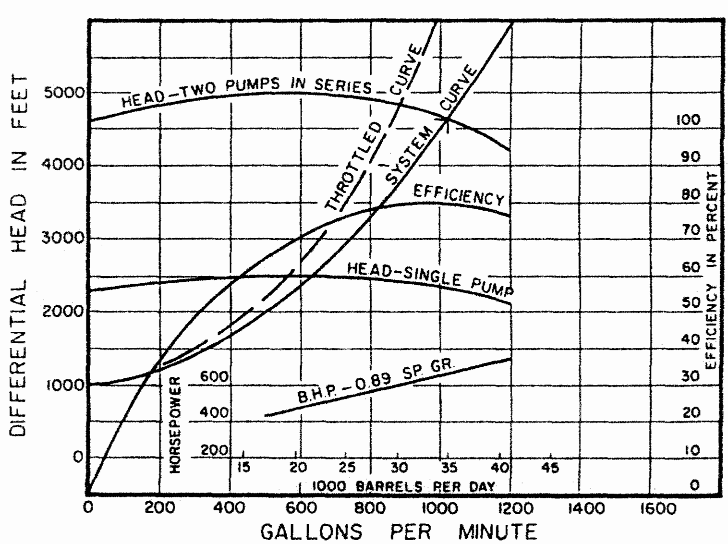


Figure 10-26. Performance of pump from Figure 10-25 for single pump operation and two pumps in series plotted against system head curves.

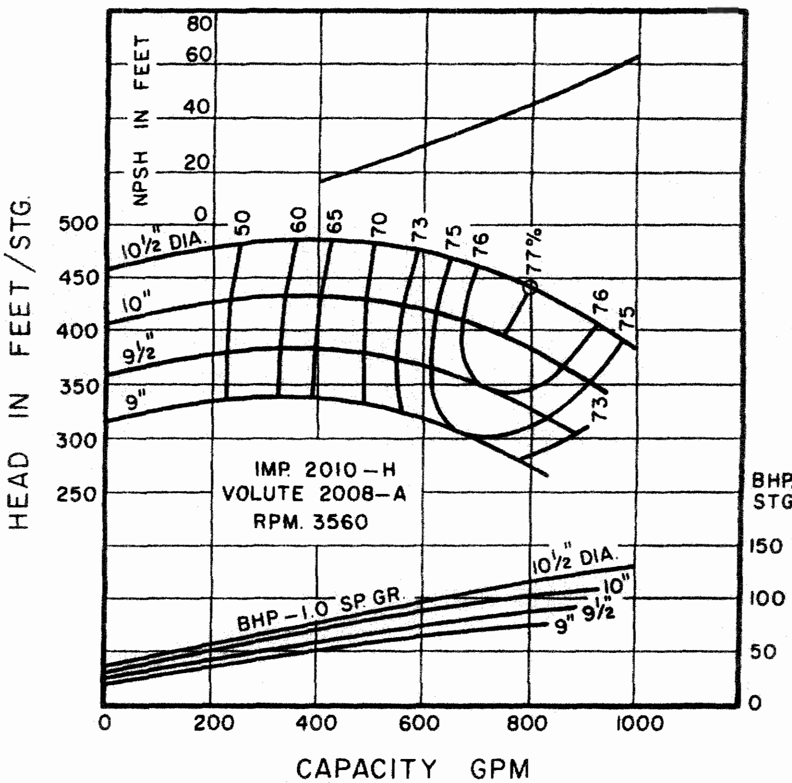


Figure 10-27. Performance from Figure 10-25 modified by installation of low-capacity impellers for "initial" condition.

4,850 ft. The system requires only 3,750 ft, therefore 1,000 ft is lost to friction (head) across the control valve. Note with one pump operating and no throttling, the capacity will be 630 GPM and pump efficiency will be 72%. For reduced rates, throttling and wasting of energy will be avoided by running one pump as much time as possible and making up by running two only as necessary.

Having determined size and configuration for the mainline units, let's consider how operation and efficiency can be improved in the initial 437 to 875 GPM capacity range. Figure 10-27 shows performance of the pump selected with impellers changed to pattern #2010-H (low capacity). This impeller, which peaks at 800 GPM, is more efficient at capacities below 760 GPM and would be a good choice for initial operation. One of the five-stage pumps, say the #1 unit, can be furnished destaged to four stages. By operating with four, five, or nine stages, various rates can be

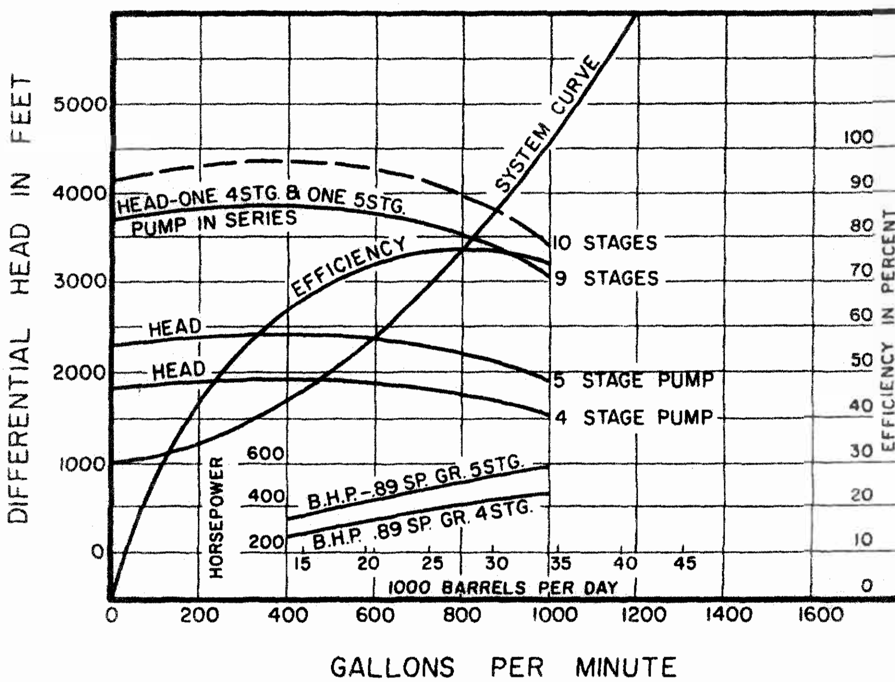


Figure 10-28. Performance of pump from Figure 10-27 in four, five, nine, and ten stages plotted against system head curve.

attained without throttling (Figure 10-28). When rates exceed 830 GPM, the #1 unit can be upstaged. At this point, throughput is approaching the 35,000 BPD future design rate and it is time to consider changing impellers to pattern #2008-H (high capacity).

Let's now assume after many years' operation there is a need to further increase capacity to 45,000 BPD (1,312 GPM). With 6,800 ft (2,620 psi) differential head, two stations are required to stay within the pipe pressure rating. The intermediate station is located near midpoint for hydraulic balance. Differential head required at each station is then half the total or 3,400 ft.

A new station system curve is developed (Figure 10-29). At 1,312 GPM and 3,400 ft with 76% efficiency, the station BHP would be 1,319. With two pumps in series, head required of each pump would be 1,700 feet, and 660 BHP per unit would be within horsepower rating of existing drivers. In this instance, volutes can be chipped to provide a throat area equivalent to volute pattern #2204-A. Without changing impellers, the higher capacity performance can be attained (Figure 10-30).

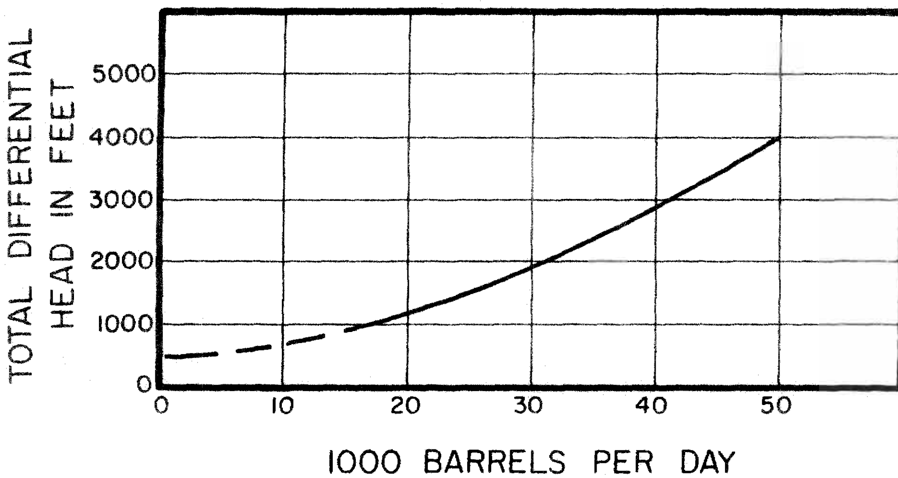


Figure 10-29. System head curve for 45,000 BPD production.

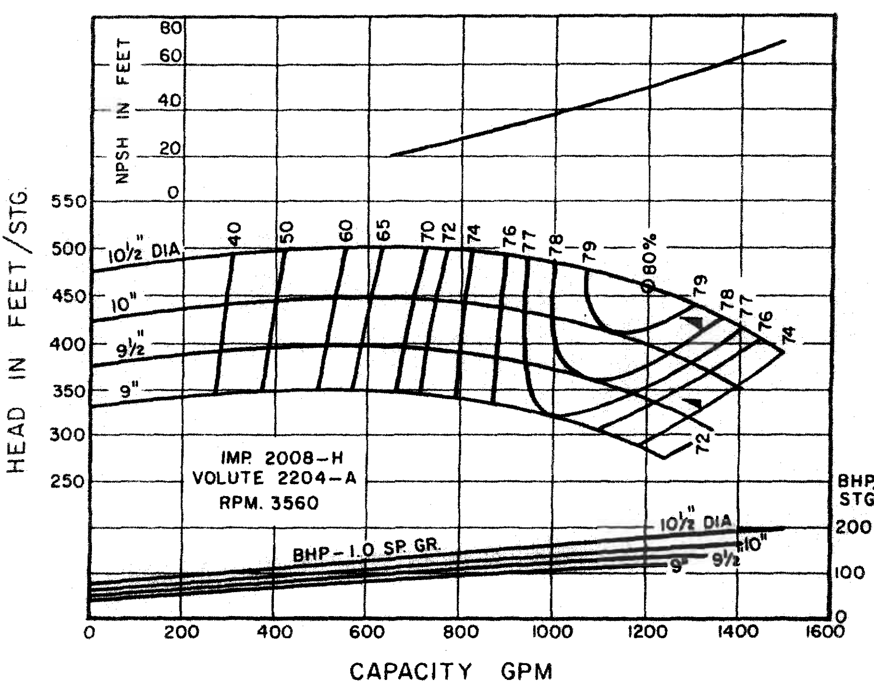


Figure 10-30. Performance from Figure 10-25 with volutes chipped to increase pipeline throughput to 45,000 BPD.

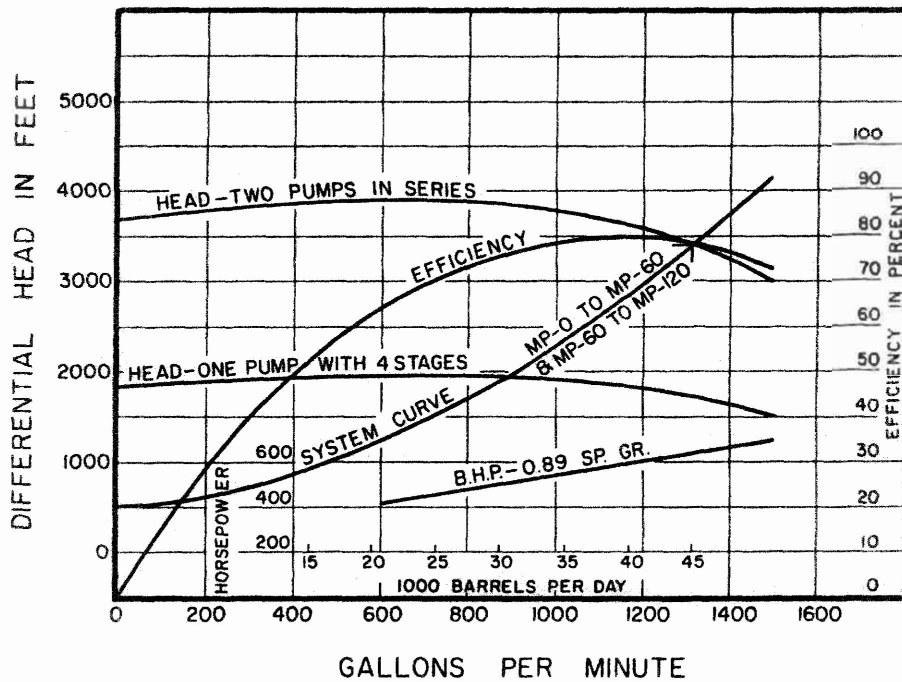


Figure 10-31. Performance of pump from Figure 10-30 in four-stage single-pump operation and two pumps in series plotted against system head curve.

To get 1,700 ft, we have two choices: The impellers of the 5-stage pumps can be trimmed to about a 9¾-in. diameter for 340 ft/stage. Alternatively, the pumps can be destaged to 4 stages and at 425 ft/stage will not require impeller trimming. Refer to iso-curve (Figure 10-30) and note how trimming affects efficiency at capacities approaching and beyond peak. At 1,319 GPM, destaging is obviously preferred. Figure 10-31 shows performance with one or two pumps operating.

Series vs. Parallel

A flat head-capacity curve is desirable for pipeline pumps installed in series. When station capacity is being controlled by throttling, less horsepower is lost than would be with a steep head-capacity curve. Capacity control of individual pumps does not present a problem since pumps operating in series will have the same flow rate.

Where pipeline pumps are to be installed in parallel, identical pumps with constantly rising head-capacity curves are usually called for. Load is

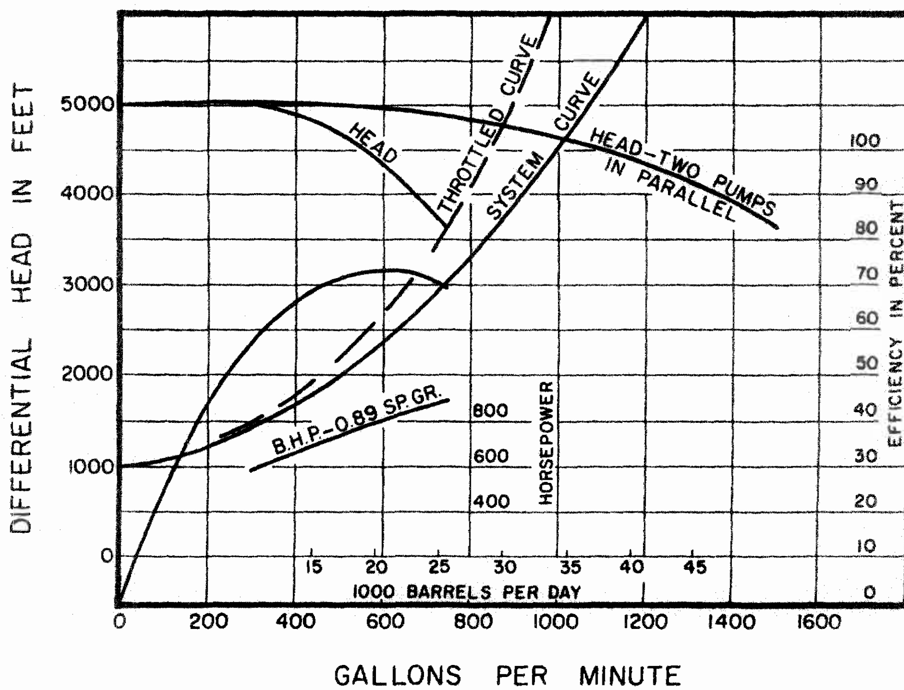


Figure 10-32. Performance of two half-capacity full head pumps installed in parallel plotted against system head curve.

shared equally, and there is less chance of a pump operating at less than minimum continuous stable flow. Figure 10-32 shows our system curve and two half-capacity full head pumps installed in parallel. In this particular system, parallel configuration is a poor choice. When only one pump is running, it is necessary to throttle to stay within operating range.

Parallel configuration should be considered in systems where a substantial portion of the head is static. When pump configuration is not clear cut, it is wise to plot each station curve along with pump curves and carefully analyze parallel versus series operation.

Waterflood Pumps

Many types of pumps are used to extend the life of declining oil fields by injecting displacing fluids at pressures ranging from 2,000 psi to 8,000 psi. In waterflooding applications (secondary recovery), centrifugals are preferred when injection capacities exceed 10,000 BPD. Typical

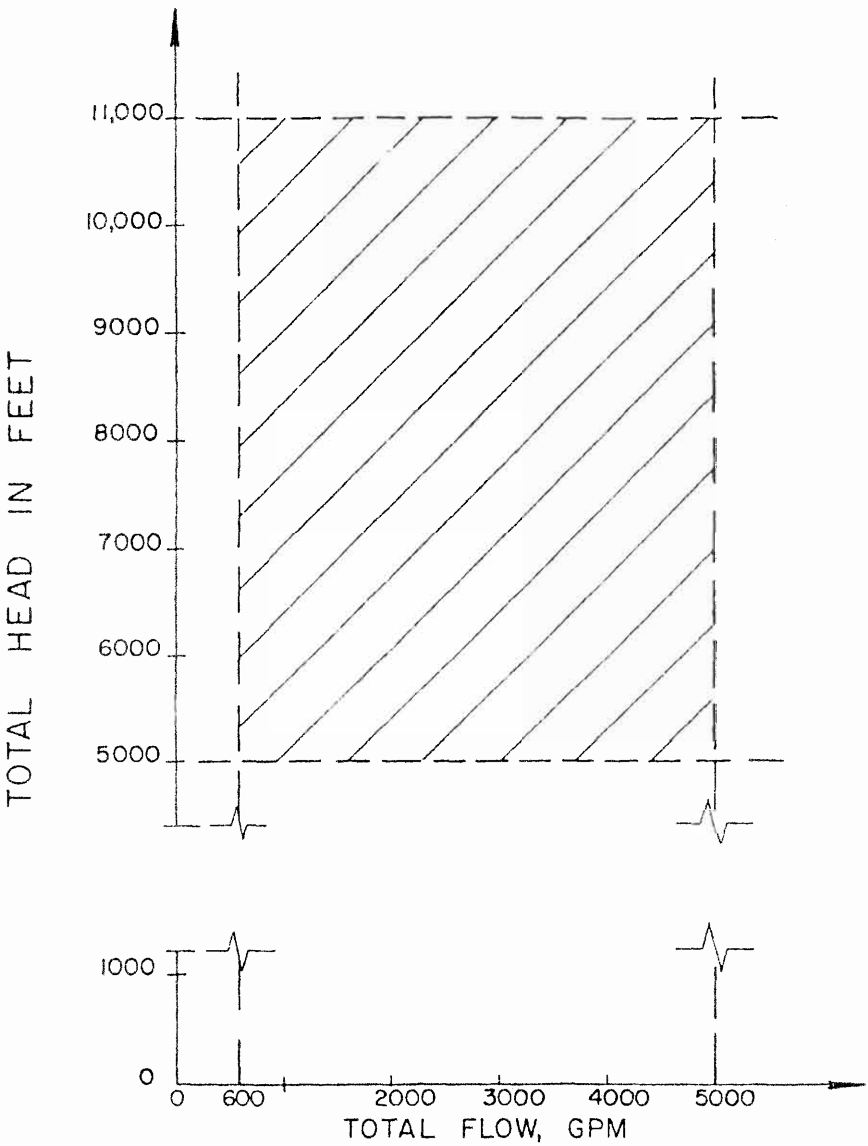


Figure 10-33. Performance coverage for typical waterflood applications.

pump conditions range in capacity up to 5,000 GPM and heads up to 11,000 feet (Figure 10-33). Pumps are normally multi-stage with horizontal-split or double-case construction. As many applications use corrosive formation water as the pump fluid, special precautions must be taken in material selection. Among the major factors that must be considered in liquid analysis are:

- Salinity.
- Aeration.
- Particulate matter.
- Hydrogen sulfide (sour gas) content.
- Inhibiting additives.
- pH.
- Continuous or intermittent service.
- Composition of attached piping.

Some materials used successfully in waterflood applications are shown in Table 10-1. The groupings are listed in order of increasing corrosion resistance.

CO_2 Pumps

When waterflood production is no longer effective, injection of CO_2 and other gases (enhanced recovery) can recover up to 70% of the remaining oil. CO_2 can be transported as a gas, using compressors or as a liquid using centrifugal pumps.

Due to the questionable lubricating properties of liquid CO_2, special precautions must be taken where the possibility of internal metal-to-

Table 10-1
Materials for Waterflood Service

Group	Pressure Casing	Impeller	Shaft	Wear Parts
1	Steel	CA-6NM*	410 SS*	410 hardened
2	CA-6NM*	CA-6NM*	410 SS*	410 hardened
3	316 SS	316 SS or 17-4 PH	Monel K-500 or 17-4PH	316L With stellite overlay
4	Duplex SS	Duplex SS	Monel K-500	Duplex alloy with stellite overlay

** When H_2S is present or suspected, hardness shall not exceed R_c22 (240HB).*

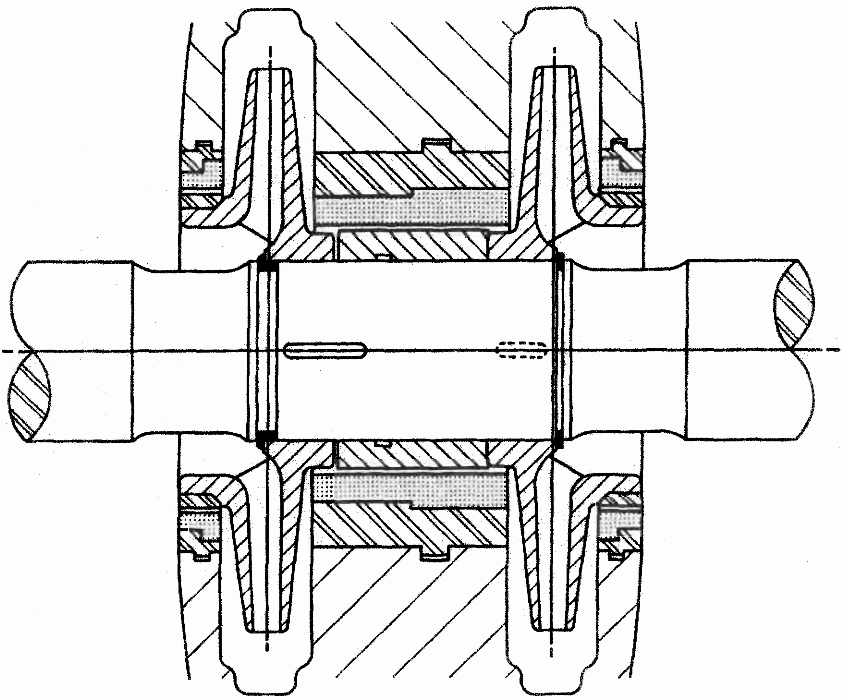


Figure 10-34. Bronze impregnated graphite inserts for dry running applications.

metal contact exists. One approach used successfully is to install bronze-impregnated graphite inserts into all stationary wear rings (Figure 10-34). Rings of this type have operated with no damage during dry running tests on multi-stage pumps where dynamic deflection permitted internal contact with rotating and stationary rings.

Mechanical Seals

A number of solutions in the sealing of CO_2 are offered by the various seal manufacturers who in their selection process must consider the poor lubrication quality, the possibility of icing at the faces, and the typical high-suction pressure. The CO_2 pipeline pump shown in Figure 10-35 had a pumping rate of 16,000 GPM and a maximum suction pressure of 1,800 psia. In this application, double seals were chosen using a compatible buffer fluid with adequate lubrication properties.

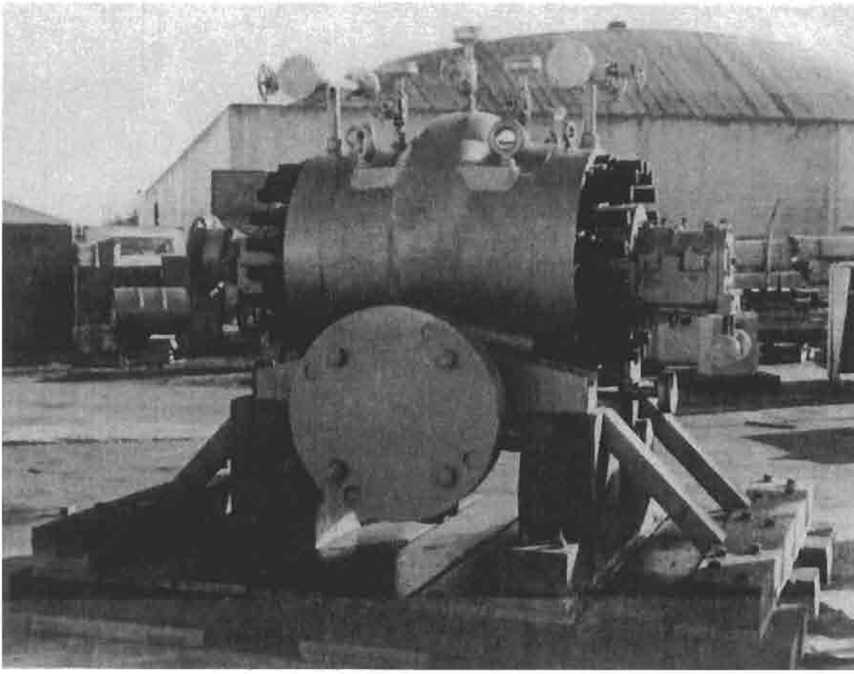


Figure 10-35. CO_2 pipeline pump (courtesy BW/IP Internal, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

Horsepower Considerations

As liquid CO_2 is compressible, special consideration during the pump selection process must be given to horsepower requirements. Gas horsepower (GHP) and brake horsepower (BHP) should be calculated, and it is recommended that pump shaft and driver be sized to accommodate the larger of the two. Depending on the thermodynamic properties, GHP can be greater or less than BHP. To calculate GHP, it is necessary to predict the behavior of the liquid across the pump. This can be done either from interpolation of thermodynamic tables (assuming pure CO_2) or by computer calculations using the new equation of state for the actual composition (see Starling 1973). This method has been widely applied to predict the behavior of any mixture of hydrocarbons. While it is recommended that final pump design be based on computer calculations, preliminary pump selection can be based on thermodynamic tables. For a constant throughput in units of standard cubic feet per day (SCFD), the inlet and outlet pump capacity in GPM will change with change in specific volume (Figure 10-36).

Calculation Procedure

The following example assumes the use of appropriate Thermodynamic Tables.

Given

$P_1 = 1400$ psia
$P_2 = 1650$ psia
$T_1 = 80°F$
$F_R = 450 \times 10^6$ SCFD

Step 1. Find Inlet Conditions

A. From P_1, and T_1, find H_1, V_1, and S_1 from tables

$H_1 = -3858.81$ Btu/lb
$V_1 = 0.01925$ ft³/lb
$S_1 = 0.7877$ Btu/lb °R

B. Calculate γ_1, SG_1, Q_1, and H

$$\gamma_1 = 1/V_1 = 1/0.01925 = 51.948 \text{ lb/ft}^3$$
$$SG_1 = \gamma_1/62.33 = 51.948/62.33 = .8334$$
$$Q_1 = F_R \times .116 \times 7.48/1440 \times \gamma_1$$
$$Q_1 = 450 \times 10^6 \times .116 \times 7.48/1440 \times 51.948 = 5220 \text{ GPM}$$
$$H = (P_2 - P_1)\ 2.31/SG_1$$
$$H = (1650 - 1400) \times 2.31/.8334 = 693 \text{ ft}$$

C. Select Pump
Select pump for 5220 GPM and total pump head of 693 ft
Note, pump efficiency in this example is 85%.

D. Calculate BHP

$$BHP = Q_1 \times H \times SG_1/3960 \times \text{Pump Efficiency}$$
$$BHP = 5220 \times 693 \times .8334/3960 \times .85 = 896$$

Step 2. Find Outlet Conditions

A. Assume constant entropy $S_1 = S_2$

B. From $P_2 = 1650$ psia and $S_2 = 0.7877$ Btu/lb °R interpolate tables to find H_2, V_2 and T_2

$$H_2 = -3857.93 \text{ Btu/lb}$$
$$V_2 = 0.0191665 \text{ ft}^3\text{/lb}$$
$$T_2 = 82.88 \text{ °F}$$

C. Calculate ΔH, and ΔH^1

$$\Delta H = H_2 - H_1$$
$$\Delta H = (-3857.93) - (-3858.81)$$
$$= 0.88 \text{ Btu/lb}$$
$$\Delta H_1 = \Delta H/\text{Pump Efficiency}$$
$$= .88/.85 = 1.0353 \text{ Btu/lb}$$

Step 3. Correct Outlet Conditions For Pump Efficiency

A. With P_2 remaining same calculate H_C

$$H_C = H_1 + \Delta H^1$$
$$= -3858.81 + 1.0353$$
$$= -3857.77 \text{ Btu/lb}$$

B. From H_C and P_2 interpolate tables to find V_C, S_C, and T_C

$$V_C = 0.019192 \text{ ft}^3\text{/lb}$$
$$S_C = 0.78798 \text{ Btu/lb °R}$$
$$T_C = 83.078 \text{ °F}$$

C. Calculate γ_C and Q_2

$$\gamma_C = 1/V_C = 1/0.019192 = 52.105 \text{ lb/ft}^3$$
$$Q_2 = F_R \times .116 \times 7.48/1440 \times \gamma_C$$
$$= 450 \times 10^6 \times .116 \times 7.48/1440 \times 52.105$$
$$= 5204 \text{ GPM}$$

Step 4. Calculate GHP

$$GHP = \Delta H^1 \times M/2545$$

$$\begin{aligned} \text{where } M &= Q_1 \times 500 \times SG_1 \\ &= 5220 \times 500 \times .8334 \\ &= 2{,}175{,}174 \text{ lbs/hr} \end{aligned}$$

$$\begin{aligned} GHP &= 1.0353 \times 2{,}175{,}174/2545 \\ &= 885 \end{aligned}$$

Step 5. Size Driver and Pump Shaft for 896 BHP

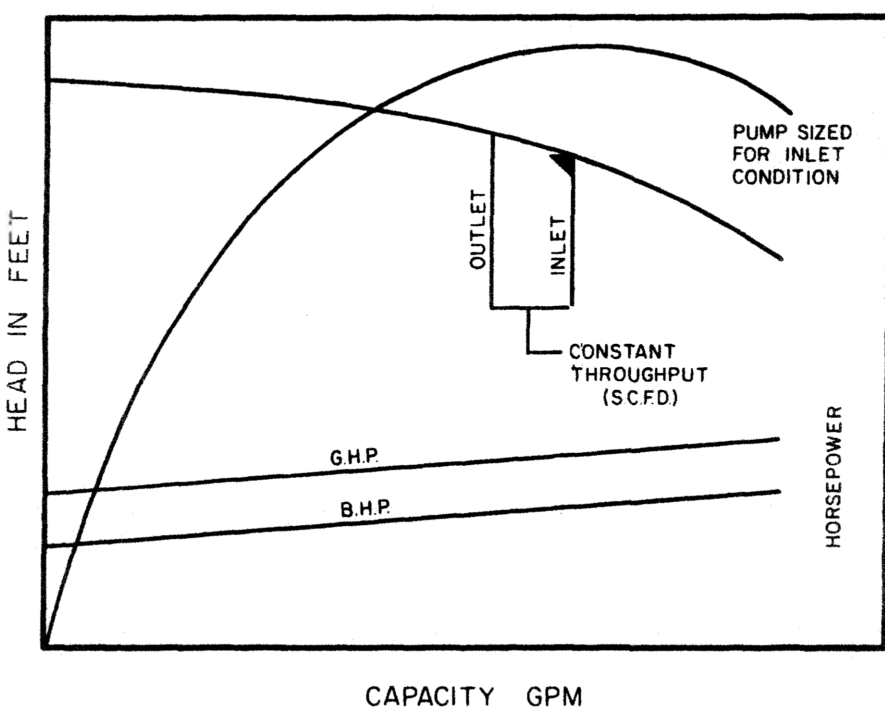


Figure 10-36. Performance curve for CO_2 pump.

Notations

BHP	Brake horsepower
F_R	Flow rate (SCFD)
f	Blade passing frequency = N × vane no./60
f_1	Acoustic frequency = velocity of pulsation (ft/sec)/wave length l (ft)
GHP	Gas horsepower
H	Total pump head (ft)
H_1	Inlet enthalpy (Btu/lb)
H_2	Outlet enthalpy (Btu/lb)
H_C	Outlet enthalpy corrected (Btu/lb)
ΔH	Differential enthalpy (Btu/lb)
ΔH^1	Differential enthalpy corrected (Btu/lb)
M	Mass flow rate (lbs/hr)
N	Speed (RPM)
P_1	Inlet pressure (psi)
P_2	Outlet pressure (psi)
Q_1	Pump capacity at inlet (GPM)
Q_2	Pump capacity at outlet (GPM)
S_1	Specific entropy at inlet (Btu/lb °R)
S_2	Specific entropy at outlet (Btu/lb °R)
S_C	Specific entropy at outlet corrected (Btu/lb °R)
SG_1	Specific gravity inlet
T_1	Temperature at inlet (°F)
T_2	Temperature at outlet (°F)
T_C	Temperature at outlet corrected (°F)
V_1	Specific volume at inlet (ft^3/lb)
V_2	Specific volume at outlet (ft^3/lb)
V_C	Specific volume at outlet corrected (ft^3/lb)
γ_1	Density inlet (lb/ft^3)
γ_C	Density corrected (lb/ft^3)

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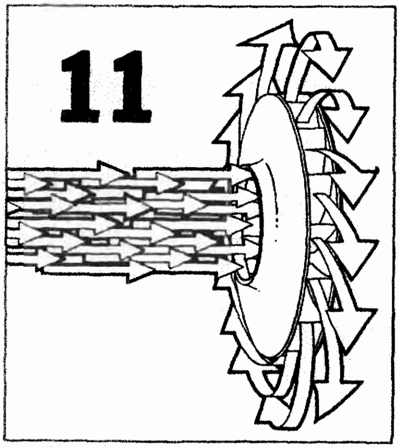


High Speed Pumps

By **Edward Gravelle**
Sundstrand Fluid Handling Division

The trend toward higher process pressures, which has developed over the past half century or so, has provided impetus to exploit the advantages of high speed to better provide high head capability in centrifugal pumps. High head centrifugal design may be provided by using high rotating speed, by series multi-staging, or by a combination of both.

The advantages of high-speed design are several. Fewer and smaller stages are required to meet a given head objective, and not infrequently, single-stage designs can provide capability that would otherwise require multi-staging. Smaller, more compact design tends toward shorter shaft spans that can result in lowered shaft deflection and improved shaft dynamics. Compactness, involving fewer and smaller components, is economical of materials, which becomes increasingly important when expensive materials are required for handling severe process fluids. Minimal spares inventory and relatively quick and easy maintenance are attributes of high speed, which are often very attractive to users, to whom pump availability is central to the viability of their businesses. Lightened pump weight can translate into smaller and less expensive mounting foundations.

Conversely, other considerations are involved in a movement toward higher speeds. About 95% of all pumps in industry are driven by electric motors in a world built around 50 and 60 Hz electric systems, or 3,000 and 3,600 RPM speed limits with two pole motors. This introduces the need for speed-increasing gear systems, which must be justified in exchange for high-speed pump advantages. NPSHR increases with increasing speed, placing limits on speed for a given NPSHA. Need arises to improve suction performance as much as possible to extend speed limits. Material capabilities must be recognized to keep stress levels within prudent design limits. Modern seal technology is required to meet the demands of combined high speeds and high pressures. Bearing design sophistication is frequently required to ensure reliable operation, and recognition of the influence of bearings on shaft dynamics is often necessary. Increased noise generation can occur with high-speed equipment due to high power densities and lightweight construction.

Industrial acceptance of high-speed pump technology is illustrated by Karassik, who has indicated that the introduction of high-speed boiler feed pumps in 1954 was followed by the steadily increasing use of these machines, culminating in total abandonment of the older 3,550 RPM equipment by 1971.

This chapter will deal primarily with an unconventional pump type particularly suited for operation at high to very high speeds to produce typically very high heads at low to moderate flow rates. Although this pump type has enjoyed wide acceptance in industry, comparatively little on this design has appeared in the literature.

An early commercial application of this design began in 1959 for an aircraft service. The pump was used in the Boeing 707 for takeoff thrust augmentation in jet engines, at a time when engine power was relatively low and the world had not caught up with the generally longer runstrip requirements for jet aviation. This pump rotated at 11,000 RPM and delivered 80 GPM of water to the combustors at 400 psi (well above combustor pressure to allow atomization), to increase the engine mass flow rate and increase thrust by 15% for takeoff. The unit weighed only 8½ pounds, including step-up gearing from the 6,500 RPM power takeoff pad to pump speed. Some 250 units were produced for this service.

In 1962, an industrial version of the new pump type rated to 100 HP and 6,000 feet of head in a single stage was introduced to the petrochemical industry. In the past two decades, this concept has grown into a family of products ranging from 1 to 2,500 HP, direct drive to 25,000 RPM and heads to 12,000 feet. Most commonly these products consist of a single high-speed stage, but as required, employ two or three stages to satisfy need for extreme heads or the combination of high head and low NPSHA. Over the past two decades many thousand machines of this type have been placed in service around the world.

History and Description of an Unconventional Pump Type

Developmental work on the pump type central to the discussions in this chapter was initiated in Germany prior to World War II to meet urgent wartime requirements and after 1947 continued in Britain. Need for a simple, lightweight, and easily manufactured pump suited to produce high heads at low flow rates existed in connection with aircraft and rocket propulsion systems. An unorthodox high-speed centrifugal pump concept resulted and was described by Barske in papers published in 1955 and 1960.

This pump is described as an open impeller type and is exemplified as highly unorthodox by Barske himself who states: "To a skilled designer the pump which forms the subject of this paper will, at first glance, appear most unfavorable and may well be regarded as an offense against present views of hydrodynamics." Reasons exist, however, to break with conventional design practice to meet objectives which would otherwise be difficult to achieve. Intentionally flaunting the rules, in fact, provides a pump design that can equal or exceed the performance of conventional pumps in the head-flow design range for which it is intended and for which it is best suited.

Typical Barske-type pump construction is illustrated by the sketch in Figure 11-1. The salient features of the design start with a simple open impeller, which rotates within a case bored concentrically with the impeller centerline. A single emission throat with a conical diffuser section is oriented tangentially to the case bore. Conical diffusers provide high recovery efficiency because of their minimal wetted area. A cone angle of 10° is commonly used, providing good recovery potential and reasonable cone length requirements.

Radical departure from conventional design practice exists in the exceptionally tall blade geometry used, with the impeller tip height, b_2, set equal to or moderately greater than the emission throat diameter, d_1. Blade angle, θ, is unimportant except that the flow area in the impeller eye must at least equal the area of the suction passage. Further obvious deviation from normal practice is the use of plain radial blades, with no attempt made to match inlet flow streamlines.

Performance trends of the Barske pump are generally as indicated in Figure 11-2A. The head at zero flow, or shutoff, is about equal to the design head, with a head peak a few percent higher than design in the neighborhood of half design flow. This curve shape is referred to as an unstable curve and is often viewed as undesirable, as described in Chapter 1. A stable curve is one in which the head rises continuously as flow is reduced from design to shutoff. Head drops rapidly for flows above design, and zero head or cutoff normally occurs around 130% of design

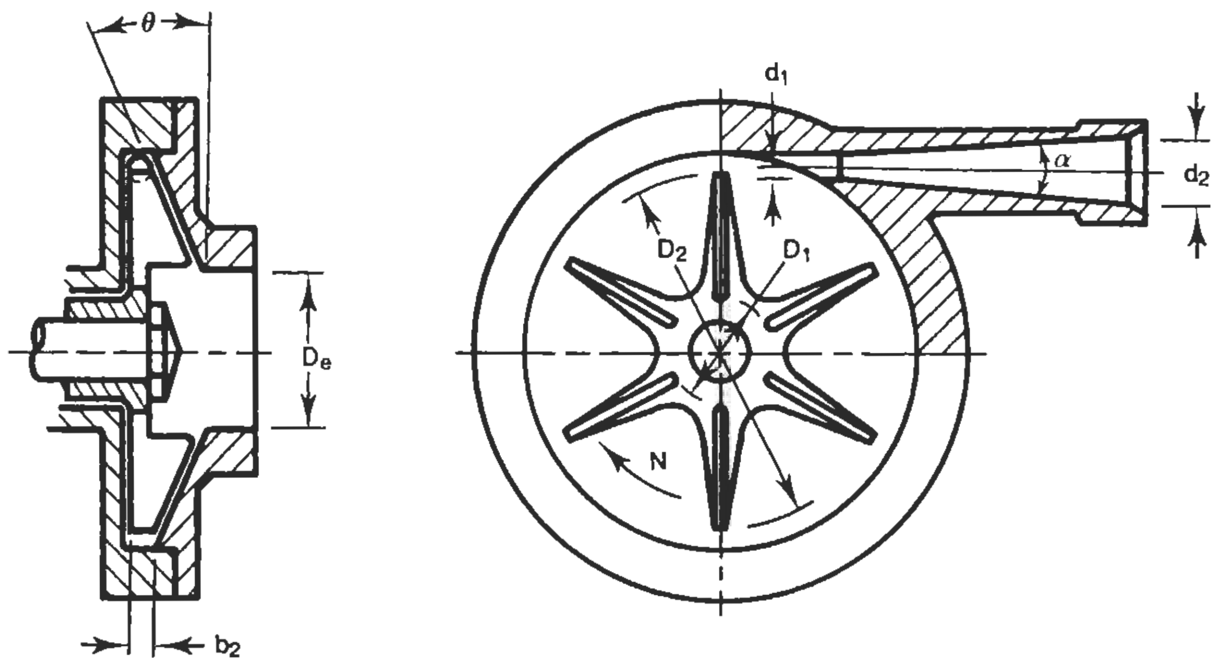


Figure 11-1. Barske open impeller centrifugal pump.

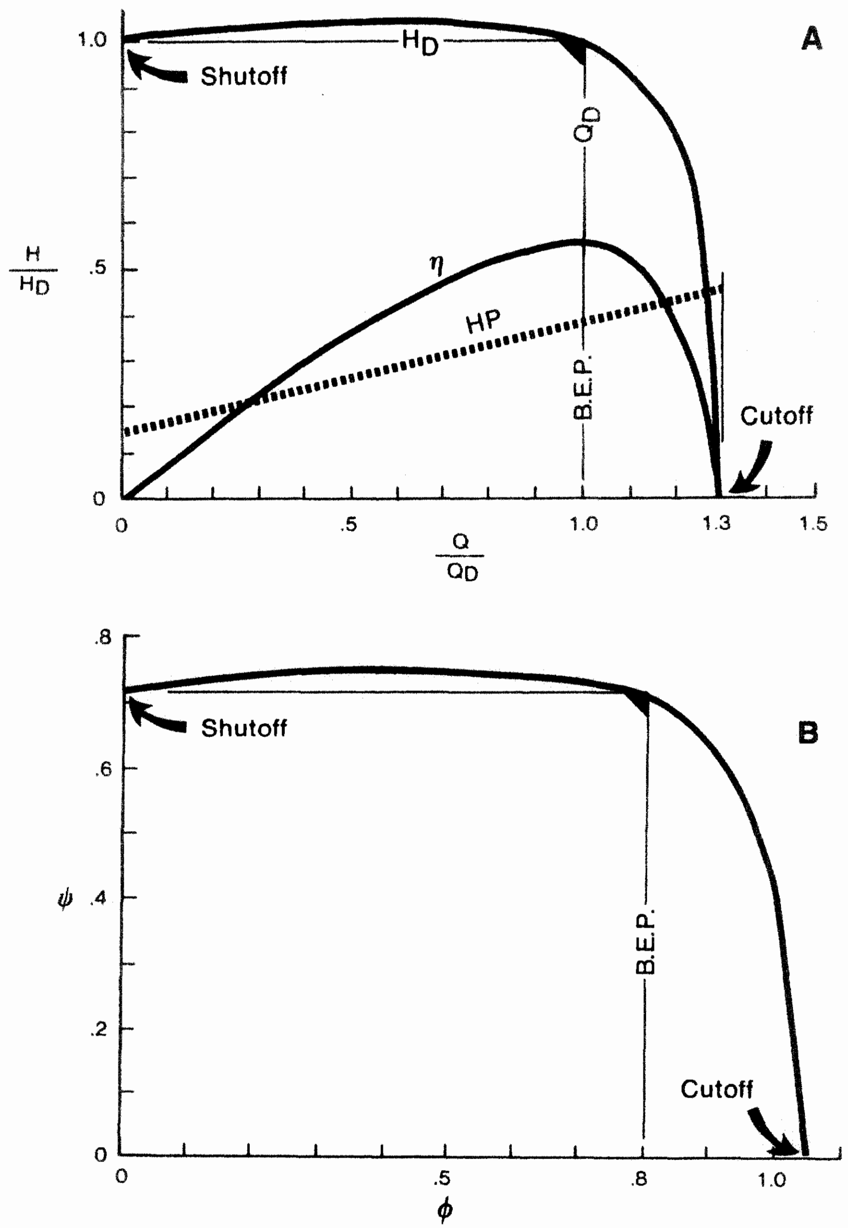


Figure 11-2. (A) Typical performance trends for Barske pump; (B) typical head coefficient vs. flow coefficient for Barske pump.

flow. Very high suction pressures move the cutoff point to higher flow rates.

Figure 11-2B shows the typical head coefficient and flow coefficient characteristics of Barske, which is simply a reflection of the head-flow curve. The tall radial blade impeller geometry produces relatively high head coefficients typically in the range of ψ = .7 to .75.

The Barske pump does not require close operating clearances to provide good performance. Open impellers present a leakage path from the impeller tip back to the eye through the impeller front side clearance, but only low sensitivity to this clearance has been found. Clearances normally used in commercial pump sizes range typically from .03 to .05 in., which simplifies manufacture and maintenance. The open impeller concept frees the pump from performance decay which can occur with wear ring construction when erosion or rubbing contact increases the ring clearance.

Hardware is physically small and geometrically simple allowing production by straightforward machining operations. Surface finishes typical of ordinary shop practice are adequate to avoid excessive losses, which would be likely to exist with relatively rough cast surfaces. Very little or no benefit is available through polishing the case surfaces. Impellers are usually made from castings which are trimmed to match the case geometry. Surface finish on the impellers is unimportant due to the use of tall impeller blades, which results in low radial flow velocities.

Terminology

The Barske pump design deviates from that of higher specific speed designs, which are generally referred to as full emission (F.E.) radial or Francis types. Francis-type pumps are generally suited for relatively high flow rates and moderate head rise, and meet these objectives with the highest attainable efficiency of any centrifugal pump type. Full emission designs almost universally use backswept impellers configured according to refined hydraulic practices so as to provide constant meridianal velocity, to avoid design-point flow separation, to avoid incidence losses and so forth. These designs are characterized by flow which exits uniformly through the full impeller periphery, hence the description: full emission. But these design procedures become less beneficial with high stage head and low flow design objectives, i.e., in low specific speed designs. This occurs because flow passages are being decreased in size simultaneously with increasing impeller diameter, with an attendant disproportionate increase in friction losses and lowered efficiency.

It has been established through experience that high-flow machines can be made to work relatively well at low flow rates by simply plugging some portion of the exit flow path; for example, plugging some of the diffuser passages in a vaned diffuser. This, of course, results in impeller passages that are oversized for the lower flow rates according to conventional design practice, but in fact can produce efficiencies superior to those attainable with the very narrow passages that would result in F.E. design procedures. The term partial emission (P.E.) arose to describe such pump geometry, apparently coined by Balje.

The Barske pump is correctly classified as a partial emission type, since the emission throat area is much smaller than the impeller emission area. More to the point, net through-flow in the Barske pump can occur only in a path extending generally from the inlet eye to the vicinity of the emission throat. This is true for the simple reason that the remainder of the case cavity is concentric with the impeller and is filled with incompressible fluid, precluding any possibility of a radial flow component. High circumferential fluid velocities exist in the forced vortex created by the impeller, which are superimposed on the through-flow stream extending from eye to throat. Through-flow is then in essence a fluid migration, where a given element of fluid makes a number of circuits within the forced vortex and moves to successively higher orbits in the eye-to-throat flow region.

Alternatively, the Barske pump can be referred to generically and geometrically as a concentric bowl P.E. pump or simply a concentric bowl pump. This is convenient for easier differentiation of the original pump type from its evolutionary offshoots to be described later.

Partial Emission Formulae

Use of tall, radial-bladed impellers in P.E. pumps results in flow conditions that must be described as disorderly. No attempt is made to match inlet geometry to the flow streamlines. Very low mean radial flow velocities combined with high tip speeds reduce the discharge vector diagram to essentially the tangential tip speed vector, u_2. Calculation procedures for P.E. pumps then are based on simple algebraic expressions involving impeller tip speed rather than on the vector diagrams used in F.E. design.

Barske starts with the assumption that the fluid within the case rotates as a solid body or forced vortex, and neglects the negligibly low radial component, resulting in a theoretical head of:

$$H' = \frac{u_2^2 - u_1^2}{2g} + \frac{u_2^2}{2g} \tag{11-1}$$

The first term represents the vortex or static head and the second term represents the velocity head or dynamic head. Even within the Barske paper, question arose as to whether inclusion of the $U_1^2/2g$ term in the static head expression recognizing the blade inlet diameter is appropriate. Low through flow and a strong forced vortex might well combine to extend rotation to the impeller centerline, i.e., might introduce prerotation of inlet flow. Measurements indicating that static pressure at the tip is close to $u_2^2/2g$ reinforce this view. Also, inlet prerotation is indicated by quite respectable suction performance despite radial blade inlet geometry. Thus, the theoretical head normally used in practice simplifies to

$$H' = \frac{u^2}{g} = \frac{u^2}{2g} + \frac{u^2}{2g} \tag{11-2}$$

When the subscript is dropped, u is taken to indicate the impeller tip speed. Actual head for P.E. pumps is then stated as:

$$H = \psi \frac{u^2}{g} \tag{11-3}$$

To further understand the workings of the P.E. pump, a somewhat simplistic exercise involving a mixture of theory and experience is put forth to establish how actual head is generated. As has been indicated, tall blade geometry produces a strong forced vortex resulting in a static head coefficient near unity, so the actual static head produced by the impeller is simply $u^2/2g$. Tests have shown that diffusion efficiency is nearly flat through much of the flow range, so we state that $\eta_d = .8$. Diffusion recovery potential is in accordance with the diffuser area ratio

$$\left[1 - \left(\frac{A_1}{A_2}\right)^2\right]$$

And finally, the P.E. flow coefficient is in the vicinity of $\phi = .8$. So actual head generation may be written

$$H = \frac{u^2}{2g} + \eta_d \frac{(\phi u)^2}{2g}\left[1 - \left(\frac{A_1}{A_2}\right)^2\right] \tag{11-4}$$

Say, then, that the diffuser terminates in two throat diameters, i.e., has an area ratio of 4, so the actual head should be

$$H = \frac{u^2}{2g} + .8\,\frac{(.8u)^2}{2g}\left[1 - \left(\frac{1}{4}\right)^2\right]$$

$$H = .74 \frac{u^2}{g}$$

This is to say that the estimated head coefficient in this breakdown is $\psi = .74$, which falls within the $\psi = .70$ to $.75$ range typically occurring in test experience.

No pretense exists about the theoretical elegance of the actual head exercise, but it does provide some insight into the workings of the pump. It is clear that roughly 2/3 of the total head is provided by the forced vortex in the bowl and that the remaining 1/3 comes from diffusion recovery. Or, for example, assume that it would be possible to improve diffuser efficiency to 90% as is attainable in the relatively idealized case of a venturi meter, thus increasing the head coefficient to $\psi = .77$. Using this result, we go to the following expression relating head coefficient to efficiency:

$$\eta_2 = \eta_1 \frac{H_2}{H_1} = \eta_1 \frac{\psi_2}{\psi_1} \tag{11-5}$$

Then, assuming an original pump efficiency of 60%, the improved diffusion recovery would increase the pump efficiency to 62.4%. A dramatic (and probably unachievable) 10% improvement in diffusion recovery would dilute to 2.4% improvement in the overall pump efficiency. Similarly, truncation of the diffuser cone from an area ratio of 4 to an area ratio of 3 would reduce the overall efficiency only from 60% to 59%.

It is useful to express head in convenient terms. Non-homogeneous units are used in this chapter as is commonly done in everyday practice, so constants in the main result from unit conversions. Impeller tip speed is:

$$u_2 = \frac{\pi D_2 N}{720} \tag{11-6}$$

Specific Speed

To those familiar with algebra, but unfamiliar with pump technology, it would appear that specific speed, described in Chapter 2, can be altered by simply changing the rotational speed. Not so. To illustrate this, we note from the affinity laws (also described in Chapter 2) that flow is proportional to speed and head is proportional to the square of speed. Start with a given pump with a specific speed of:

$$N_s = \frac{NQ^{.5}}{H^{.75}}$$

Then, if the rotational speed is changed by some factor x, we have:

$$N_s = \frac{xN(xQ)^{.5}}{(x^2H)^{.75}} = \frac{NQ^{.5}}{H^{.75}}$$

So we see that specific speed is unchanged with change in rotational speed; head and flow change in such a way as to keep specific speed constant.

Mathematically, the specific speed expression is seen to vary from zero at zero flow or shutoff, to infinity at zero head or cutoff. By definition, specific speed has meaning only at the best efficiency point or flow rate at which maximum efficiency occurs. Then, the head expression can be expanded to read

$$H = \psi\frac{u^2}{g} = \psi\left(\frac{D_2N}{1{,}300}\right)^2 \tag{11-7}$$

Flow is proportional to the product of tip speed and discharge throat area according to the following expression

$$Q = \frac{720}{231}\phi u A_1 = \frac{\pi^2}{924}\phi D_2 N d_1^2 \tag{11-8}$$

In the flow expression, it is seen that the flow coefficient, ϕ, is simply the ratio of discharge throat velocity to impeller tip speed. It is at first surprising to note that $\phi = 1.3$ at cutoff indicates that throat velocity is outrunning the impeller tip speed by 30%. This is in reality the case, and is explained by conversion of the bowl static head into velocity head so that total head, less losses, appears as velocity head in the discharge throat.

Further perception of the meaning of specific speed for P.E. pumps is desirable. To provide this insight, we simply insert the head and flow expressions of Equations 11-7 and 11-8 involving impeller diameter, throat diameter and speed into the specific speed expression

$$N_s = \frac{NQ^{.5}}{H^{.75}}$$

which yields an alternate expression for specific speed:

$$N_s = 4{,}847\frac{\phi^{.5}d_1}{\psi^{.75}D_2} \tag{11-9}$$

This expression may be described as the geometric form of specific speed and shows first that specific speed is independent of rotational speed, and secondly that specific speed is basically defined by the ratio of emission throat diameter to impeller diameter, i.e., is related to the ratio of flow capacity to head capacity of the pump stage. Since the head and flow coefficients of the P.E. pump do not range broadly, the specific speed of a given P.E. pump geometry is expressed approximately as:

$$N_s \approx 5{,}500 \frac{d_t}{D_2} \tag{11-10}$$

Accumulated experience reflecting the efficiency potential of well-designed pumps versus specific speed are shown in Chapter 2. Impeller geometry trends toward relatively large diameters and small flow passageways as specific speed decreases.

A first observation is that pumps with the highest efficiency potential have a specific speed in the neighborhood of 2,000, and that efficiency starts to drop substantially for specific speeds below 1,000. The fundamental reason for lowered efficiency potential at low specific speed lies in the disproportionate losses incurred in low specific speed design, particularly disk friction and flow losses. Disc friction, neglecting a modest Reynold's number modifier, is well known to vary as the cube of speed and the fifth power of diameter. Pump power is proportional to the product of pump head and flow or the cube of impeller diameter and speed, $(DN)^3$. Then, without pretense of mathematical completeness, the impact of disk friction on efficiency can be expressed as follows:

$$\eta = \frac{\text{Output}}{\text{Input}} = \frac{\text{Output}}{\text{Output} + \text{Loss}} \propto \frac{D^3N^3}{D^3N^3 + D^5N^3}$$

This expression illustrates the disproportionate influence of disk friction on efficiency for low specific speed pumps which tend toward large diameter impellers. Further, for a given head objective, design choices are such that the product of DN is a constant, so indicating the general advantage inherent in selection of high speed in return for a smaller impeller diameter.

A second observation is at first disappointing in that a family of dimensional parametric curves indicative of pump size appear on the otherwise dimensionless N_s—η plot. Small pumps are always less efficient than hydraulically similar large pumps. The prime reason for this scale or size effect is mostly easily explained by a pipe flow analogy: skin friction arises from the inner circumference of the pipe and so is proportional to the diameter, while flow or throughput is proportional to the cross-sectional area and so is proportional to the square of the diameter. Small pipes thus experience relatively higher flow loss than do large pipes. In fact, pipe friction data provide excellent corollary with the $N_s-\eta$ data for pumps in that pipe diameters commonly appear as parameters on a dimensionless plot of friction factor versus Reynolds Number.

With specific speed as well as head and flow expressions having been defined for P.E. pumps, convenient expressions for impeller and throat size may be derived

$$D_2 = \frac{1{,}300}{N}\left(\frac{H}{\psi}\right)^{.5} = \frac{1{,}300}{N_sH^{.25}}\left(\frac{Q}{\psi}\right)^{.5} \qquad (11\text{-}12)$$

$$d_1 = .268\left(\frac{Q}{\phi}\right)^{.5}\left(\frac{\psi}{H}\right)^{.25} \qquad (11\text{-}13)$$

Power for any pump is

$$HP = \frac{HQ(SG)}{3{,}956\eta} = \frac{pQ}{1{,}714\eta} \qquad (11\text{-}14)$$

The concentric bowl pump has been unjustly criticized as having only low efficiency potential, probably because this pump type is frequently designed for very low specific speed where only low efficiency potential exists. Barske states that efficiency was of secondary importance in his development efforts, yet reports an efficiency island of 57% in the vicinity of H = 1,000, Q = 40, N = 28,000 (N_s = 1,000). This is seen to be representative of good pump performance as indicated by the general pump population data discussed in Chapter 2.

Because partial emission pumps range so widely in speed, it is sensible to use impeller diameters for scale or size parameters on $N_s-\eta$ maps, rather than the flow parameters widely used for the higher specific speed types. Direct comparison of P.E. and F.E. efficiency potentials from these data is a little elusive since these maps define explicitly only two of the four parameters involved in the specific speed expression. But by making the quite reasonable assumption that the low specific speed data collected by Karassik derived from pumps at 3,600 RPM, direct comparison can be made as shown in Figure 11-3. The dotted curves reflect the Karassik data and the solid curves represent P.E. performance at 3,600 RPM. Distinct P.E. efficiency superiority is seen to exist at low specific speeds and low to medium flow rates.

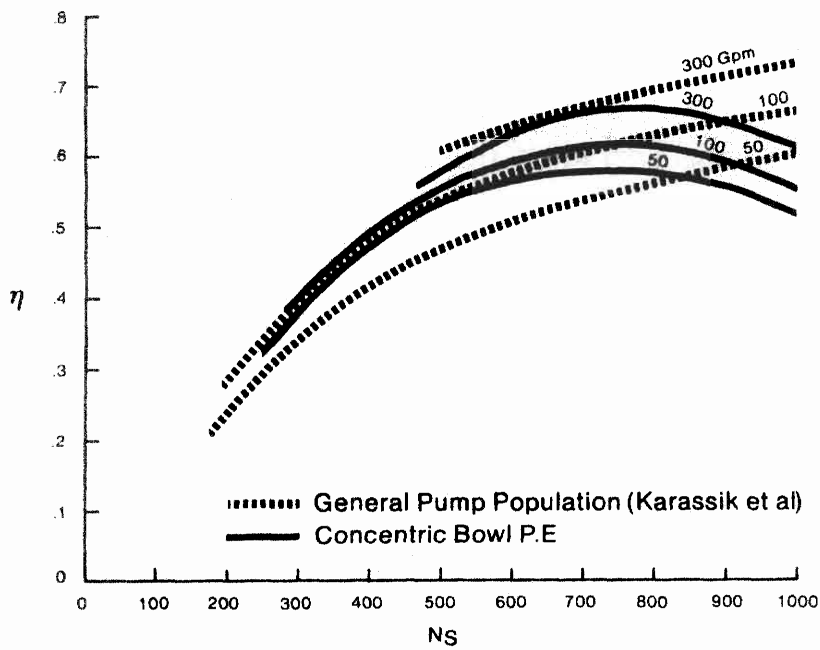


Figure 11-3. Partial emission efficiency comparison.

Concentric bowl pump peak efficiency occurs at about $N_s = 800$, and declines in efficiency at higher specific speeds as shown. This characteristic of P.E. pumps exists in general because with high specific speeds the inlet and discharge passages enlarge toward overlap causing declining head coefficients and efficiency reduction. This characteristic establishes a boundary region delineating the suitability of P.E. and F.E. pump types.

Although very low specific speed design must inherently entail efficiency sacrifice, such design can have overall attraction. Efficiency is not the sole consideration in pump selection, and can be overridden by factors such as simplicity, low initial cost, and quick, easy maintenance. These alternative considerations tend toward dominance at modest power levels and for intermittent or low-usage services.

Suction Specific Speed

Another dimensionless parameter highly important in pump design is known as suction specific speed involving a parametric group nearly identical to the pump specific speed expression. This subject is discussed in detail in Chapters 2 and 8.

$$S_s = \frac{NQ^{.5}}{NPSHR^{.75}}$$

Suction specific speed ranges typically from S_s = 7,500 to 10,000 for P.E. pumps, which is in about the same range as for higher specific speed pumps of single suction, overhung impeller design. High S_s values translate into low NPSHR, or good suction performance. The range mentioned varies with flow rate where high S_s values are associated with low flow rates.

Solving the specific speed equation for NPSHR yields a suction expression in terms of speed and flow, which can alternately be converted into terms of head and specific speed:

$$NPSHR = \frac{N^{1.333}Q^{.666}}{S_s^{1.333}} = H\left[\frac{N_s}{S_s}\right]^{1.333} \tag{11-15}$$

Contrary to what might be expected, the inlet radial blade geometry used in P.E. pumps achieves suction specific speed parity with the higher specific speed pumps utilizing more sophisticated inlet shapes. It is apparent, however, that high speed pumps will be demanding from the standpoint of NPSHR. The bracketed term in the latter expression is known as the Thoma cavitation parameter, usually designated by sigma:

$$\sigma = \frac{NPSHR}{H} = \left[\frac{N_s}{S_s}\right]^{1.333} \tag{11-16}$$

The Thoma parameter, then, states NPSHR as a fraction of pump head and is a function of the ratio of specific speed to suction specific speed. Low specific speed thus offsets to a degree the higher NPSHR associated with high speeds.

The inlet eye size in the prior expressions is assumed to be generously sized, as is generally done so that only small NPSHR impact exists. The NPSHR expression expanded to include inlet eye size effect becomes:

$$NPSHR = \left[\frac{NQ^{.5}}{S_s}\right]^{1.333} + \frac{Q^2}{386D_e^4} \tag{11-17}$$

Inlet eye size has been found to have an influence on the efficiency potential of the pump, which as we have just seen, affects NPSHR. Availability of efficiency advantage via eye sizing then in reality hinges on NPSHA in the application. More will be said on this subject in the section "Partial Emission Design Evolution."

As an aside, pump users should be aware that overly conservative statements of NPSHA in an application can work to their disadvantage. The pump manufacturer must meet the stated NPSHA, so understated suction conditions can force the design toward lower speed or more and larger stages, which can result in an efficiency penalty or higher initial cost.

Inducers

Need to improve suction performance becomes quickly apparent in the move toward exploitation of high speed advantage. Inducer development began more than 50 years ago to provide this improvement. An inducer is basically a high specific speed, axial flow, pumping device roughly in the range of $N_s = 4{,}000$ to $9{,}000$ that is series mounted preceding a radial stage to provide overall system suction advantage. Inducers are characterized by relatively few blades, shallow inlet blade angles, and generally sophisticated hydraulic design.

The inducer must put up enough head to satisfy the needs of the radial impeller stage but in itself has a suction level requirement that establishes a new lower NPSHR for the system. Inducers are an important element in high speed pump design, and so have been and continue to be the subject of considerable interest and developmental work. Inducer design should be such that maximum suction performance is achieved, and such that cavitation erosion in the inducer itself is avoided in long-term operation.

Inducer performance is generally taken as the suction specific speed which corresponds to 3% pump head depression as NPSH is decreased. Theory exists establishing optimum suction performance in an expression known as the Brumfield criterion. A form of the Brumfield criterion developed in a comprehensive document on inducer design developed by NASA is as follows:

$$S_s = 3{,}574 \frac{(1 - 2\phi^2)^{.75}}{\phi} \tag{11-18}$$

Where ϕ is the inlet flow coefficient or the ratio of meridianal flow velocity to inducer tip speed:

$$\phi = \frac{93.62}{D_i^3}\left(\frac{Q}{N}\right) = \tan \beta \tag{11-19}$$

A plot of the Brumfield criterion is shown in Figure 11-4. It should be emphasized that this expression is theoretical but tempered by practical

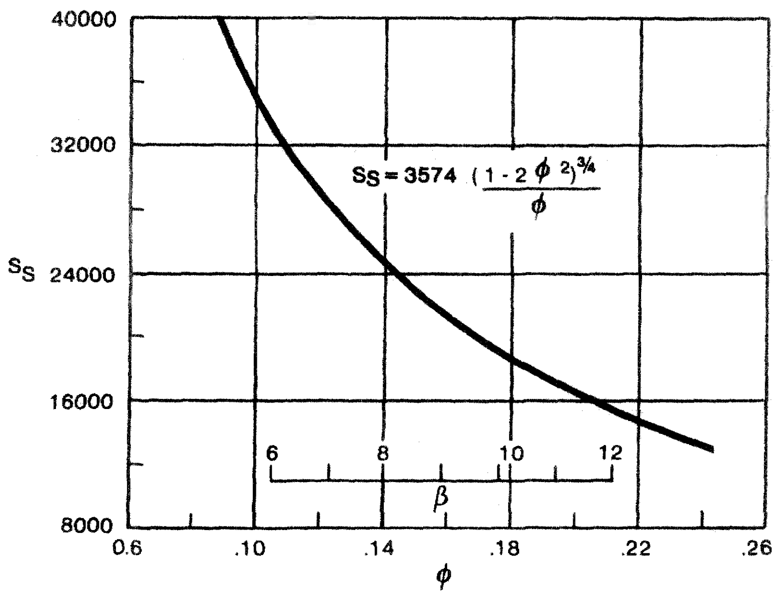


Figure 11-4. Brumfield performance criterion.

design considerations and that many details of inducer design are not addressed. The angle β is the fluid angle at inlet and differs from the blade angle, β_1, by a positive incidence angle, α. Prerotation is assumed to be zero, and other considerations such as blade shape, blockage, hub geometry, and leakage are simply ignored. The intent here is primarily to show the fundamental influence of the inlet blade angle on suction performance potential.

Optimal inducer design is distinctly a high-tech endeavor which must conform to hard-earned design guidelines and hydraulic disciplines. A well-designed inducer should possess a "sharp" breakdown characteristic as illustrated by the solid curve in Figure 11-5, rather than the "gentle" curve shown in broken line. The NPSHR disadvantage with gentle breakdown is evidenced by the NPSHR differential which exists at the 3% head depression level. It should not go unnoticed that the 3% head depression level refers to the inducer-pump combination, so the level of the 3% line on the inducer headrise curve will vary according to the head of the pump to which the inducer is coupled. Lower head units will suffer an NPSHR disadvantage with a gentle breakdown inducer compared to high head units equipped with the same inducer.

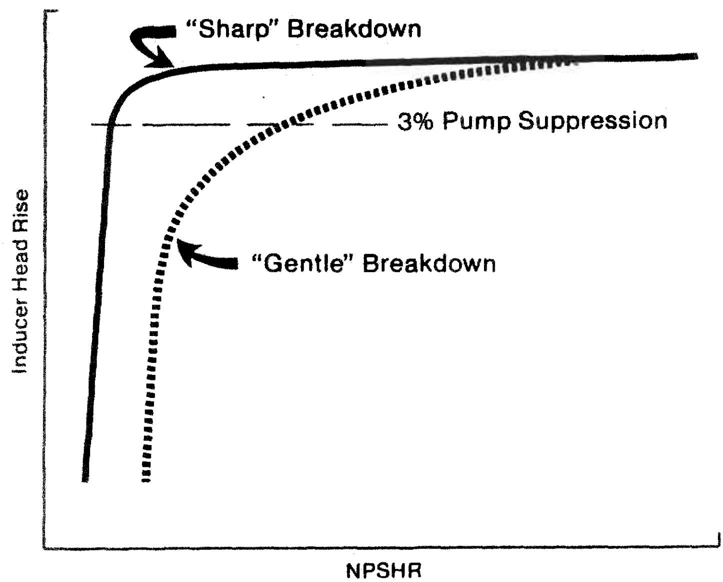


Figure 11-5. Inducer characteristics curves.

Commercial inducers have tended to result in performance suction specific speeds on the order of $S_s = 20{,}000$ to $24{,}000$. This level of performance entails some compromise from the standpoint of consideration of inducer cavitation but provides respectable inducer design, though perhaps not ultimate design. These suction specific speeds generally produce dramatic improvement in suction performance, frequently providing up to 80% NPSHR reduction over uninduced pumps.

Freedom from long-term cavitation erosion to the inducer itself is provided by observing experimentally established cavitation limitations in the inducer design. The cavitation limitation is related to tip speed, fluid specific gravity, and the inducer material as follows:

$$S_s = \frac{K}{u_i(SG)} = \frac{720K}{\pi D_i N(SG)} \qquad (11\text{-}20)$$

where K is an experimentally established constant which varies with the inducer hydraulics and material of construction.

Commonly used 316 stainless steel is considered to possess "good" resistance to cavitation erosion, but exploitation of materials with superior cavitation resistance and a given level of hydraulic sophistication may well pay dividends in excess of their added cost.

The trends associated with the performance and cavitation criteria for inducers are shown in Figure 11-6. The ideal inducer size lies at the intersection of the curves, where maximized performance is provided within cavitation limitations. By limiting consideration to flow coefficients of $\phi < .2$ and combining the Brumfield and cavitation limit criteria, the following expression results, providing means of estimating the optimum inducer diameter

$$D_i = \frac{(6KQ)^{.25}}{N^{.50}(SG)} \tag{11-21}$$

Exact parity between the performance and cavitation criteria is, of course, not essential in each inducer application. The challenge in designing a family of inducers is to provide the most useful combinations of characteristics within a reasonable family size.

This section has presented a superficial overview of a complex subject: inducer design and application. A few closing remarks are in order. Brumfield does not represent an ideal inducer such as with paper thin

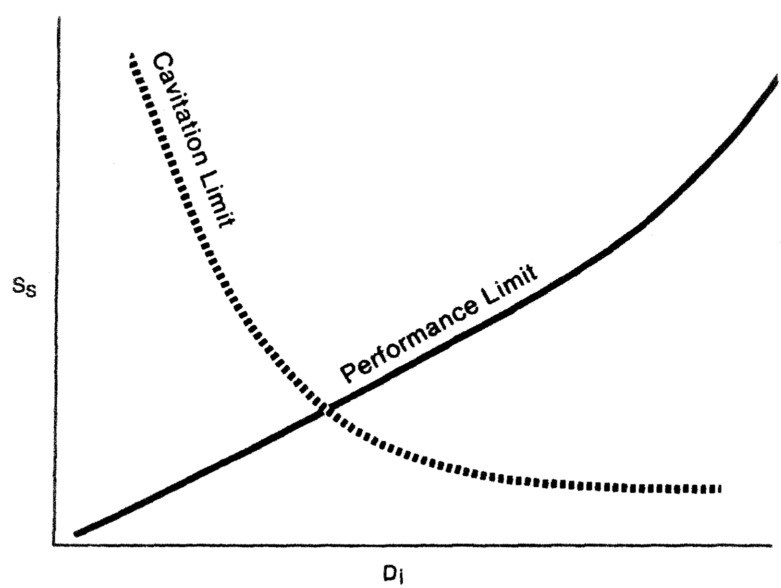


Figure 11-6. Trends of performance vs. inducer diameter.

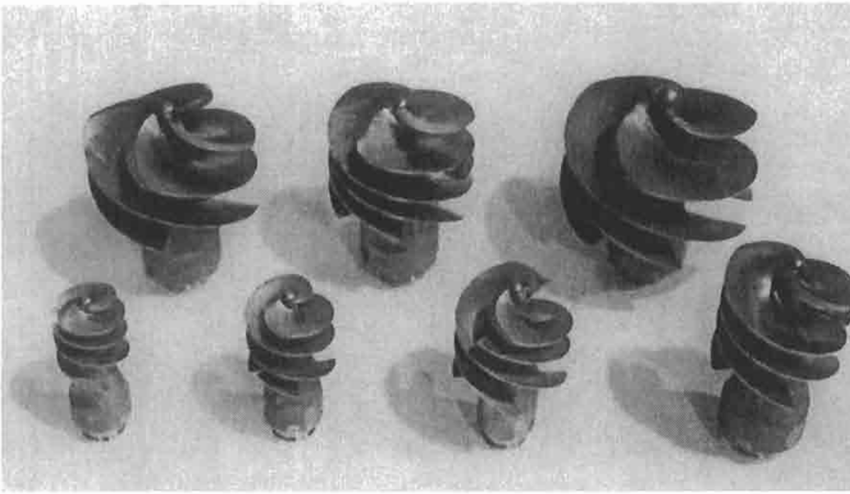


Figure 11-7. Inducer family for partial emission pumps (courtesy Sundstrand).

blades, razor sharp edges, etc., but rather a useful "optimum" relating reasonable suction performance expectation to the inlet flow coefficient. The Brumfield suction performance can be exceeded with well-designed inducers. Tradeoff or compromise is not always required in performance and cavitation considerations, because modest speed, small inducers, or low specific gravity fluids can result in operating regimes far removed from cavitation concern. Substantial effort has been devoted to inducer development and will undoubtedly continue in the future, since inducers are a key element in extending the frontiers of high-speed pump technology.

A family of inducers for P.E. pumps is shown in Figure 11-7 which range from 1.25 to 3.5 inches in diameter and provide coverage from Q/N = .0005 to .1.

Partial Emission Design Evolution

Beyond the very substantial NPSHR improvement provided by inducers, improvement of the concentric bowl pump has been pursued in other areas including efficiency, curve shape, and noise reduction. Positive results have been achieved in all three of these areas as summarized below.

Areas exist where the concentric bowl pump is wanting by a few efficiency points to be more fully competitive with other pump types. A clue

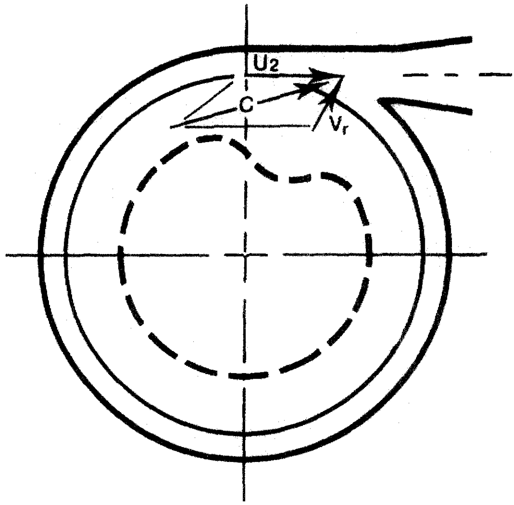


Figure 11-8. Diagram of concentric bowl static pressures and flows.

to a basic hydraulic "fault" in the concentric bowl pump exists in the eye-to-throat flow path described in the section "Terminology." The sketch in Figure 11-8 illustrates a concentric bowl pump, where the dashed line represents a polar plot of static pressure within the bowl. The static pressure is depressed in the vicinity of pump discharge, and this depression increases with increasing flow rate. This indicates unfavorable exchange of static head for velocity head in the area of discharge, which must be reconverted to static head in the diffuser. Furthermore, fluid approaches the discharge throat in the direction indicated by vector c, the vector sum of u_2 and v_r, detracting from the diffuser recovery potential.

These adversities can be eliminated by abandoning the concentric bowl geometry in favor of a volute collector geometry. This modification has been shown to improve efficiency by about 6 points with specific speeds in the range of $N_s = 800$ to 1,000, but this advantage fades to parity with the concentric bowl configuration at specific speeds of about $N_s = 300$ to 400.

Radial side load results from standing pressure variations around the impeller periphery. These hydraulically imposed radial loads are proportional to the product of pump head times the projected area of the impeller, and must be reckoned with from the standpoint of bearing loads. Side load trends vary dramatically with the pump design geometry as indicated in Figure 11-9, where magnitudes are shown in the upper plot and vector direction trends are shown by the polar plots within the lower fig-

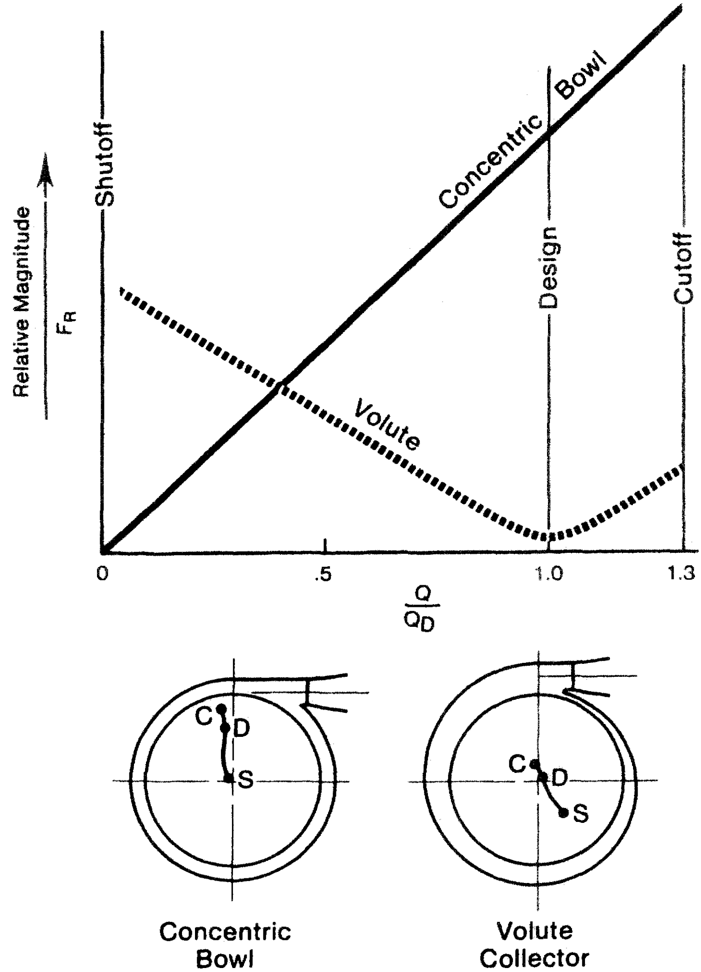


Figure 11-9. Radial load trends.

ures along with the letters S, D, and C, indicating shutoff, design, and cutoff flow rates. The concentric bowl side load increases continuously with flow and is always oriented in the general direction of the discharge throat. The volute radial load virtually vanishes at design flow, but undergoes about a 180° reversal in direction over the full flow range. These considerations are significant when fluid film bearings are used and the lube feed spreader groove location must not encroach into the bearing load zones.

Change to a volute collector raises the question of terminology, since fluid exists via the full impeller periphery and it may be asked whether or not the description "partial emission" should be abandoned. On balance, tall radial blade geometry is carried over from the concentric bowl design, so the impeller emission area remains much larger than the discharge throat area. Disorderly flow conditions similar to those in concentric bowl design prevail due both to retention of tall blades and the radial inlet geometry. The volute modification perhaps is most accurately viewed as a P.E./F.E. transitional design, but we will choose here to remain with the P.E. classification, in part because this modification stems from the original concentric bowl concept to which the P.E. designation clearly applies.

Further efficiency improvement is possible through optimization of the inlet eye diameter. By analytic means, this optimum has been established as:

$$D_e = 5.1 \left(\frac{Q}{N}\right)^{.333} \tag{11-22}$$

Test experience has shown that optimum eye sizing can improve efficiency by about four points over that attainable with large eye diameters on the order of twice the optimum diameter, which are often used in the interest of minimizing NPSHR. Equation 11-17 shows that eye size influences NPSHR as a fourth power function of diameter, so freedom to exploit the eye size efficiency advantage often does not exist. For inducerless design and ample NPSHA, near-optimum eye sizing should always be used. This situation nearly always exists, for example, in stage 2 of series-staged machines.

Curve shape improvement and noise reduction have been achieved through use of high-solidity impellers, i.e., by adding more impeller blades, which increases the ratio of blade cord length to blade spacing. High-solidity impellers tend toward minimizing flow stratification and blade loading because the total power is divided between a larger blade complement. To obtain benefit with high solidity impellers, it is necessary that all blades penetrate equally into the impeller eye. Use of splitter geometry or blades alternating in length as is often done in turbomachines to avoid eye crowding results in dominance of the larger blades and provides no advantage in P.E. pumps.

A rising-to-shutoff or stable curve shape can be provided with high solidity radial-bladed impellers as shown by the solid curve in Figure 11-10. This is contrary to the generally held view that backswept blade design is essential to providing a stable curve. The reason that improved curve shape results with high-solidity impellers is believed to lie with im-

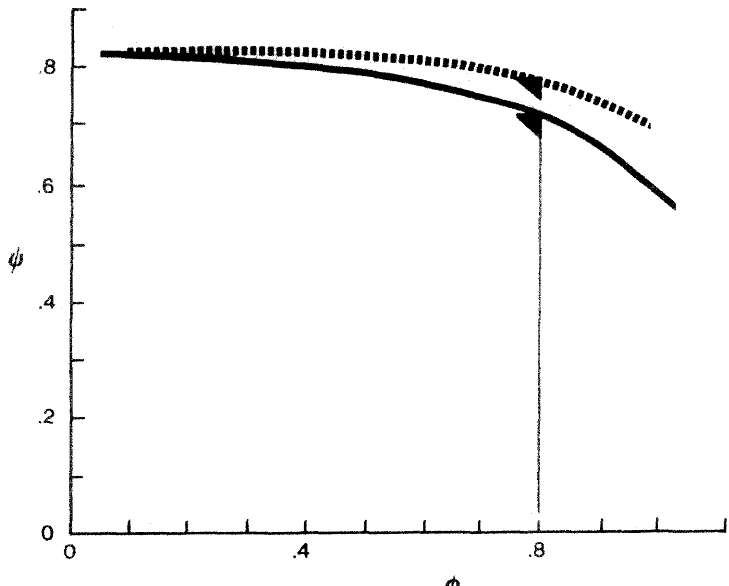


Figure 11-10. Curve shape variation with impeller eye size.

proved pitot recovery at low flow rates. Referring back to the concentric bowl characteristics shown in Figure 11-2, it is seen that shutoff head is about the same as design head. The ideal velocity head is $u^2/2g$, and some portion of this head may be converted to static head by either diffusion or pitot recovery. As flow is reduced, diffusion recovery potential decreases while pitot recovery potential simultaneously increases. The shutoff head coefficient would be expected to be only $\psi = .5$ without pitot recovery. A stable curve shape evidently results from improved pitot recovery provided by high solidity impellers.

As would be expected, increased impeller solidity results in an increased head coefficient. In "Partial Emission Formulae," the impeller blade inner diameter, D_1, was said to have small effect on pump performance, so was neglected in an illustrative exercise. But the impeller eye choice does in fact affect curve shape and efficiency with a volute collector, so presenting a designer's choice. A rising curve characteristic is achieved by setting D_1 appreciably larger than the inducer diameter. Choice of small D_1, or deeper blade penetration into the eye, provides higher design head and moderately higher efficiency, but results in a relatively flat curve shape.

It is interesting to note that the shutoff head remains unchanged with D_1 variation, so the characteristic curves "hinge" about the shutoff point. Although a rising curve is generally viewed as being desirable, it should be noted that many thousands of pumps have operated with satisfaction through the years despite an unstable curve characteristic.

Pump Noise

Noise generation becomes a subject of increasing concern with the increased power densities associated with sometimes very compact high-speed pumps. Investigation has shown that noise generation traces to hydraulic origins, and more specifically is at blade pass frequency in P.E. pumps. The underlying noise generating mechanism lies with pronounced flow stratification within the impeller, so that flow jets issue from the pressure side of each blade at the impeller exit. These jets impinge on the geometric anomalies in the discharge vicinity and produce pressure pulses which set the container walls in motion, which in turn broadcast airborne noise. Noise also propagates through metallic and fluid paths making noise control by means of lagging or enclosures extremely unattractive because the entire pumping system must be treated, including the pump, driver, base, and piping system.

High-solidity impellers have provided noise reduction typically on the order of 10 dbA, which viewed in different contexts translates into 90% sound power reduction but is perceived by the human ear as being half as loud. Roughly 5 dbA additional noise reduction is available by increasing the case size relative to the impeller size with virtually no sacrifice in efficiency. Industrial and governmental noise standards can always be met or exceeded with high-solidity impellers, whereas their low-solidity counterparts are sometimes marginal in this regard.

It should be noted that proper volute sizing is a prerequisite to achieving either the efficiency gains available with volute collectors or the noise and curve shape advantages associated with high-solidity impellers. Experience has taught that the cross-sectional area swept by the volute should be about 15% to 20% greater than the discharge throat area.

The photograph of Figure 11-11 compares low- and high-solidity partial emission pump impellers equipped with inducers. Advantages associated with the high-solidity impellers will result in this type supplanting in large measure their low solidity counterparts.

Design Configuration Options

High-speed partial emission pumps are well suited to provide high to very high heads. Inducers have augmented suction performance so that a

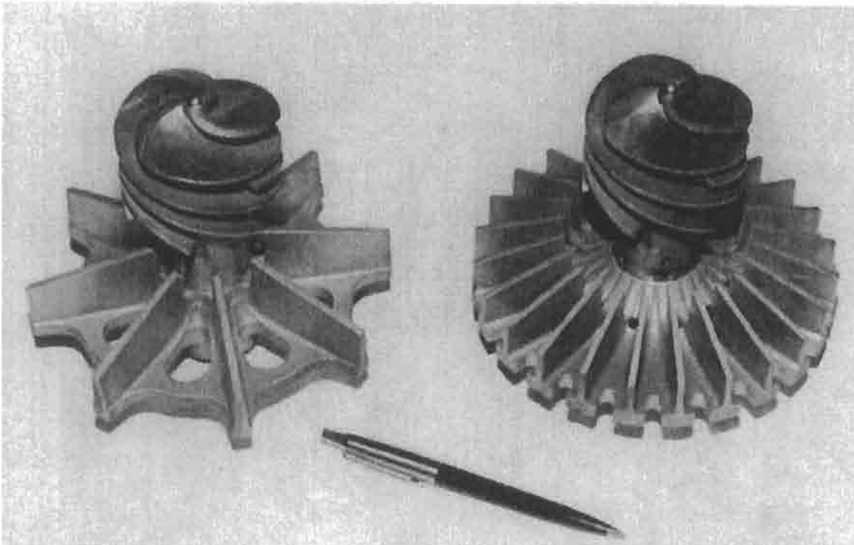


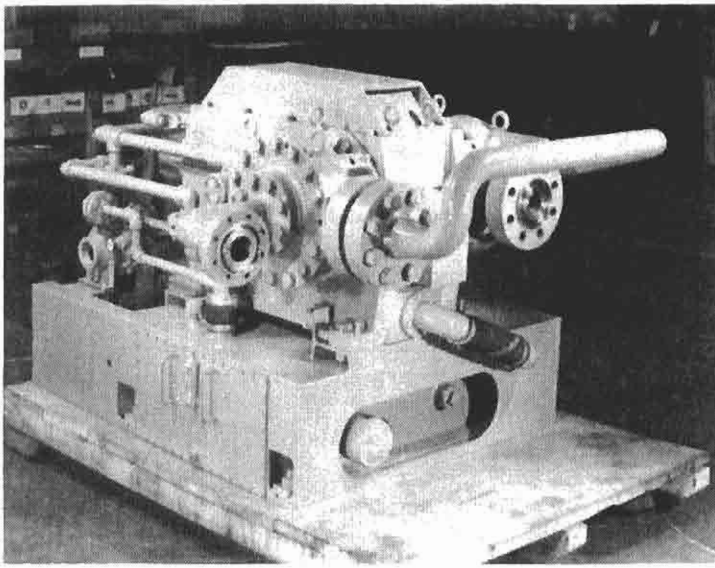
Figure 11-11. Comparison of high and low solidity impellers with inducers *(courtesy Sundstrand).*

majority of requirements can be satisfied by machines with single-stage simplicity. More difficult requirements can be met by staging arrangements that provide extremely high heads or very low suction requirements or a combination of both.

Samples of a family of pump designs that have been evolved to provide wide coverage and flexibility are illustrated in the collage shown in Figure 11-12 A-E, briefly described as follows:

A. Single-stage or two-stage HP to 1,500 and 2,500. Single-stage to H = 6,000, Q = 400. Series staging to H = 12,000, Q = 400. Parallel staging to H = 6000, Q = 800.
B. Three-stage same as Pump A with boost stage to provide extreme heads combined with low NPSHR.
C. Two-stage, two-speed. HP to 400 and 750. To H = 6000 with low NPSHR or H = 12,000 ft with ample NPSHR. Q to 400.
D. In-line vertical. HP to 50, 200 and 400. H = 6000 and Q = 400. Direct drive versions available to 75 HP.
E. Integral flange motor. HP from 1 to 200 in 3 size versions. H to 3500, Q to 400. Frame mounts optional.

As is readily apparent, a great deal of design capability and flexibility is available in this family of machines. The suction constraints associated



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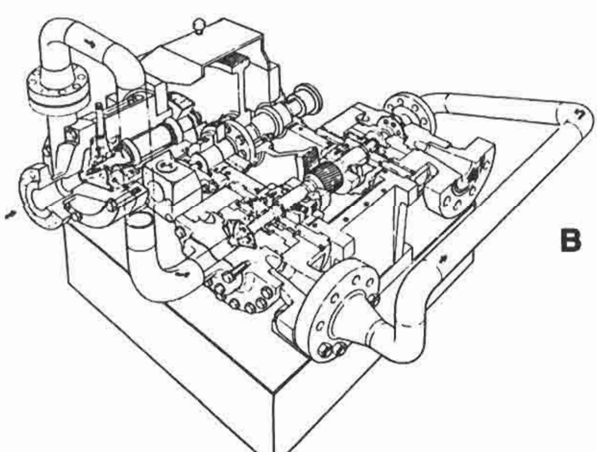
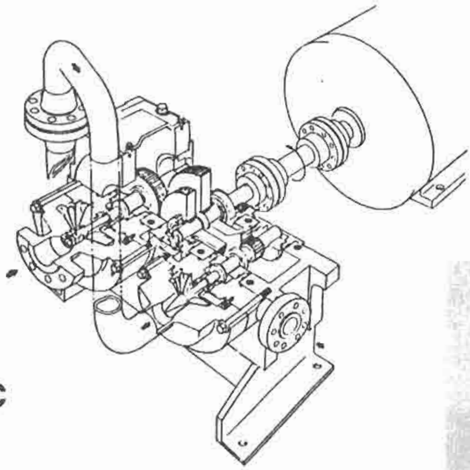


B

Figure 11-12. Examples of partial emission high speed pumps: (A) single- or two-stage pump; (B) three-stage pump; (C) two-stage, two-speed pump; (D) in-line vertical pump; (E) integral flange motor pump.



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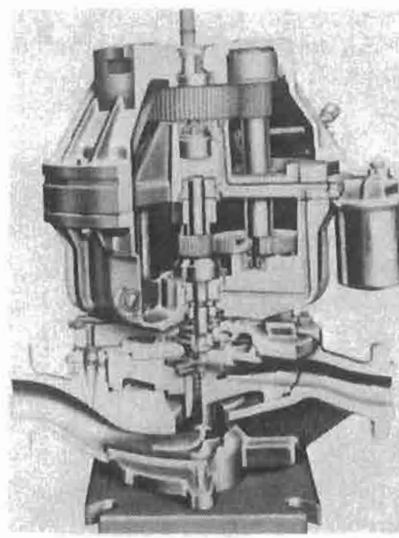

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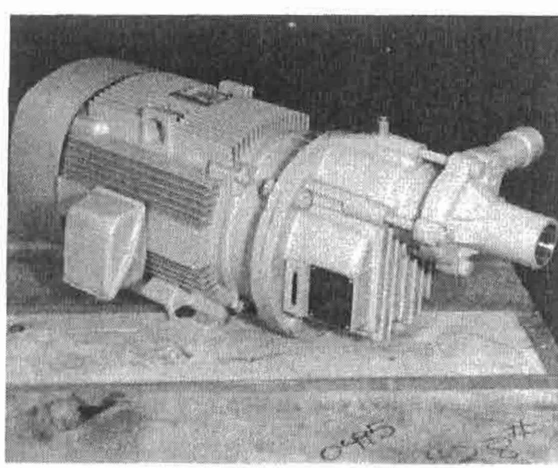

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with high-speed pumps can be virtually eliminated without efficiency sacrifice via inducer and staging options. It is interesting to note that a two-stage machine with equal head split optimizes at 60% of single-stage speed, which can reduce NPSHR up to about 70%.

Other High Speed Considerations

Aside from hydraulics, a number of other considerations exist in high-speed design. Several of these design facets are touched upon in the following sections. Each is broad in scope so can only be briefly highlighted here.

Stress and Deflection. Commercial availability of stage heads up to about 3,000 feet has been indicated by Karassik for full-emission, high-speed pump types. As has been indicated, partial-emission design allows heads to 6,000 feet per stage even with relatively low-strength 316 stainless steel material, this potential accruing from rugged blade impeller design. Simple impeller geometry allows easy extension of this head limit, if such need arises, through use of high strength-to-weight materials such as 17-4 PH stainless steel or titanium alloys.

Size reduction associated with high-speed design is dramatically illustrated in Figure 11-13 showing high- and low-speed multi-stage rotors with equivalent pumping capabilities. Fewer high-speed rotors and the exponential relationships of span and shaft diameter combine to allow geometries with lower shaft deflection in the high-speed design.

Gears. Speed-increasing gearboxes are generally required for pump drive speeds above electric motor limits. Integral gear systems are frequently used with single overhung impeller design, where the gearbox high-speed shaft doubles to support the pump impeller. Multi-stage machines are usually designed with straddle-mounted rotors and free-standing gearboxes, so requiring strict attention to alignment and high-speed coupling design.

The single most important attribute of high-speed, high-pitch line-velocity gearing is precision. Hardened and ground gearing is attractive because modern gear grinding equipment provides very high precision capability, along with substantial size reduction over soft gearing. Hardened gearing does not undergo geometric change during break-in, so the required gear and mounting precision must exist at assembly.

Industrial gearing is generally designed in accordance with American Gear Manufacturers Association (AGMA) specifications in which complete design guidelines are presented. Gears corresponding to AGMA

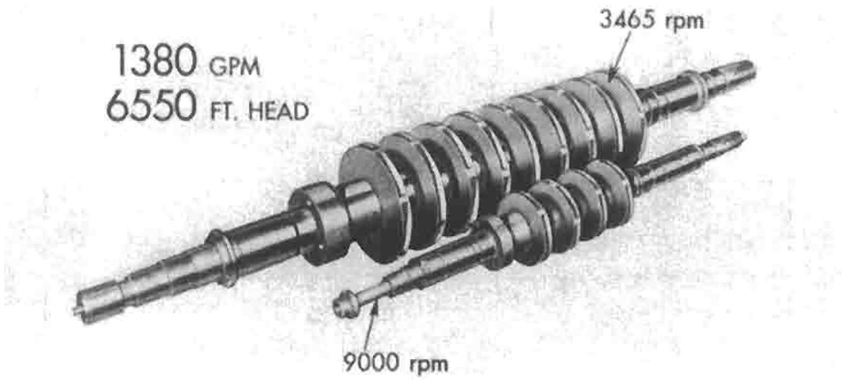

Figure 11-13. Multi-stage high and low speed rotor comparison (courtesy Worthington Division, McGraw Edison Company).

precision class 10 to 12 are commonly used in moderate- to high-pitch line-velocity gearing. The American Petroleum Institute (API) also publishes gearing specifications that are derived from AGMA, but demand more design conservatism. AGMA ratings compare to API ratings roughly in a ratio of 5:3.

Gear design considers ratings from two standpoints: strength and endurance. Strength rating is based upon evaluation of the gear tooth as a cantilever beam, and dominates in lower-speed, high-torque situations. Control of the case depth is important in hardened gear design to avoid through-hardening, or brittle teeth. Endurance rating evaluates gear design from the standpoint of wear resistance and becomes increasingly dominant with increasing pitch line-velocities. The lesser of the strength and endurance ratings at a given operating condition establishes the gearing rating. Best balance between strength and endurance results from coarse tooth selections for the lower operating speeds to fine-tooth selection in the high-speed ranges.

A tendency has existed to select spur gears for moderate power transmission and helicals for the higher power ranges. Spur gear geometry forces design with a contact ratio between 1 and 2, that is to say that the load is alternately carried by a single tooth or shared by a pair of teeth. Rating, then, is based upon single-tooth contact. Helical gears provide smooth meshing and continuous multi-tooth contact, and so in theory provide substantial increased capacity within a given envelope. This helical advantage, however, is highly dependent on gear precision, and is usually assumed to provide added design margin rather than increased capacity rating.

Helical gears introduce thrust which must be reacted by the bearing system. Thrust can be eliminated by use of herringbone gear design, but this option loses some packaging attraction with ground gears in that a wide central groove must be provided between the gear working halves for grinding wheel runout. Piecing helical gears back to back to provide a herringbone design detracts from already stringent precision requirements. Helical gears are generally (but not universally) believed by gear authorities to offer the advantage of lower noise, but here again precision looms more important than the spur-versus-helical choice per se. In any event, hydraulic noise in high-speed pumps has been found usually to overwhelm gear noise, divorcing noise consideration as a factor in gear-type selection.

For P.E. pumps, where speed is always tailored to a given application, desired speed is provided by simple gear size selection of standardized gears to fit within standardized gearboxes. Rarely do power and speed combine to require the maximum gearbox rating, so most often an added design margin exists at the rated power in a given application.

Bearings. Ball bearings have evolved to a high state of perfection and are attractive from the standpoints of low friction loss and modest lube system demands. Roller bearings have higher capacity than ball bearings, but are not well suited for high speeds due to a tendency of the rollers to skew in operation.

In general industrial equipment, the API guidelines are sometimes viewed as being unnecessarily conservative, and are modified in the interest of simplicity and low cost. Life projections should be tempered by full realization that only contact stress is considered, and that the quality of lubrication and many other practical aspects of bearing application are not addressed. In any event, high-speed/high-power design imposes bearing demands which soon outrun any realistic expectations of design adequacy with rolling contact bearings.

Hydrodynamic bearings possess capability to operate for indefinite periods at high load levels and high speed. This bearing type is self-acting with a film of lubricant separating the bearing elements in steady-state operation, precluding metallic contact and thus providing zero wear. The term "thick film" is used to describe these bearings, but this description must be taken in context since "thick" usually implies film heights of only a few ten-thousandths of an inch. Metallic contact cannot be tolerated in high-speed bearings, so the need for precise alignment is obvious. Materials selection is important largely because boundary lubrication or rubbing contact exists during start-stop cycles where full fluid-film separation cannot be achieved.

Plain journal bearings have excellent capacity and are nearly always suitable at pump speeds. For extreme speeds and powers, tilting pad journal bearings are sometimes used to take advantage of their excellent stability characteristics. Plain thrust bearings are inexpensive, and are generally used up to their load limits of 100 to 150 psi. More severe thrust loads require use of tilting pad thrust bearings with about 500 psi unit load capacity.

Bearings are obviously important elements in high-speed design, but the temptation to oversize bearings in the interest of unwarranted design conservatism should be suppressed because hydrodynamic bearing parasitic losses are not negligible.

Lube Systems. In small units equipped with ball bearings, lubrication needs can often be met with simple splash systems. Higher power units equipped with hydrodynamic bearings generally require a pressure lube system, including a lube pump, over-pressure relief valve, filter, and heat exchanger.

Free-standing lube pumps are sometimes used, but a pump driven from the gearbox input shaft is preferable, because lubricant is supplied during coastdown from high speed in the event of a power failure. Auxiliary lube pumps are sometimes required when start-up demands are severe. An example of this is an application with very high suction pressure acting over the shaft seal area producing high thrust at start-up. The thrust bearing must be copiously lubricated at start-up in order to survive the short-term boundary lubrication conditions existing until sufficient speed is achieved to provide lift-off to full film separation. Large machines and machines with very high stand-by suction pressure are often equipped with auxiliary lube pumps to provide full lubricant flow and start-up.

Shaft Dynamics. Shaft dynamics is a rather complex discipline that has evolved substantially over the years to ever higher levels of sophistication along with other engineering sciences. The advent of modern computer technology has raised analytic prowess to heights which would be otherwise impractical if not impossible.

The dynamic behavior of a shaft is strongly influenced by the characteristics of the bearings upon which it is invariably mounted; the important bearing characteristics being the spring rate and the damping coefficient. Rolling contact bearings have high, but finite, spring rates as opposed to relatively low spring rates in fluid-film bearings. Critical shaft speeds decrease with decreasing bearing spring rates. The high spring rates of rolling contact bearings usually vary over only a narrow range, so past experimental spring rate information generally suffices in shaft dynamics analyses. For hydrodynamic bearings, the relationship between load and film height are well established, and the spring rate is calculated by taking the first derivative of the W/h relationship, dW/dh. It must be recognized that the hydrodynamic spring rate will vary with load, so the ranges associated with gear loading and impeller hydraulic loads must be considered in detailed analyses.

All this appears a bit intimidating at first glance, but in practice shaft dynamics has generally shown itself to present neither incessant nor insurmountable problems. It should be pointed out that critical speed operation is not always destructive. Cases exist where a shaft can be made to run continuously at its critical for long periods, but needless to say, comfortable margin should always be provided between critical and design speed. With a new product family, thorough critical speed analysis is used in the design phase, and the analysis is confirmed by test experience in the hardware phase. Need for a full-blown analysis for each minor pump variation is alleviated. Normally, each production pump is tested at full rated capacity so any dynamic distress can be detected and corrected prior to shipment. Single-speed machines offer advantage in this regard since change of shaft stiffness, bearing stiffness, rotating mass, or a combination of all three can often cure a problem with modest hardware alteration.

Field problems with shaft dynamics are by far the exception rather than the rule. Such exception has an increased chance of occurring when full field operating conditions cannot be duplicated in the manufacturer's test facility. For example, water is the universally used test fluid, so pumps designed for low gravity fluids must often be operated at off-design conditions to simulate their full-power or full-speed operating characteristics. Or, a pump can interact differently with a user's system or foundation than it does in the laboratory. The computer has proven to be an invaluable aid in such occasional situations when a problem occurs.

Instrumentation is readily available to continuously monitor machine health if so desired. Noncontacting probes can directly observe high-speed shaft motion and can be arranged to provide display, alarm, or shut-down in the event of trouble. But this option is generally reserved for large and costly equipment. It is probably safe to say that this instrumentation is seldom opted for in machines under a few hundred horsepower. Experience has shown that the reliability and endurance of high-speed machines can be assumed to match that of their lower-speed counterparts, so similar ground rules on protective instrumentation should apply.

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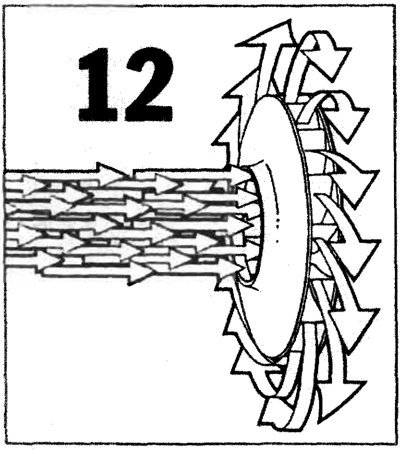

Double-Case Pumps

by **Erik B. Fiske**
BW/IP International, Inc.
Pump Division

Double-case pumps are also known as double-casing, barrel-case, or barrel-type pumps. They are used for temperatures and pressures above the range of single-case horizontally split, or diffuser-type multi-stage pumps, and pressures above the capabilities of radially split process pumps. The following are typical temperature and pressure limitations for centrifugal pumps. Horizontally split multi-stage pumps, such as those illustrated in Figures 6-1 and 10-13 are limited to operating temperatures of 400°F [1]. They can be designed for discharge pressures up to 4,000 psi, but often are limited to lower pressures by user specifications. Radially split process pumps are suitable for temperatures up to 800°F. Radially split double-suction, single-stage process pumps cover operating conditions up to 500 psi discharge pressure. Two-stage process pumps cover conditions to 1,000 psi. Double-case pumps are typically applied for operating conditions above these limits.

Configurations

Pump Casing

Double-case pumps got their name from being constructed with two cases: an inner case assembly that contains the complex shape of the stationary hydraulic passages and an outer case (barrel) that acts as the pressure boundary for the pumped fluid. The inner case assembly is subjected to external differential pressure, so any bolting required to hold it together is minimal. The outer barrel is designed as an unfired pressure vessel, and can be constructed to the requirements of well established industry codes.

Volute Casing with Opposed Impellers

Figure 12-1 illustrates the volute-type opposed-impeller configuration. The inner case assembly is horizontally split and consists of two identical halves, cast from the same pattern. The double-volute construction provides radial hydraulic balance. The opposed impeller arrangement minimizes resultant axial thrust, providing inherent axial thrust balance.

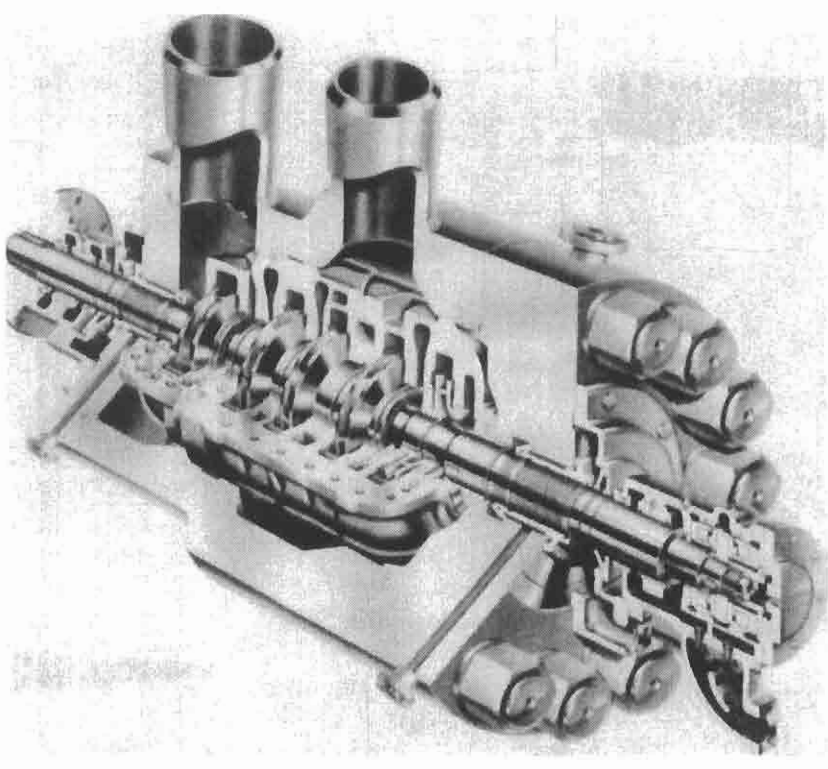

Figure 12-1. Volute-type opposed-impeller double-case pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

Diffuser Casings with Balance Drum

Diffuser-type pumps have all impellers facing in one direction, resulting in high axial thrust forces. Figure 12-2 shows a diffuser-type pump with a balance drum to carry the axial thrust forces. The inner case assembly is vertically split, and the symmetry of the diffusers provides radial hydraulic balance.

Diffuser Casings with Balance Disk

Figure 12-3 shows the diffuser-type configuration with a balance disk that carries the axial forces. Except for the axial balancing device, the construction is similar to the previous diffuser-type design. Balance disk construction is used for clean services such as boiler feed because of its ability to completely balance axial thrust at all operating flow rates.

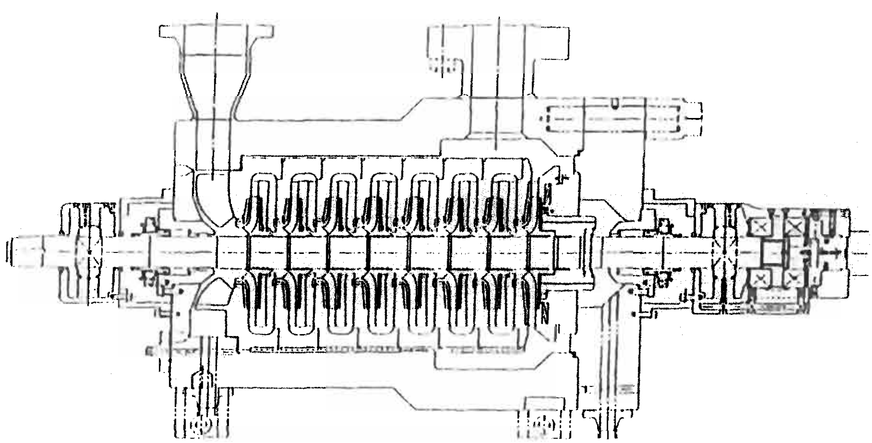

Figure 12-2. Diffuser-type in-line impeller, double-case pump with balance drum (courtesy Dresser Pump Division, Dresser Industries, Inc.).

Applications

Common applications for double-case pumps are:

- Boiler feed pumps in central station and large industrial fossil-fueled power plants.
- High pressure and/or high temperature pumps in oil refineries or chemical plants.

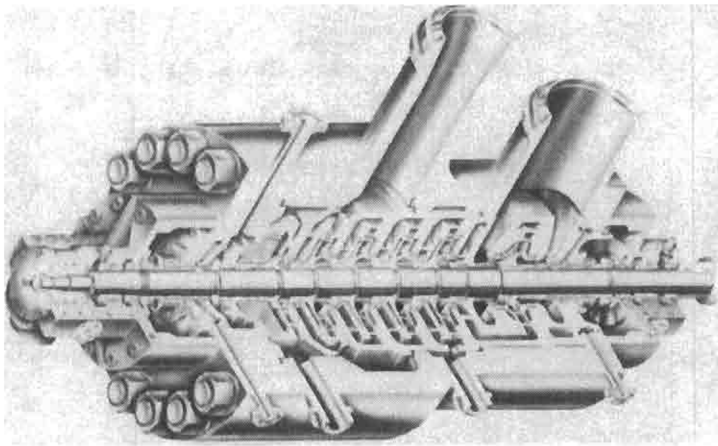

Figure 12-3. Diffuser-type in-line impeller, double-case pump with balance disk (courtesy Ingersoll-Rand Company).

- High pressure oil field water injection and offshore hydrocarbon condensate reinjection pumps.
- Pipeline pumps for unusually high pressures, very high vapor pressure hydrocarbons (typically above 200 psi), or offshore hydrocarbon condensate.

Boiler Feed Pumps

The most common application for double-case pumps is for boiler feed service in fossil-fueled power plants. These pumps must combine high efficiency with maximum reliability. Feedwater pump outages were estimated to have cost more than $408 million in replacement power alone in the United States in 1981 [3]. Several multi-million dollar efforts to reduce this cost have been implemented by users and manufacturers worldwide. These efforts have resulted in increased product knowledge that now can be applied to high-energy pumps, system design, and operation. Research in this area is continuing.

Charge Pumps

Oil refinery charge pumps handle liquids that are flammable and often toxic, at very high temperatures and pressures. Wide variations in viscosity of the feed stock or the presence of abrasives may add to pump design problems. In spite of inherent application problems, these pumps must combine maximum reliability with good efficiency.

Waterflood Pumps

Oil field water injection pumps operate at capacities to 5,000 gpm. Double-case pumps provide differential heads to 11,000 feet and discharge pressures to 8,000 psi from two pumps operating in series. This application is covered in more detail in Chapter 10.

Pipeline Pumps

The vast majority of pipeline pumps are of the horizontally split, multistage design, covered in Chapter 10. Double-case pumps are used only when unusually high pressures are required or when handling hydrocarbons near their supercritical condition.

Design Features

Removable Inner Case Subassembly

Modern double-case pumps have a fully separate inner case subassembly (including rotor). The inner case subassembly for a volute-type pump is shown in Figure 12-4. This subassembly can be removed, after disassembling the outboard cover, without disturbing the suction piping, discharge piping or the driver. It is common practice to have a spare subassembly available for replacement, thereby reducing maintenance turnaround time or the downtime caused by unscheduled outages.

If the pumped fluid is hot, time is needed to lower the temperature of the components before maintenance work can begin. Time to cool by ambient air is extended because the pump is normally well insulated. Forced liquid cooling can be helpful, but must be preplanned to avoid subjecting the pump to unacceptable thermal gradients.

In some designs the inner case subassembly includes the radial and thrust bearings. This feature further reduces downtime because the replacement rotor is aligned before the outage. A boiler feedwater pump of this construction, called "cartridge," "full cartridge," "pullout," or "cartridge pullout" design, is shown in Figure 12-5.

A saltwater injection pump with full cartridge pullout is shown in Figure 12-6. The configuration shown is said to save at least 40 manhours of labor, compared to conventional construction, each time the inner-case subassembly is replaced. This design features a springplate on the high pressure end to preload the internal gasket between the inner volute case and the outer barrel. This gasket seals the full differential pressure. The springplate design compensates for manufacturing tolerances to assure interchangeability among spare inner assemblies and also compensates

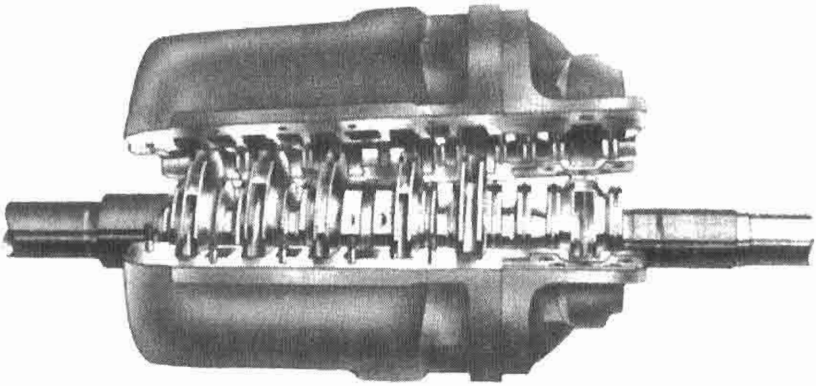

Figure 12-4. Inner case subassembly for a volute-type pump (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

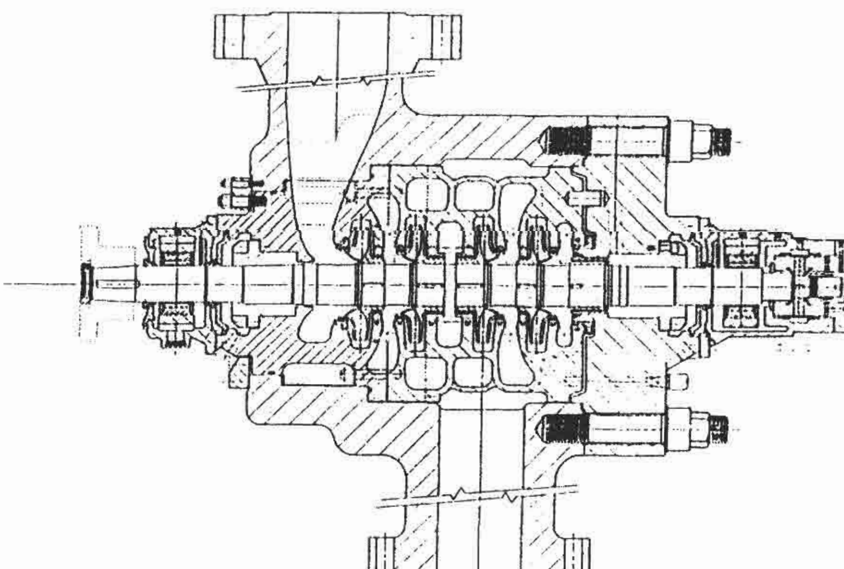

Figure 12-5. Boiler feedwater pump of cartridge pullout design to reduce maintenance turnaround time (courtesy Sulzer Bingham Pumps Inc.).

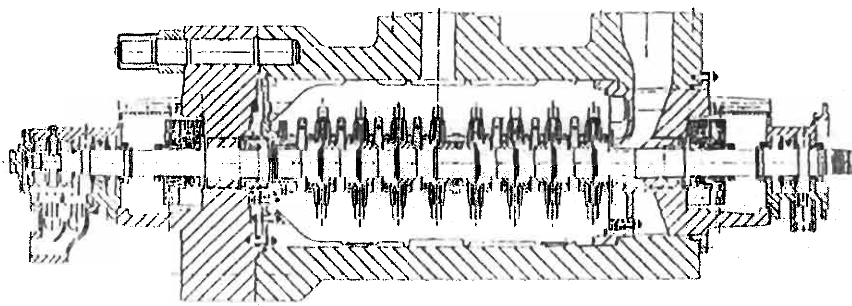

Figure 12-6. Water injection pump with full cartridge pullout, featuring a springplate on the high pressure end to preload the internal gasket that seals full differential pressure (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

for differential thermal expansion. Finite element analysis of the springplate assures that all design goals are achieved. This mechanical design, with the proper materials of construction, is also suitable for boiler-feed service.

The cartridge pullout design is especially advantageous when the pumps are located in an unfavorable environment, such as an offshore oil production platform. All critical assembly operations are performed in a service shop where high quality mechanical work is more easily achieved.

Auxiliary Take-off Nozzles

As shown in Figure 12-3, double-case boiler feed pumps are well suited for incorporating auxiliary take-off nozzles that are necessary when reheat and superheat attemperation sprays are required.

Double-Suction First-Stage Impellers

Figures 12-1 and 12-3 show double-case pumps with double-suction first-stage impellers to reduce NPSH requirements. In some installations, this feature eliminates the need for a separate booster pump.

Mounting of the Impellers

Impellers are assembled on the shaft with a shrink fit to prevent mechanical looseness under all operating conditions and are positioned axially by split rings on the suction side of each impeller hub. This is the preferred mounting.

An alternative mounting employs a stack with sliding fits along the shaft. A spacer sleeve is fit snugly between each pair of impellers and an outside locknut is used to secure the impellers and sleeves. Great care is required to assure that all spacer sleeve and impeller faces are parallel to each other, perpendicular to the shaft, and smooth. Small errors in parallelism or perpendicularity will misalign the stack.

Impeller Wear Rings

Impeller wear rings are generally specified for refinery pumps [1], but must be secured with great care on high-speed, high-pressure pumps. Because boiler feedwater is a relatively clean liquid and rapid wear or seizure is unusual, impeller wear rings are seldom used in large, high-speed double-case boiler feed pumps. These boiler feed pump impellers are designed with extra stock on the wearing surfaces. When worn, impellers are skim cut true, and the pump then fitted with case wear rings that are undersized to match the impeller wearing surfaces.

Shaft Seals

Shaft seal failure is the most common cause of unscheduled outage for double-case pumps. Selection of a shaft seal system designed for the application is therefore critical to reliable pump operation.

Face-Type Mechanical Seals. Oil refinery pumps almost universally use mechanical seals. Reliable seals for high temperature oil and light hydrocarbons now exist. New federal air quality laws limiting hydrocarbon emissions encourage the use of tandem-type seal systems. Figure 12-7 shows a tandem seal assembly with bellows-type seals for hot oil service.

In the United States, only small to medium-size (up to 7,000 hp and 5,000 rpm) boiler feed pumps use mechanical seals. In Europe, they are also used in medium to large boiler feed pumps. A typical boiler feed pump seal is shown in Figure 12-8.

Most waterflood pumps have mechanical seals, operating at ambient temperatures, but in corrosive liquid.

Mechanical seals are described in detail in Chapter 17.

Throttle Bushings. The most reliable shaft sealing system for large boiler-feed pumps consists of throttle bushings with a custom designed, cold (90°–120°F) condensate injection system. Pumps have operated for 40 years or more with their original throttle bushings.

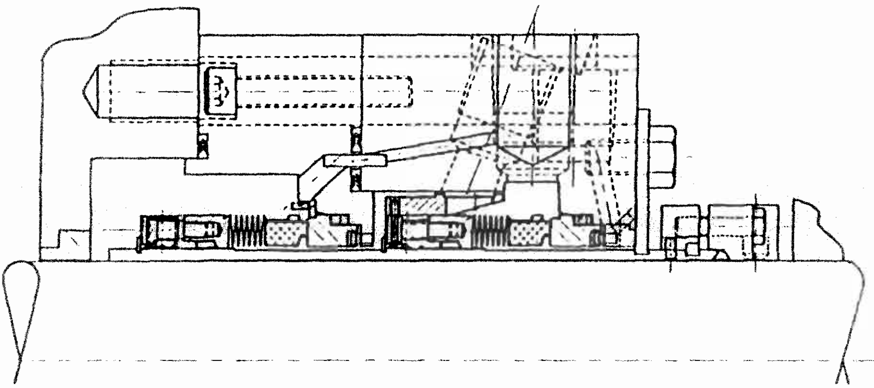

Figure 12-7. Tandem seal assembly for hot oil service (courtesy BW/IP International, Inc. Seal Division, manufacturer of BW Seals).

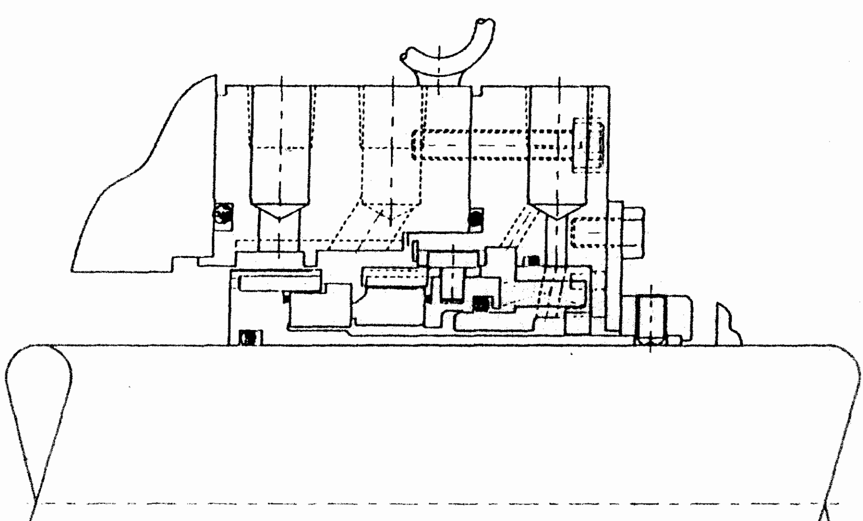

Figure 12-8. Shaft seal for boiler feed pump service (courtesy BW/IP International, Inc. Seal Division, manufacturer of BW Seals).

The throttle bushing bore, the shaft under the bushing, or both should be grooved. To obtain the desired effect, the following design parameters are varied: the groove cross section, the number of groove starts, the "hand" of the grooves, and the length of the grooved section. The grooves reduce leakage for a given running clearance and increase tolerance to solid particles in the feedwater. They also reduce the possibility of seizure if the pump is subjected to severe operating transients, such as flashing. Shaft sleeves under the throttle bushings are undesirable. They reduce the ability to resist seizure during severe temperature transients.

There are at least five types of condensate injection control systems [2]. The type of control is normally recommended by the pump manufacturer based on the purchaser's feedwater system design. A simple pressure-controlled system is shown in Figure 12-9. Temperature-controlled systems are more common. A drain-temperature control system is shown in Figure 12-10.

Two waterflood pumps such as the one shown in Figure 12-11 operate in series. The downstream pump has 4,000 psi suction pressure and 8,000 psi discharge pressure. Mechanical seals would not be practical for these pressures. Therefore the pumps are fitted with long throttle bushings that discharge into collection chambers with suitable drain connections. The cold leakage is expendable, and no re-injection system is needed.

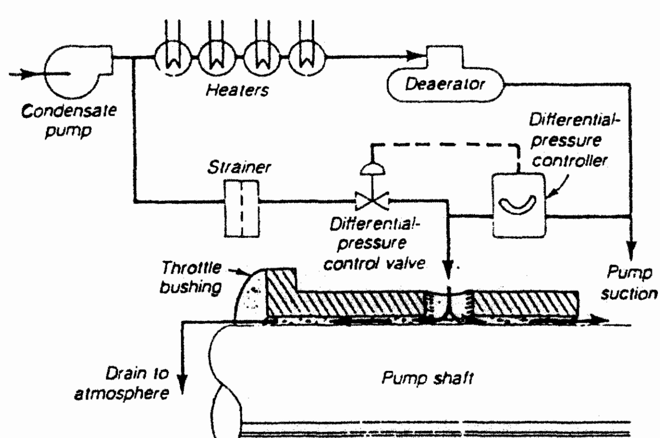

Figure 12-9. Pressure-controlled throttle bushing injection system (from Ashton [2]).

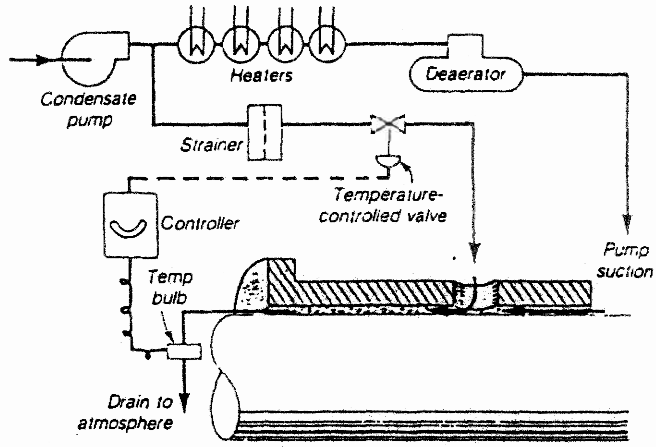

Figure 12-10. Drain-temperature controlled throttle bushing injection system (from Ashton [2]).

Floating-Ring Seals. Boiler feed pumps with floating-ring seals were common in the 1970s, but proved to fall short in reliability. A survey of boiler feed pumps in 1977 found 748 seal failures in 730 pumps with floating-ring or mechanical seals vs. 32 seal failures in more than 300 pumps with throttle bushings [5]. Specifications of most major architect/engineers in the United States no longer allow floating-ring type shaft seals [3].

Radial Bearings

Ball Bearings. Ball bearings can be used in smaller double-case pumps below 4,000 rpm, but generally are not favored.

Sleeve-type Bearings. Plain sleeve-type bearings are satisfactory for shaft diameters up to 3.50 inches in diameter at 3,600 rpm. They can be lubricated with oil rings.

Anti-oil-whip Bearings. Rotor instabilities, typically taking place at higher speeds, can be caused by lightly loaded hydrodynamic bearings. If oil whirl (half-speed whirl) is to be avoided, a bearing design that is more stable at lower loads is required (see Chapter 19). Pressure dam bearings have been used successfully for shaft diameters up to 6.75 inches at

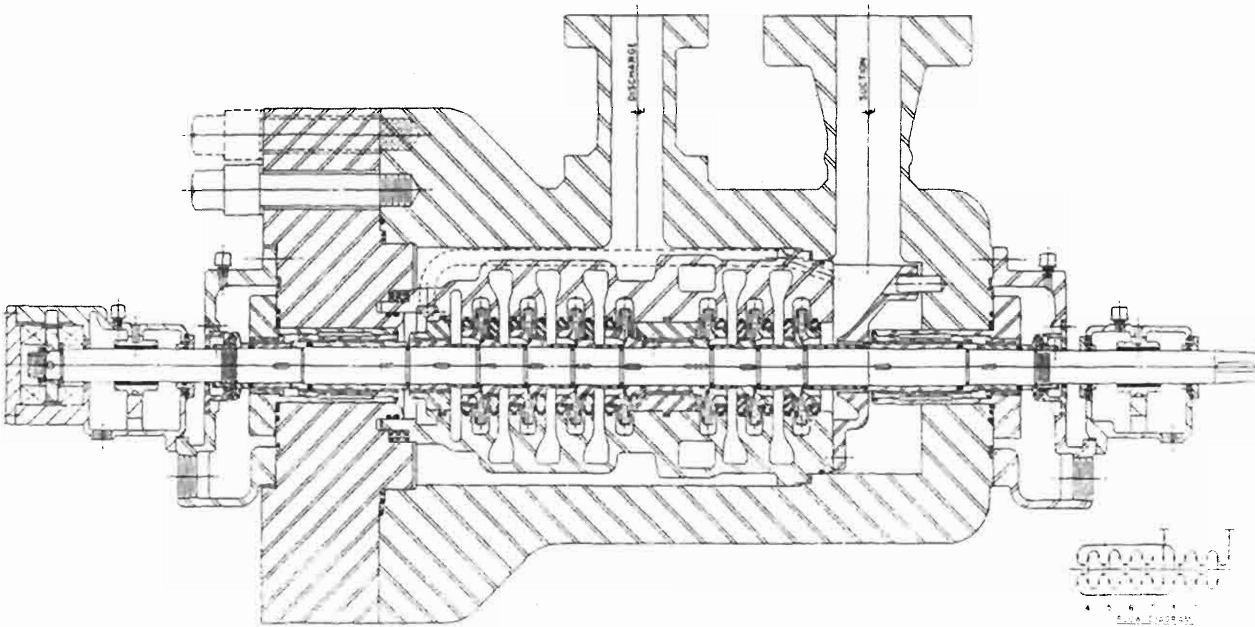

Figure 12-11. Double-case waterflood pump for 4,000 psi suction pressure (courtesy BW/IP International, Inc. Pump Division, manufacturer of Byron Jackson/United™ Pumps).

5,400 rpm. For larger shafts and higher speeds, tilting pad radial bearings are favored. These bearings require force-feed lubricating oil systems.

Thrust Bearings

Ring-oiled sleeve-type radial bearings generally use anti-friction thrust bearings. When force-feed lubricated radial bearings are used, thrust bearings should be of the tilting pad type.

Baseplates and Foundations

Baseplates and foundations must be designed so that misalignment between the pump, the driver, and other drive-train elements (such as gearboxes or variable-speed devices) is within allowable limits and so that they provide optimum dynamic support to minimize vibration. The traditional solution is to make the baseplate very stiff and to secure it rigidly to the foundation by bolting and grouting. A variation of this design technique is the use of sole plates under the pump and driver that are rigidly attached to the foundation and the use of very stiff concrete pedestals for the centerline mounted pump.

An alternative solution is to mount a fully rigid baseplate on springs or other flexible members to isolate the pumping unit from the foundation. This system has been used for many years in installations that include very large boiler feed pumps, and the experience gained is now being used to isolate pumping units from the potentially large motions of flexible offshore oil production platforms. Because weight is critical, honeycomb structures that are very light and stiff have been used for such baseplates. A sophisticated three-point mounting system can make the rotating equipment almost insensitive to large motions of the platform because no in-plane bending or torsion is generated by the deck motions.

In either case, large or high-speed pumping systems should be subjected to modal analysis and detailed finite element dynamic analysis to avoid mechanical and fluid (acoustic) resonances. Such analyses are especially important for variable-speed units. Some of the analysis methods used are described in Chapter 19.

Mounting of the Barrel

Double-case pumps for hot service are mounted at the pump centerline and the barrel is restrained from horizontal movement. This will assure that the horizontal and vertical position of the shaft axis is maintained during unit heatup and cooldown. The barrel is secured on the baseplate so that the thermal expansion is away from the coupling. This maintains the axial gap at the pump-to-driver coupling.

Design Features for Pumping Hot Oil with Abrasives

One of the most difficult double-case pump applications is the pumping of hot oil (500°F and above) with substantial quantities (2% or more) of entrained abrasive solids.

Surface Coating. The key to prolonged periods of operation without maintenance is the application of a hard surface coating, which may extend service life by a factor of 4 or more. The coating should have a minimum hardness of 60 Rockwell C. It should be applied to all wear surfaces, to all accessible hydraulic passages in impellers and inner cases, and to the outside of the impeller shrouds. The coating is typically applied with a high-velocity spray process that produces a strong mechanical bond. High coating density and proper coating thickness are critical.

Impeller and Case Wear Rings. Impellers designed with extra stock on the integral wear ring surfaces are generally preferable to replaceable impeller wear rings. Worn impeller wear ring surfaces can be re-coated and ground to size. They run against case wear rings that are coated in the bores and on the ends, and are sized to match the impeller running surfaces.

Keyways. Unless special design precautions are observed, rapid erosion occurs in keyways that are subjected to more than one stage of differential pressure. The two center stage impellers of opposed-impeller type pumps should be welded together at the hubs, and the key (or keys) terminated blind in a relief. A shrink-fit land is provided between the impeller bore and the shaft at the high-pressure end to seal against leakage and prevent erosion. Similar construction should be used for the sleeve under the throttle bushing of an opposed-impeller pump, or under the balancing drum of a pump with inline impellers.

Double-Case Pump Rotordynamic Analysis

The rotordynamic analysis requirements for a double-case pump depend on the size, rotational speed, and horsepower of the specific pump. Dry and wet critical speed analyses are adequate for small and medium-size pumps running at 5,500 rpm and below. This type of analysis is described in Chapter 19. Most current specifications only require critical speed analyses, and a full scale test with specified vibration limits.

Requirements of new supercritical power plants, new oil refinery processes, and high pressure oil field water injection facilities have increased the demand for predictably reliable, high speed, high horsepower double-case pumps. In addition to critical speed analyses, these pumps should be subjected to a rotor stability analysis as part of the design process. They also may be subjected to a rotor response analysis. At the present time, computer programs for rotor response analysis are available [7]. Response analysis includes consideration of excitation forces from both mechanical and hydraulic origins. Mechanical excitation forces from sources such as dynamic unbalance, misalignment, and shaft bow are well known. Hydraulic excitation forces are generated at the wear rings, long annular seals (such as balance drums or throttle bushings), and impellers. The magnitudes of hydraulic excitation forces (especially those generated by impellers) are less well known, but are believed to be much greater than mechanical excitation forces in large double-case pumps (which are precision manufactured to minimize mechanical forces). Some cutting edge research on hydraulic excitation forces has been conducted and is ongoing [7].

The Effect of Stage Arrangement on Rotordynamics

The opposed-impeller stage arrangement of volute-type pumps offers greater rotordynamic stability than the inline arrangement with all impellers facing in the same direction, which is common to diffuser-type pumps. This has been shown for a number of years [4] by critical speed analyses. A recent comparison, based on the stability analysis of an 8,000 rpm pump [6], is given in Table 12-1. Here an analysis of a pump with opposed-impellers is compared with the analysis of an "equivalent" inline impeller arrangement. Identical impeller forces and annular seal coefficients were used. With design clearances and smooth (not grooved) annular seals, the analyses showed stable operation and no subsynchronous whirling up to 14,000 rpm for opposed impellers, but only up to 8,000 rpm for inline impellers. The difference is attributed to the extra center bushing in the opposed-impeller design, where a strongly stabilizing Lomakin effect is generated. This advantage is reduced as the internal annular seals wear and clearances increase.

The Effect of Impeller Growth from Centrifugal Forces

Radial growth of high-speed pump impellers caused by centrifugal forces is significant [6]. The unsymmetrical impeller deformation caused by centrifugal forces, pressure loading, and shrink fit to the shaft for a four-stage, 8,000-rpm boiler feed pump is shown in Figure 12-12.

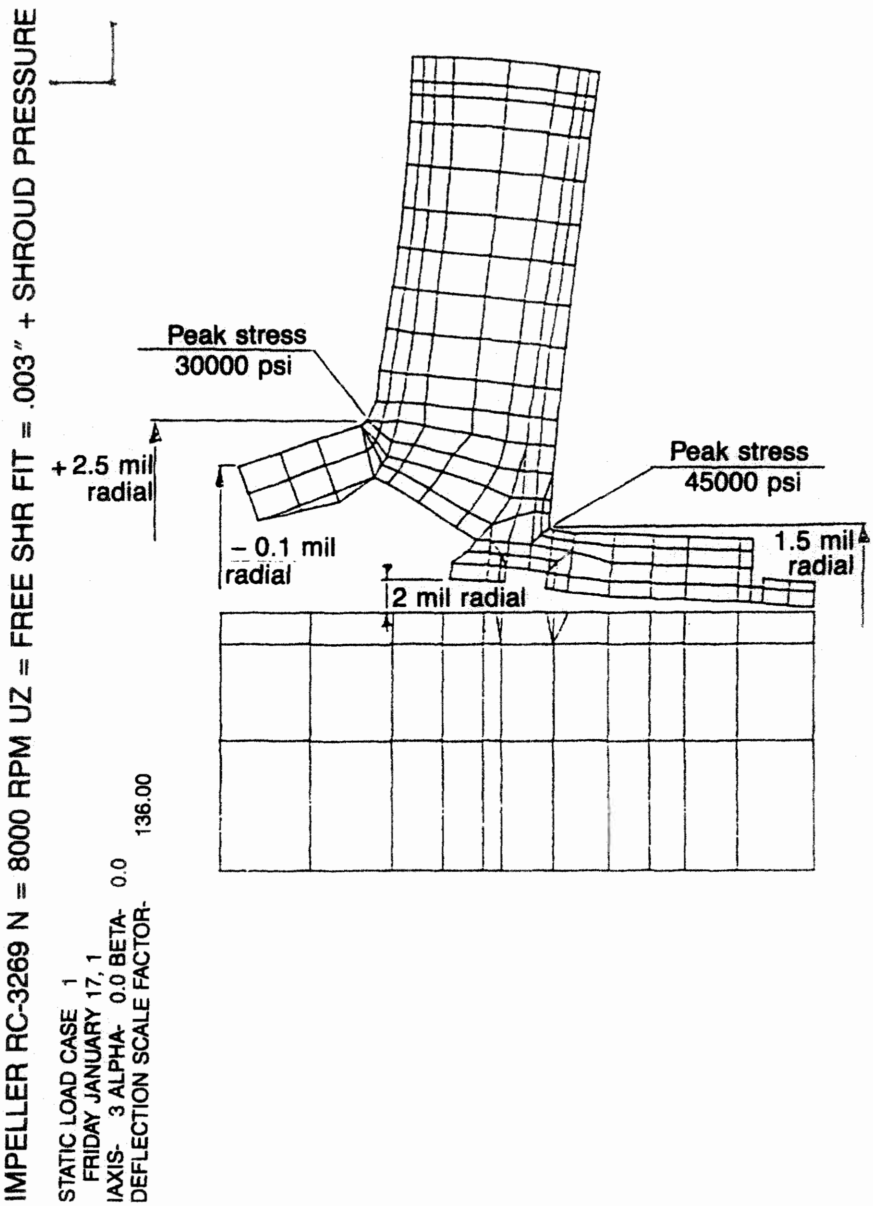

Figure 12-12. High-speed impeller deformation caused by centrifugal forces, pressure loading, and shrink fit (from Verhoeven [6]).

Reduced Annular Seal Clearance. Of particular interest is the growth of the impeller eye seal diameter. As pump speed increases, the decrease in annular seal clearance reduces leakage loss as a percentage of input power and improves pump efficiency. Mechanical friction losses and impeller disk friction losses also are reduced, but the reduced leakage loss is dominant. These effects are most noticeable in pumps of low specific speed. Factory testing of a 1,010 specific speed pump at 2,950 rpm and 7,000 rpm demonstrated an increase of six points of efficiency (from 49% to 55%) at the higher speed. Low speed pumps are less affected. The difference in efficiency between 1,800 rpm and 3,600 rpm operation for most pumps is negligible.

Comparison of Diffuser Casings with Volute Casings

Both diffuser-casing construction with inline impellers and stacked inner case assembly, and volute-casing construction with opposed impellers and horizontally split inner case assembly can be applied with success to critical centrifugal pump applications. There are, however, significant differences. A comparison of these differences follows.

Diffuser Casings

Lower Cost. The diffuser-casing construction with inline impellers results in lower manufacturing cost. Additionally, if damage occurs, a single casing segment can be replaced instead of a complete volute case. All series impellers and series casings can be of the same design and any number of stages can be stacked to produce the required head. Opposed-impeller construction requires right-hand and left-hand impellers and right-hand and left-hand volutes. Major pattern changes or separate patterns are required to produce inner casings with the desired number of stages needed to produce the full range of head requirements.

Precision Diffusers. The critical portions of the diffuser passages (those in which high fluid velocity is converted to pressure) can be investment-cast or machined, thus manufactured with less roughness and more precision than the typically sand-cast inner volute cases.

More Compact. The inline impeller configuration is more compact. All else being equal, the result is a shorter rotor and a shorter pump, and because the inline impeller design has no crossover, the inner casing and barrel diameters also become smaller than those of the opposed impeller construction.

Simple to Destage. If the pump differential head requirement is reduced, it is very easy to remove one or more stages from a pump with inline impellers. Pumps with opposed impellers can be destaged, but the process is more complex.

Volute Casings

Rotordynamic Stability. As illustrated in Table 12-1, the opposed-impeller configuration has better rotordynamic stability. Swirl brakes are not required except to solve very unusual application problems.

Dynamic Balance. Because the volute casings are horizontally split, the fully assembled rotating element is dynamically balanced in its final

Table 12-1
Subsynchronous Whirl and Stability Threshold Speeds of Inline and Opposed Impeller Arrangements for Different Conditions

Condition	Opposed Impeller Arrangement		Equivalent Inline Impeller Arrangement	
	Grooved Annular Seals	Smooth Annular Seals	Grooved Annular Seals	Smooth Annular Seals
100% design clearance	SSW 8,000 rpm, Unstable 9,000 rpm	SSW 14,000 rpm, Unstable 15,000 rpm	SSW 5,500 rpm, Unstable 6,500 rpm	SSW 8,000 rpm, Unstable 10,000 rpm
200% design clearance		SSW 9,250 rpm, Unstable 10,000 rpm		SSW 6,600 rpm, Unstable 7,400 rpm
300% design clearance		SSW 7,700 rpm, Unstable 8,500 rpm		SSW 5,500 rpm, Unstable 6,300 rpm
400% design clearance		SSW 5,150 rpm, Unstable 5,900 rpm		SSW 5,000 rpm, Unstable 5,600 rpm
100% design clearance with swirl brakes	SSW 12,000 rpm Unstable 13,500 rpm		SSW 9,000 rpm Unstable 10,500 rpm	

SSW = Subsynchronous Whirling

form. This is not possible with the diffuser-casing construction. The latter requires alternate assembly of impellers and diffuser casings and therefore dismantling and reassembly of the impellers on the shaft after dynamic balance. Exact restoration of dynamic balance after the rotor has been dismantled cannot be assured.

Sag Bore. The double-volute design lends itself to machining of the equivalent natural deflection of the rotating element into the bottom volute case half. The result is that all running clearances remain concentric because the shaft will operate in its deflected position. This is especially important when operating on turning gear in hot-standby condition. This refinement, known as sag bore, cannot readily be effected with the diffuser design with its multiplicity of concentric fits.

Running Clearance Check with Feeler Gauge. By placing the rotating element, including all rotating and stationary wear parts, into the bottom volute case half, all running clearances can be checked with a feeler gauge to verify that a reconditioned rotor has proper clearances, or to determine if wear has taken place in a used rotor.

Easier Rotor Replacement. A rotating element can be quickly removed from its volute case and a spare one installed. Stocking of a spare inner volute is optional because it is not considered a wearing part. Many users stock a spare rotating element only. In order to expedite disassembly and assembly time with a diffuser design, the user must purchase a complete inner case assembly because field assembly of the rotor and diffuser cases is a time-consuming procedure.

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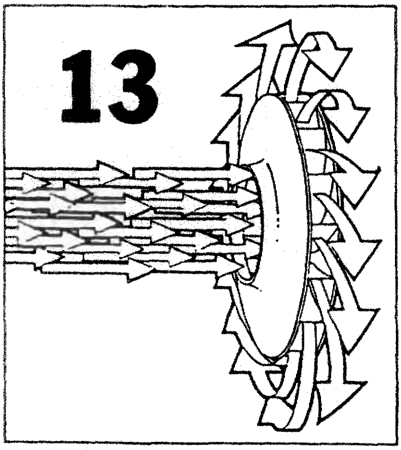


Slurry Pumps

by **George Wilson**
Goulds Pumps, Inc.

There are various types of centrifugal slurry pumps, which are identified by their capability to handle solids ranging in size, hardness, concentration, and velocity. An understanding of these important factors will lead to an optimum choice of pump design where the materials of construction and rotational speed are ideally matched to the process system.

Slurry Abrasivity

The abrasiveness of slurries is difficult to define due to the number of variables involved. It is dependent on the nature of the slurry being pumped and the materials of construction of the pump liquid end components.

Wear increases with increasing particle size. For example, the rate of erosion wear when pumping a silica sand slurry is approximately proportional to the average particle size raised to the power of 1.4.

Wear increases with concentration; the relationship is linear up to about 10% by volume. At higher concentrations, the rate of wear will level out due to the cushioning effect of the particles as they collide.

Wear increases rapidly when the particle hardness exceeds that of the metal surface being abraded. The effective wear resistance of a metal will depend on the relative hardness of the metal to that of the particle.

An approximate comparison of hardness values of common ores and minerals is given in Figure 13-1.

Wherever possible, the hardness of the pump liquid end metal components should exceed the particle hardness. It should be noted that the measurement of hardness is not the only criteria and that the structure of the metal material itself has to be considered. An example of this is the

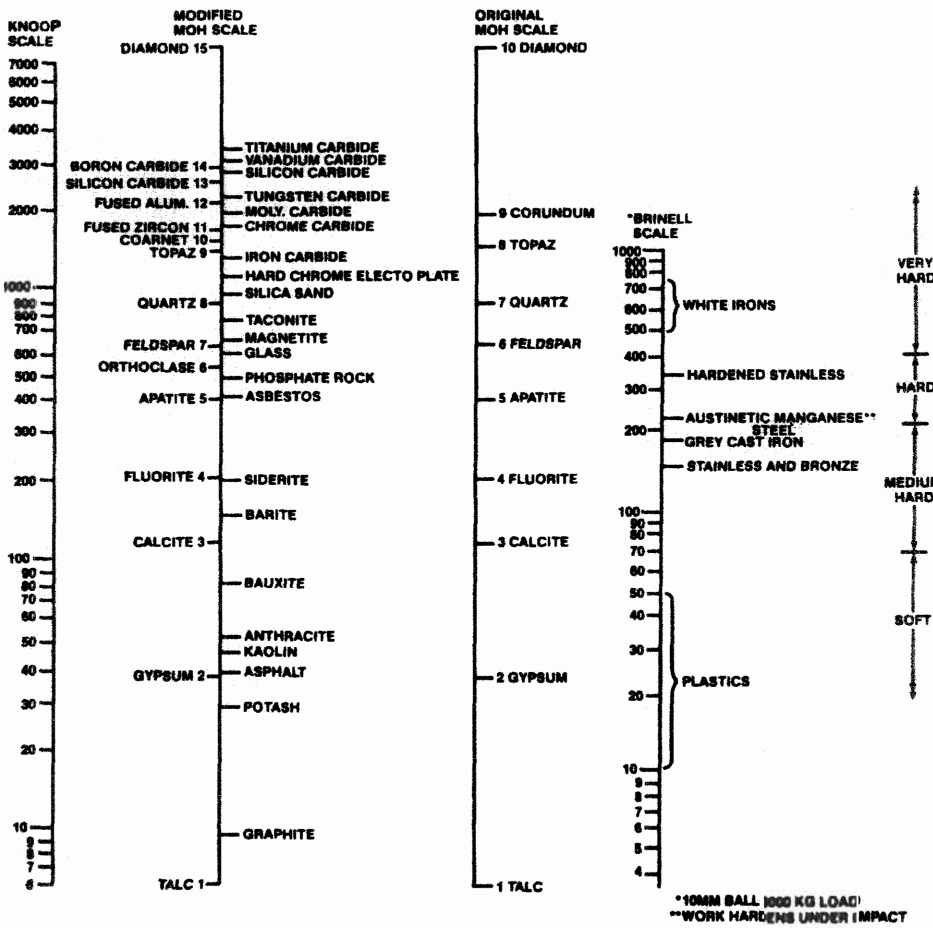


Figure 13-1. Approximate comparison of hardness values of common ores and minerals.

large degree of very hard chromium carbide (1800 Knoop hardness) precipitation that can be achieved in high chrome iron. Wear increases:

- When the particles are angular.
- With particle density.
- With increasing particle velocity such that the rate of wear is directly proportional to V^m where m can vary from 2.5 to 4.

Parts life can be significantly extended if the system head requirements are reduced and a lower-rotational-speed pump is selected.

Where pumps are applied to a slurry that is both corrosive and abrasive the predominant factor causing wear should be identified and the materials of construction selected accordingly.

To make the correct pump selection the following factors must be specified:

- *Particle size distribution*—From which can be determined the average particle size.
- *Particle shape*—State whatever particles are angular or smooth.
- *Solids concentration*—For convenience a nomograph relationship of concentration to specific gravity of aqueous slurries is given in Figure 13-2.
- *Particle hardness*—Given in terms of Mohs or Knoop scale.
- *Particle specific gravity*—The mixture specific gravity can be determined from Figure 13-2 if the concentration is known. Note the pump BHP is directly proportional to the mixture specific gravity.
- *Conveying liquid*—State viscosity, temperature, and corrosiveness.
- *System requirements*—Total head and capacity. It may be necessary to correct the pump performance for the effects of the solids in the liquid.

Pump Materials to Resist Abrasive Wear

Tough materials are used to resist gouging abrasion (caused by the impingment of large dense particles). Toughness is the amount of plastic deformation a material can withstand without fracture. Generally the larger the difference is between the yield and tensile strength, the tougher the material will be.

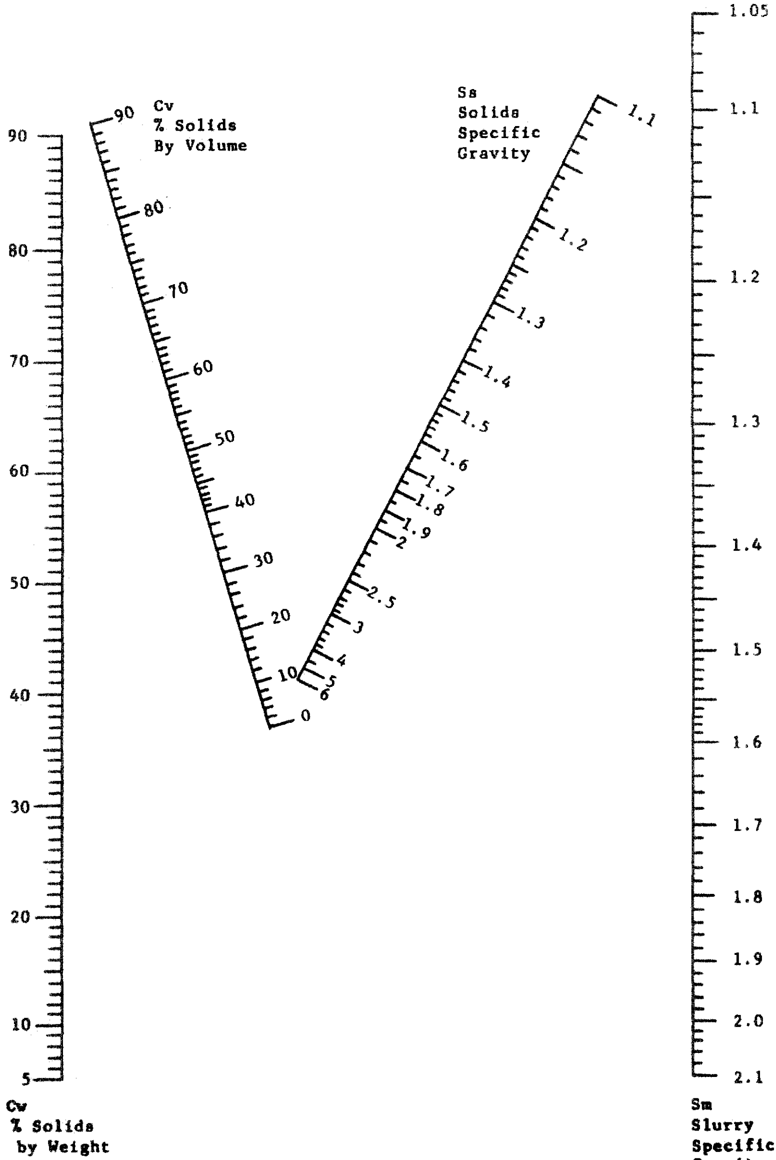


Figure 13-2. Nomograph of the relationship of concentration to the specific gravity in aqueous slurries.

Austenitic manganese steel, which work-hardens under impact, is used in very slow-speed, dredge-type pumps when handling large dense solids.

Hard metal materials are used to resist erosion abrasion (caused by the combined effects of cutting wear parallel to the surface and to a lesser extent deformation wear). Depending on the nature of the slurry, a wide selection of materials are available. (Note: Values listed in Table 13-1 are average.) A tabulation of alloys for abrasion resistance is given in Table 13-2.

It is the ability of elastomeric materials to deform elastically under impact that makes them ideally suited to resist erosion wear. Natural rubber will far outlast metal provided it is compatible with the liquid being

Table 13-1

Material Name	ASTM No. Casting	Average Brinnell Hardness	Characteristics and Typical Applications
Ni-Hard 1	A-532 Class 1	550	2.5% chrome, 4% nickel, 3.3% carbon. Good resistance to cutting-type erosion, it is not recommended for acids but can be used for mildly alkaline slurries.
Ni-Hard 4	A-532 Class 1	575	8% chrome, 6% nickel, 3.3% carbon. Compared to Ni-Hard 1, it has a higher tensile strength and is more resistant to both corrosion and erosion.
High chrome alloy	A-532 Class B	650	26% chrome, 2.8% carbon. Superior erosion resistance and better corrosion resistance down to 5 pH. This material can be machined by conventional means in its annealed state then hardened and tempered with minimum distortion.
*PACE™		400	27% chrome, 2% nickel, 2% moly, 1.6% carbon. The most suitable alloy for erosive, corrosive slurries in the range 1 to 11 pH. However its erosion resistance is inferior to high chrome iron.

** Registered trademark of Abex Corporation*

Table 13-2
Alloys for Abrasion Resistance
(Properties Sensitive to Carbon Content Structure)

Alloy	Properties
Tungsten carbide composites	Maximum abrasion resistance. Worn surfaces become rough.
High-chromium irons	Excellent erosion resistance. Oxidation resistance.
Martensitic iron	Excellent abrasion resistance. High compressive strength.
Cobalt base alloys	Oxidation resistance. Corrosion resistance. Hot strength and creep resistance.
Nickel base alloys	Corrosion resistance. May have oxidation and creep resistance.
Martensitic steels	Good combination of abrasion and impact resistance.
Pearlistic steels	Inexpensive. Fair abrasion and impact resistance.
Austenitic steels	Work hardening.
Stainless steels	Corrosion resistance.
Manganese steel	Maximum toughness with fair abrasion resistance. Good metal-to-metal wear resistance under impact.

Increasing Abrasion Resistance

Increasing Toughness

pumped and the particle size is limited to fines below 7 mesh in size. At velocities above 35 ft/sec the rubber may not have sufficient time to flex and absorbs all the impact, and as result, wear will increase. Natural rubber is limited in temperature to 150°F or less.

Where oils are present, a synthetic rubber such as neoprene should be used; however the addition of fillers will have a detrimental effect on wear resistance.

Elastomer materials generally have good corrosion resistance, but care must be exercised to prevent the slurries from penetrating behind the casing and causing corrosive damage.

Natural rubber-lined pumps with a durometer hardness of 40 shore A are usually limited to about 120 feet total head. Higher heads can be generated if fillers are added to increase hardness.

Castable urethanes in the 90 shore A hardness range exhibit good tear strength and elongation properties and in certain applications have out performed both rubber and metal.

Ceramic materials in castable form have excellent resistance to cutting erosion but because of their brittle nature are unsuitable for direct impact. Silicone carbide refrax liners and impellers in the 9.5 original Mohs hardness range are commonly used for pumping fines where the impeller tip velocity is limited to less than 100 ft/sec.

Slurry Pump Types

There is no specific demarcation point where one pump design ceases to be effective and another takes over. Figure 13-3 shows a classification of pumps and materials according to particle size. It is important to note that the selection of the pump type and its materials of construction depend also on the abrasivity of the slurry and the total head to be generated.

The abrasivity of slurries can be divided into five distinct classifications to which limits on pump selection can be applied. Table 13-3 shows a pump selection guide for wear resistance.

Specific Speed and Wear

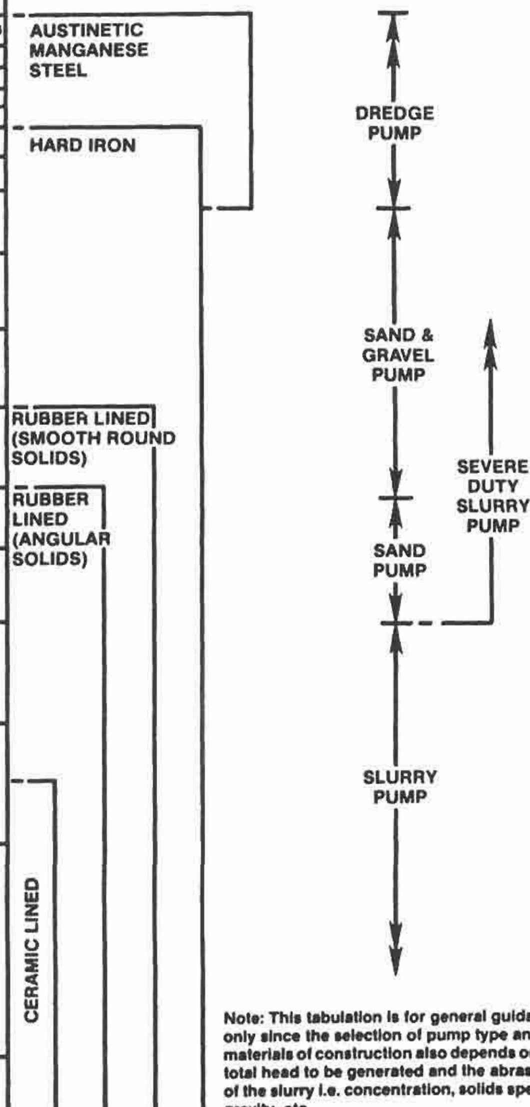
The majority of centrifugal pumps are conventionally designed to achieve the desired hydraulic performance at the highest efficiency and lowest cost when handling clear fluids in reasonably clean environments. Manufacturing limitations are not imposed on the configuration of the pump, since conventional materials such as cast iron, bronze, and stainless steel are used. Since wear is not a major consideration, the highest possible specific speed is chosen.

When a centrifugal pump is designed for a slurry service, the factors that predominantly influence the pump design are wear and materials of construction; efficiency is of lesser importance. To achieve these objectives the pump has to operate at a lower rotational speed and the impeller is typically a radial-flow type. This suggests that the pump must be of a low specific speed design in the range 600 to 1,800. Specific speed is defined in Chapter 2.

Since wear is a function of velocity it can be shown that for a given head and capacity, wear will increase with increased N_s.

Tyler Standard Sieve Series: Aperature Inch	Tyler Standard Sieve Series: Aperature MM	Mesh	Grade
160	4060	—	Very large bolders
80	2030	—	Large bolders
40	1016	—	Medium bolders
20	508	—	Small bolders
10	254	—	Large cobbles
3	76.2	—	Small cobbles
2	50.8	—	Very coarse gravel
1.5	38.1	—	
1.050	26.67	—	Coarse Gravel
.883	22.43	—	
.742	18.85	—	
.624	15.85	—	Medium Gravel
.525	13.33	—	
.441	11.20	—	
.371	9.423	—	
.321	7.925	2.5	Fine Gravel
.263	6.680	3	
.221	5.613	3.5	
.185	4.699	4	
.156	3.962	5	Very Fine Gravel
.131	3.327	6	
.110	2.794	7	
.093	2.362	8	
.078	1.981	9	Very Coarse Sand
.065	1.651	10	
.055	1.397	12	
.046	1.168	14	Coarse Sand
.039	.991	16	
.0328	.833	20	
.0276	.701	24	
.0232	.589	28	Medium Sand
.0195	.495	32	
.0164	.417	35	
.0138	.351	42	
.0116	.295	48	
.0097	.248	60	Fine Sand
.0082	.204	65	
.0069	.175	80	
.0058	.147	100	
.0049	.124	115	
.0041	.104	150	
.0035	.089	170	Silt Slimes
.0029	.074	200	
.0024	.061	250	
.0021	.053	270	
.0017	.043	325	Pulverized
.0015	.038	400	
—	.025	*500	
—	.020	*625	
—	.010	*1250	
—	.005	*2500	
—	.001	*12500	
—	.0005		Mud Clay
—	.0024		

*Theoretical values Micron = .001 mm



Note: This tabulation is for general guidance only since the selection of pump type and materials of construction also depends on the total head to be generated and the abrasivity of the slurry i.e. concentration, solids specific gravity, etc.

Figure 13-3. Classification of pumps according to solid size.

Table 13-3
Pump Selection Guide for Wear Resistance

Abrasion Class	Nature of Slurry	Selection
Mildly abrasive	Concentrations of relatively soft solids or very low concentrations (measured as ppm) of hard silt-sized particles.	Cast iron construction usually satisfactory, but hard-faced impeller rings and special attention to stuffing box area is justified. Consider stainless steel impeller. No limits on pump speed.
Abrasive	Low concentrations of hard fines or high concentrations of soft material.	Slurry pump design required with Ni-hard, chrome iron, or rubber construction. Open impellers are acceptable. Although no limits are placed on pump speed, discretion is advised.
Severely abrasive	High concentrations of hard fines or lower concentrations of coarse material.	Slurry pump design required with chrome iron construction. Restrictions are placed on allowable pump speed and total head.
Primary circuit	Maximum concentrations of fines or coarse material up to 10 mm usually.	Severe-duty slurry pump design required with chrome iron construction. Large restrictions placed on allowable pump speed and total head. Parts life is measured in months.
Dredge	Large concentrations of boulder-sized solids.	Dredge type design required with manganese steel construction to resist impact. Very low rotational speed required.

Areas of Wear

Casing

The rate of wear and the hydraulic forces within the pump will be reduced if concentric-type casing volutes are adopted over conventional spiral volutes. At "off" design point operation, the static pressure around the impeller's outside diameter will be relatively uniform, and turbulence in the vicinity of the cutwater will be effectively reduced as will the slurry velocity entering the casing throat. Recirculation flows from the

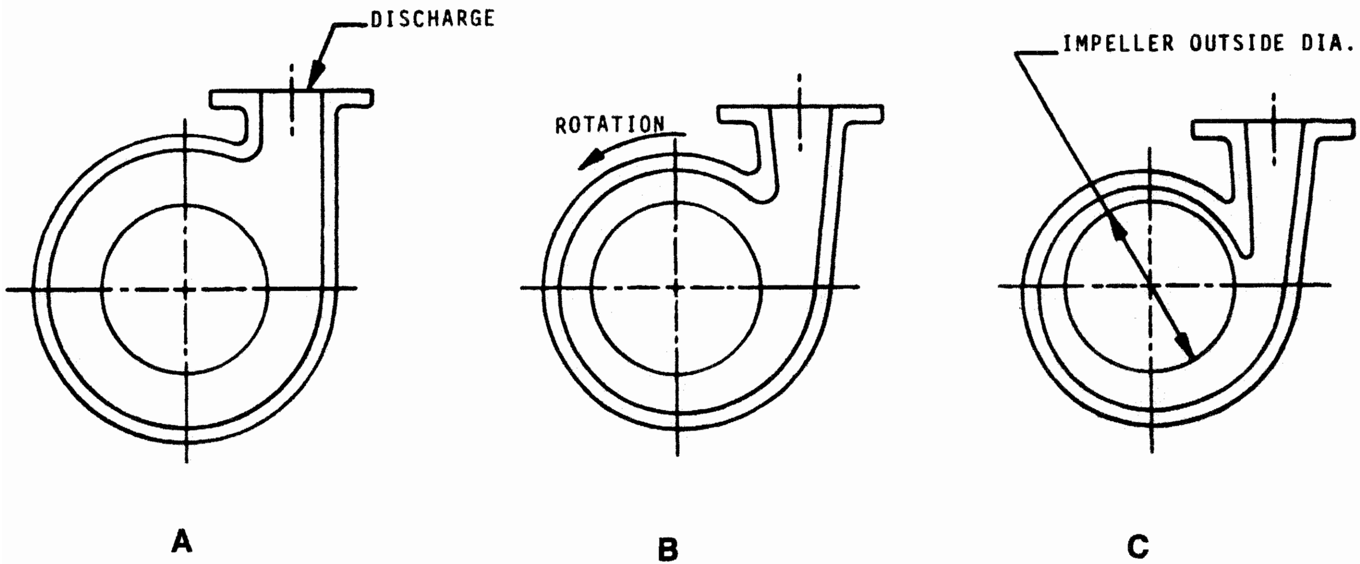


Figure 13-4. Casing configurations: (A) concentric volute; (B) semi-concentric volute; (C) spiral volute.

volute back to the suction will also be more uniform resulting in a reduction in localized wear in the vicinity of the casing near the impeller eye.

The degree of casing concentricity must be reconciled with the pump specific speed and efficiency. For example, a 12-in. slurry pump (N_s = 1,350) with a conventional volute could have a 82% peak efficiency, whereas the same pump with a semi-concentric casing will have an efficiency of 80.5%. Concentric casing designs produce flat efficiency curves that are sustained at a high level over a wide range of flows making it more amenable to off-design point operation.

Up to 1,200 N_s fully concentric casings could be adopted without too much sacrifice in efficiency (Figure 13-4a).

From 1,200 to 1,800 N_s the casings should be semi-concentric, progressing towards a spiral-volute configuration at the high end of specific speed. A compromise is therefore reached between efficiency and rate of wear (Figure 13-4b).

Above 1,800 N_s the pump should only be applied to mildly abrasive services and a spiral-volute casing will be utilized in the interests of higher peak efficiency (Figure 13-4c).

Impeller

Open impellers are used where the abrasion is not too severe as they have good air handling capabilities and are cheaper to produce. However, performance deteriorates when the front clearance opens up due to wear.

Closed impellers are preferred over open impellers for severe abrasive slurries since those impellers are more robust and will last longer. Also closed impellers are not nearly so sensitive to fall off in performance when the front clearance increases.

The requirement for extra thick impeller vanes can cause restrictions at the impeller eye and inlets. Three to five vanes are normal, depending on the pump specific speed and solids handling capability.

Pump out vanes are normally provided on the rear shroud. These vanes have the effect of minimizing the pressure at the pump stuffing box and reducing the axial hydraulic unbalance. The power absorbed by these vanes is not all wasted since it helps to generate head. A small drop in efficiency can be expected.

Wear Plates

Suction-side wear plates should always be provided on metal slurry pumps, and if the service is severe, the plate should extend into the suction nozzle. The suction-side wear plate is usually the part which needs replacement most often.

There is little to be gained by fitting a rear wear plate, provided the rear of the hard metal casing extends to the stuffing box. Experience has shown that the rate of wear in this area is not any greater than in the casing itself.

Bearing Frames

Usually the bearings are oil lubricated with a calculated life of over 50,000 hours. Slurry pumps are installed in dirty dust-laden atmospheres, and extra precautions have to be taken to seal the bearing covers and prevent the ingress of liquid and dust. In severe services, taconite seals are provided (i.e. double-lip type seals with grease cavities).

Sealing

Slurry pumps are often subjected to severe shock loading and shaft whip due to the presence of solids and system upsets. For these reasons soft compression packing is still favored as a means of sealing at the stuffing box.

The preferred method for packing a slurry pump is the "flush" seal shown in Figure 13-5a. Here the lantern ring is positioned in front of the packing rings and a copious supply of clean liquid is injected at a pressure higher than the prevailing slurry pressure in the stuffing box. The clean liquid acts as a barrier and prevents the ingress of abrasive particles that cause packing and sleeve wear. The disadvantage of this system is that large amounts of flushing water are required and the pumped product will be diluted. This system is recommended for severe abrasive services.

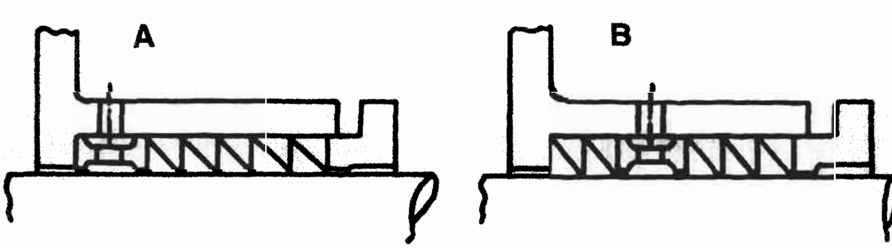


Figure 13-5. (A) Typical "flush-type" slurry pump stuffing box. Barrier flush prevents abrasive wear. (B) Typical "weep-type" stuffing box. It uses considerably less gland water but is much more susceptible to abrasive wear.

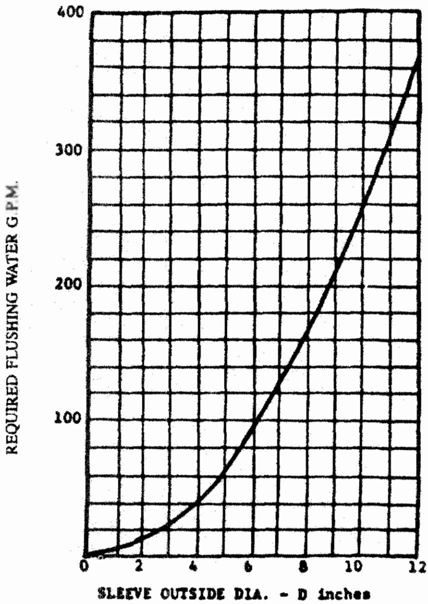


Figure 13-6. Approximation of flushing water requirements for "flush-type" slurry pump stuffing boxes where flush pressure is 15 psi above slurry pressure at the box. "Weep-type" stuffing boxes use about 5% of the water that "flush-types" use.

An alternative method for sealing is shown on Figure 13-5b. Here the lantern ring is positioned between packing rings. This configuration is called a "weep" seal. Again, clean liquid should be injected at a pressure higher than the prevailing slurry pressure near the stuffing box. Product dilution is significantly reduced compared to the "flush" seal design. However, the barrier so created is not very effective, causing abrasive particles to penetrate and cause wear. If the service is only mildly abrasive, then grease can be used in lieu of liquid.

An approximation of flushing requirements for a "flush" type packing arrangement for conventional throat restriction devices where no attempt has been made to curtail the use of flushing water and where the pressure differential is 15 psi is displayed in Figure 13-6. Such a restriction will have an annular radial clearance in the order of .007 times the sleeve diameter. The length of the throat bush will be about the same as the width of one turn of packing.

It is impossible to predict the exact amount of flushing water required when the packing is "weep" type, since this is dependent on shaft deflection and gland maintenance. However, under normal operating conditions, weepage would be in the order of 5% of the values stated in Figure 13-6 for "flush" packing arrangement.

In most cases, seals and flush requirements are provided in ignorance

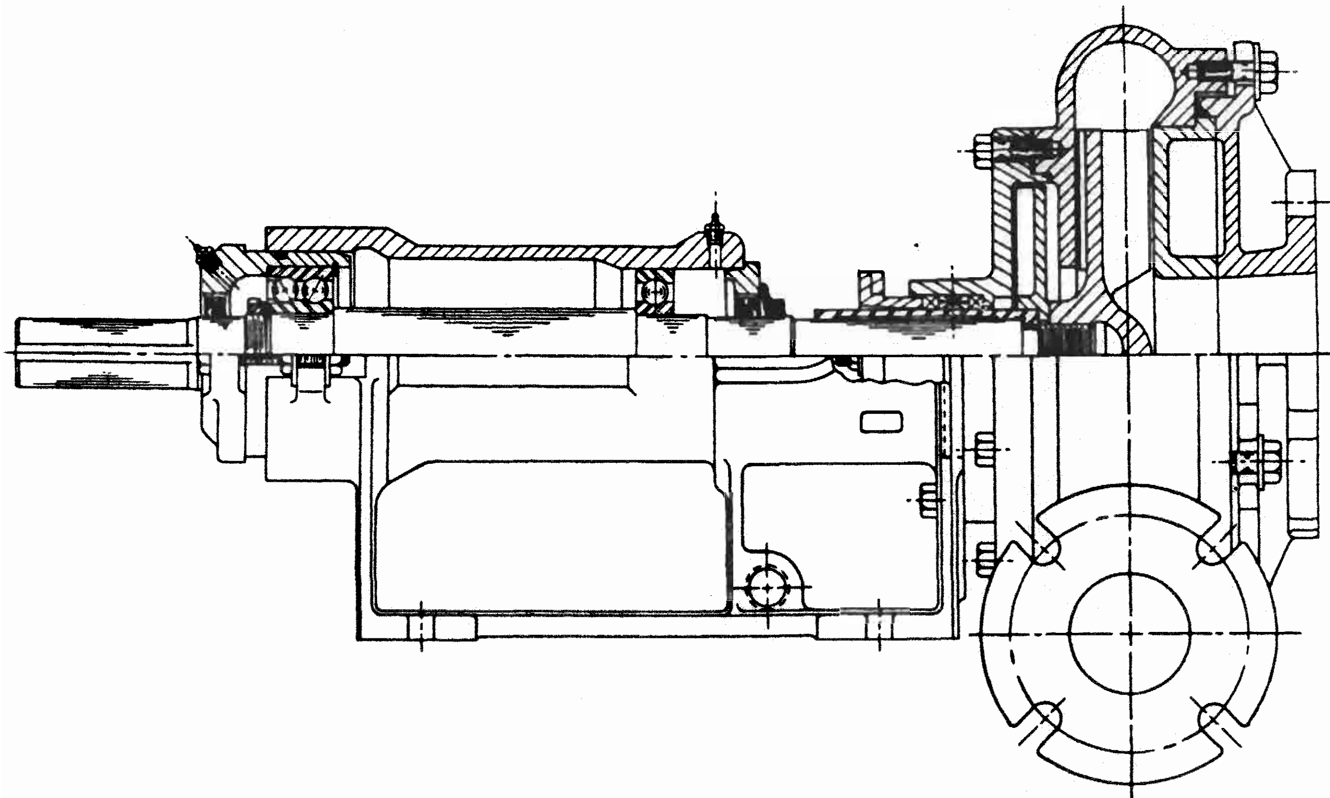


Figure 13-7. End-suction pump fitted with expeller (courtesy Goulds Pumps, Inc.).

of the real pressure prevailing at the stuffing box, which results in excessive use of gland water and increased maintenance.

Built into some slurry pump designs are methods to reduce pumped pressure at the stuffing box by hydrodynamic means. (For example, see Figures 13-7 and 13-8 for diagrams of pump out vanes on impellers and expellers.) The side-suction-pump configuration is subjected only to suction pressure and has an advantage over end-suction pumps, one not fully recognized by users. By proper application of impeller pump out vanes and expellers, the pressure at the box can be reduced to almost zero. This is called a dry box arrangement. In these cases, weep-type seal is satisfactory, with either water or grease being injected into the cavity formed by the lantern ring.

Sump Design

Many slurry pump problems will be eliminated if proper attention is given to the sump design. Design considerations are:

- The suction feed box should be placed as close to the pump as possible.
- The slurry level in the feed box above the pump center line should be at least seven times the pump suction nozzle size.
- The feed box should always have a hopper bottom sloping to the pump suction as shown. See Figure 13-9.
- The suction pipe should always have a minimum slope of at least 30°; this particularly important when handling settling-type slurries.
- The feed box should be sized so there is a minimum retention volume of slurry equal to or greater than two minutes of pump flow. If the slurry is frothy, then a greater retention time is required (e.g. eight minutes of pump flow).
- A dump gate should be provided at the bottom of the feed box.
- Turbulence near the feed box walls should be avoided to prevent excessive wear.

Pump Drive

Generally slurry pumps are belt driven because it is almost impossible to match the pump to the system by trimming the diameter of rubber and hard metal impellers, due to their design and materials of construction.

Traditionally slurry pumps are driven by V-belts so that pump performance can be adjusted to meet actual conditions of service in the field, thereby saving power and reducing wear. As wear increases, pump output is reduced. This can be easily and inexpensively rectified by increas-

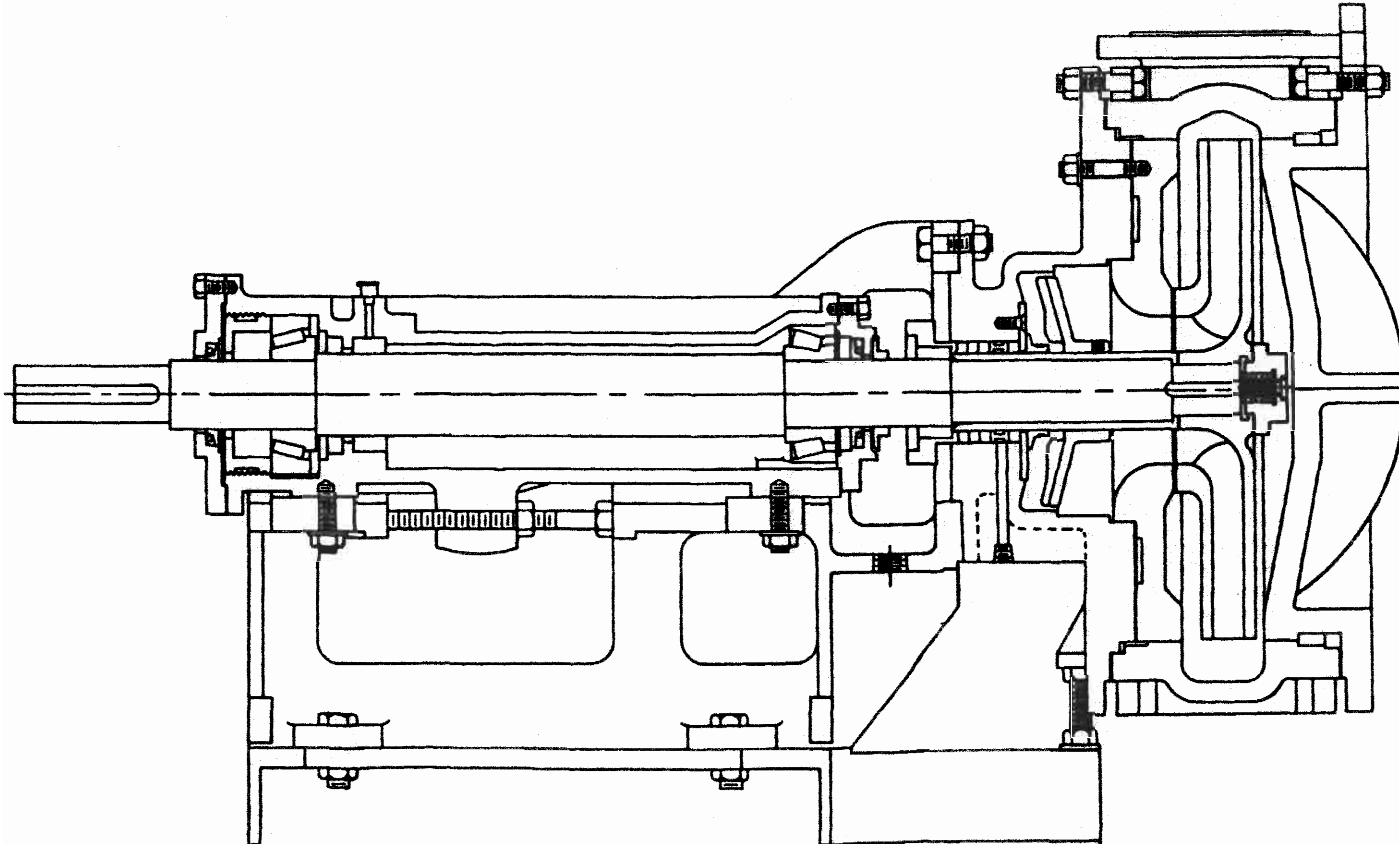


Figure 13-8. Side-suction pump fitted with expeller (courtesy Goulds Pumps, Inc.).

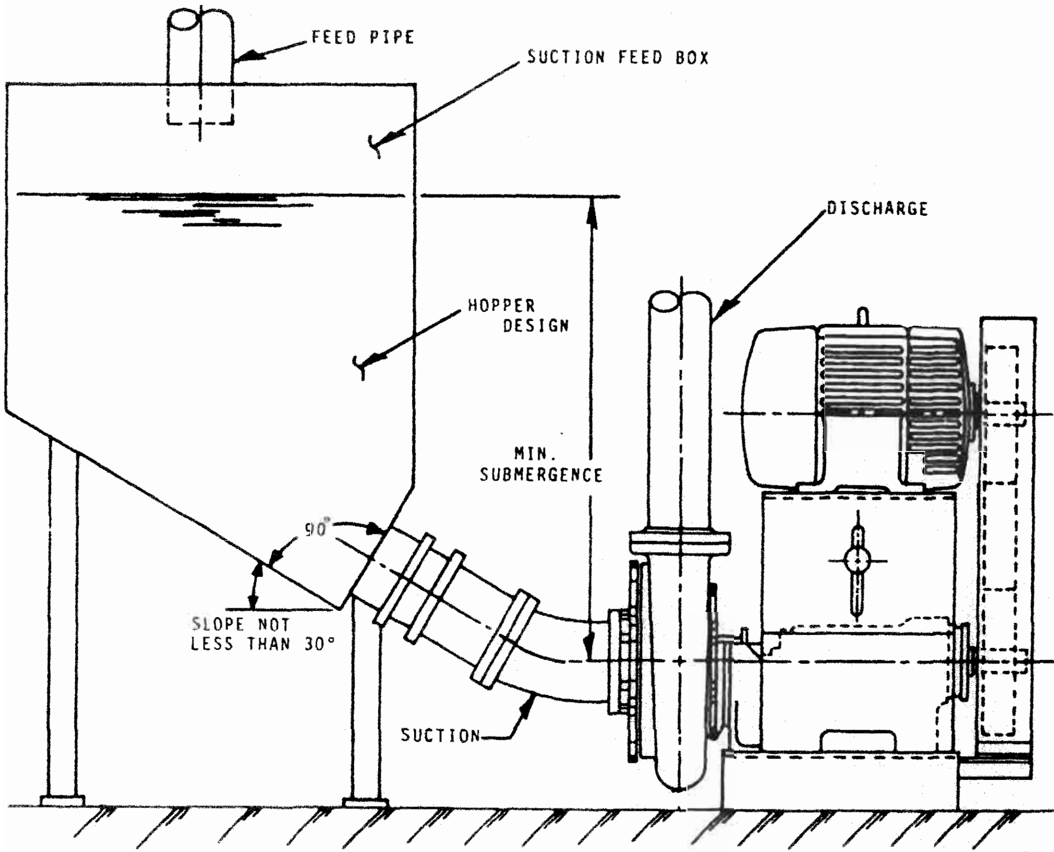


Figure 13-9. Typical suction feed box.

ing the pump speed by changing the sheave ratios. Three to five percent should be added to the motor BHP to compensate for belt losses. It is always good practice to add one more belt than is normally calculated to cover upset conditions and belt breakage.

Motors must be rated with an adequate margin to cover upset conditions such as high flow due to lower-than-expected system losses, higher concentrations, and start-up. At start-up, the concentration is often higher, and if the pump was not flushed out during the previous shutdown, the pump could be plugged with solids requiring high breakaway torques to get the impeller rotating. This undesirable condition happens all too frequently and can cause pump damage, excessive wear, and motor overload.

Under well-controlled systems, free from upset, the motor could be rated at 20% above the motor shaft BHP; however, this percentage could

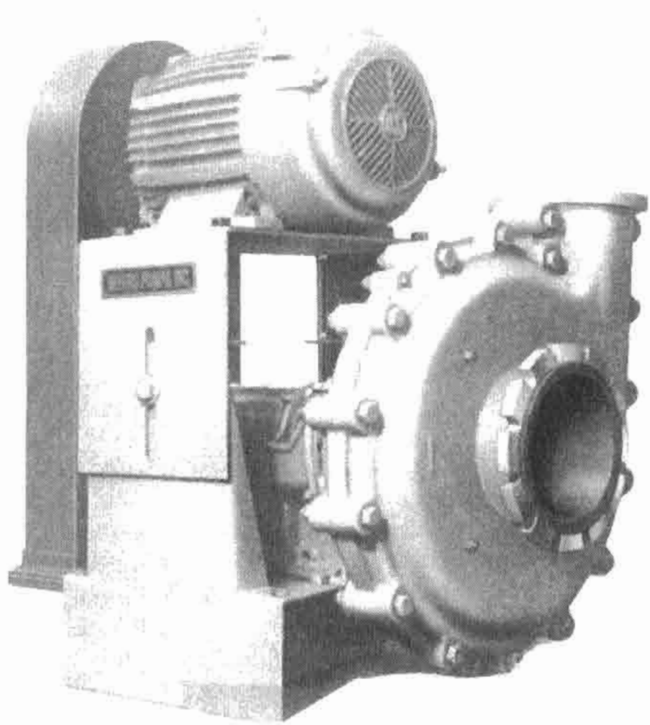


Figure 13-10. Typical belt-driven, overhead-motor-mounted slurry pump.

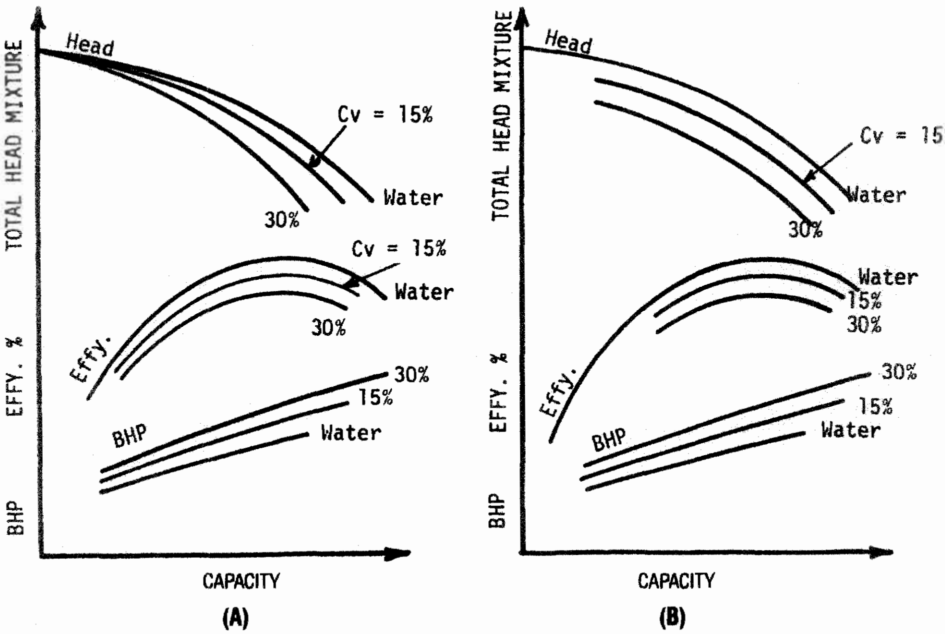


Figure 13-11. (A) Typical performance characteristic of nonsettling slurries; (B) typical performance characteristic of settling slurries.

go over 100% in badly controlled systems. For this reason large margins are built into the design of slurry pumps and motors. Usually the motor is mounted above the pump on an adjustable frame (belt adjustment) to save space and to safeguard against flooding. (Refer to Figure 13-10.)

The Effect of Slurries on Pump Performance

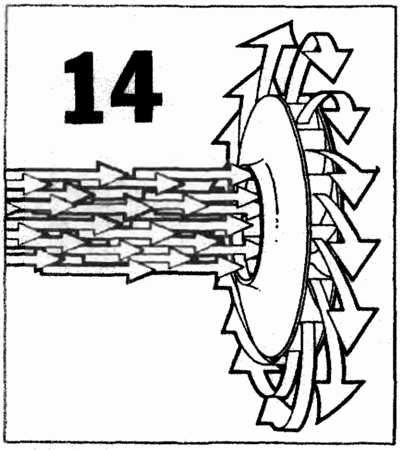
When centrifugal pumps are required to handle slurries, it is standard practice to publish pump performance curves based on clear water performance. Therefore, to predict the performance of pumps handling slurries of different characteristics, correction factors are applied.

When handling slurries, the pump performance is mainly affected by the solid particle diameter, specific gravity, and concentration. Very fine particles in a slurry can be "nonsettling" and cause it to behave as a homogeneous Newtonian liquid with an "apparent" viscosity. Slurries with very fine solids in suspension (usually less than 100 microns) will retain liquid-like characteristics at volumetric concentrations very near to the limiting voidage, and limits are related only to the effects of high viscosity. Figure 13-11a shows typical nonsettling slurry pump characteristics.

Where there exists a density difference between the conveying liquid and the solid particles, the particles will tend to settle. Usually slurries with a distribution of larger particles will be "settling," and the particles and the liquid will exhibit their own characteristics. As the liquid passes over the particles, energy is dissipated due to the liquid "drag" that reduces pump head and efficiency.

Actual tests indicate that for practical purposes, the amount of head derate will be the same as the efficiency derate. A typical pump performance characteristic for slurry mixture with coarse particles is shown in Figure 13-11b.

Performance correction factors for slurries are usually based on previous test data. In the absence of such data, reference should be made to the pump manufacturer.



Hydraulic Power Recovery Turbines

by **Rolf Lueneburg** and
Richard M. Nelson
Bingham-Willamette, Ltd.

The potential for power recovery from high-pressure liquid-streams exists any time a liquid flows from a higher pressure to a lower pressure in such a manner that throttling occurs. As in pumping, this throttling can exhibit a hydraulic horsepower (HHP) and a brake horsepower (BHP), except that in throttling, these horsepowers are available rather than consumed. Hydraulic power recovery turbines (HPRT's) are used instead of throttling valves to recover liquid power.

The two main types of HPRT's are:

1. *Reaction*—Reverse running pumps and turbomachines in single- and multi-stage configurations with radial flow, (Francis), mixed flow, and axial flow (Kaplan) type runners. They come with fixed and variable guide vanes. The axial flow Kaplan propeller has adjustable runner blades.
2. *Impulse*—Most prominent is the Pelton wheel, usually specified for relatively high differential pressures and low to medium liquid flows.

This chapter will discuss reaction-type HPRT's; namely, the reverse-running pump (Figure 14-1) and machines specifically designed to run as HPRT's.

Basically all centrifugal pumps, from low to high specific speed and whether single- or multi-stage, radially or axially split, and in horizontal or vertical installations, can be operated in reverse and used as HPRT's. The discharge nozzle of the pump becomes the inlet of the turbine; the suction nozzle or bell of the pump becomes the outlet of the turbine, and

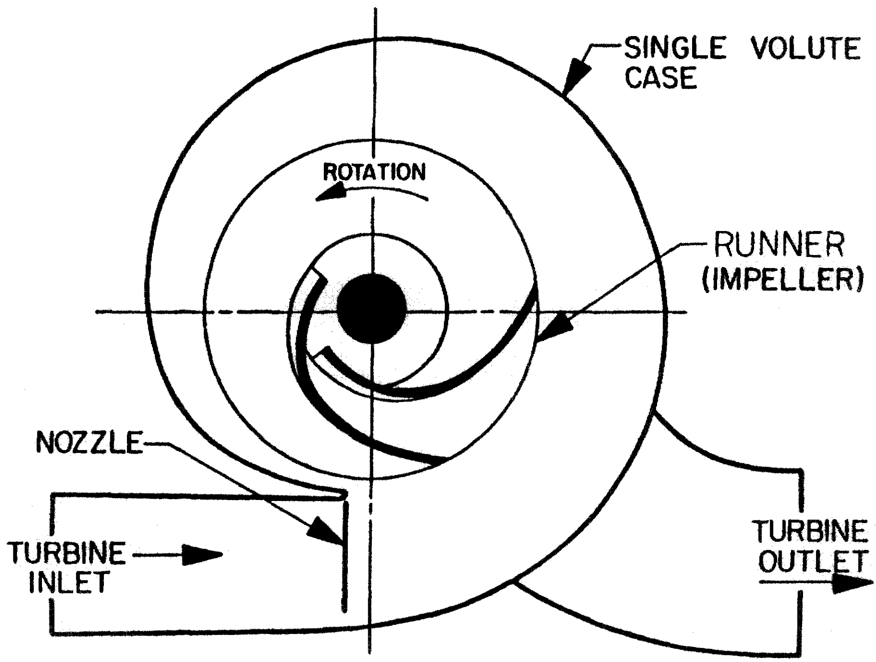


Figure 14-1. Typical single-volute-type pump used as a hydraulic turbine.

the impeller of the pump, rotating in reverse direction, becomes the runner of the turbine. Pumps are readily available, and many sizes are stock items. Reverse running pumps are an excellent alternative to conventional turbomachinery.

Centrifugal pumps operating as HPRT's have neglible operating costs. The installation costs are essentially the same as for an equivalent pump, and in terms of reliability and maintainability (R & M), they do have less maintenance costs because of their smoother and quieter operation. Also, since the efficiency of a pump operating as an HPRT is equal to or slightly better than the pump efficiency, the use of reverse-running pumps, or specially designed turbomachines, as primary or secondary drivers becomes very attractive.

The purchase price of an HPRT is generally approximately 10% greater than the price of a pump of equivalent design dimensions and metallurgy. This reflects the costs of the modifications that must be made to the impellers and volutes or diffusers, plus the complex testing that is required to verify the hydraulic performance of the finished machine. A single-stage HPRT may be profitable when as little as 30 BHP is recovered, while a multi-stage HPRT may be justifiable above 100 BHP (Figure 14-2). In general, many users find that HPRT's repay their capital cost within one to two years. However, before hydraulic power recovery can be feasible and economical, there must be sufficient flowing liquid capacity available at the necessary differential pressure as well as acceptable conditions of corrosion and erosion.

Selection Process

Before selecting HPRT's the following information is needed:

1. Available head range
2. Available capacity range
3. Back pressure at turbine outlet
4. Desired RPM
5. Chemical composition of the fluid
6. Temperature and specific gravity of the fluid at turbine inlet
7. Compressibility of the fluid
8. Gas entrainment
9. Preferable materials
10. Installation configuration
11. Vibration and noise level
12. Control equipment

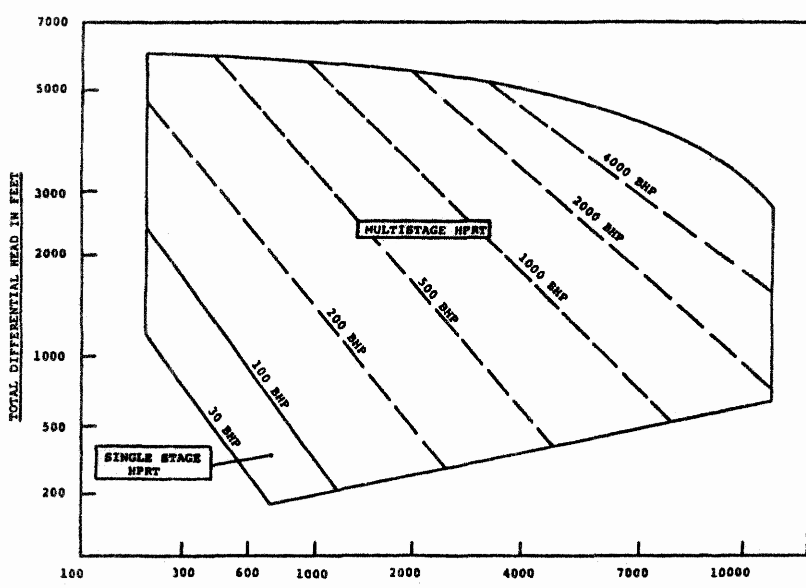


Figure 14-2. HPRT application range chart (from McClaskey and Lundquist).

The following criteria should be considered since they will help in specifying and classifying the HPRT.

Specific Speed

HPRT's are classified by their specific speed (N_s) which is a dimensionless quantity that governs the selection of the type of runner best suited for a given operating condition.

$$N_s = \frac{N \times BHP^{.5}}{H^{1.25}}$$

where N = Revolutions per minute
BHP = Developed power in horsepower
H = Total dynamic head in feet across turbine at best efficiency point (BEP)

The physical meaning of specific speed is: Revolutions per minute at which a unit will run if the runner diameter is such that running at 1-ft head it will develop 1 BHP.

The customary specific speed form used for pumps for classification of impeller-type characteristics is also applicable for HPRT (basically for reverse running pumps). The values will be similar to those for pumps.

The impulse Pelton wheels have very low specific speeds as compared to propellers (Kaplan) having high specific speeds. Francis-type runners cover the N_s range between the impulse and propeller types (Figure 14-3).

Net Positive Discharge Head

Net positive discharge head required (NPDHR) applies to an HPRT as does NPSHR to a pump to preclude cavitation and its attendant physical damage effects. Some literature refers to the term "total required exhaust head" (TREH) rather than NPDHR.

Test data have indicated that the NPDHR or TREH of a machine for the turbine mode is less than the NPSHR of the same machine for the pump mode at the same flow rate. The available net positive discharge head (NPDHA) or total available exhaust head (TAEH) at the installation side of the HPRT has to be higher than or at least equal to the NPDHR or TREH. This applies only to the reaction-type HPRT, since the impulse-type is a free jet action and is therefore not subject to low-pressure areas.

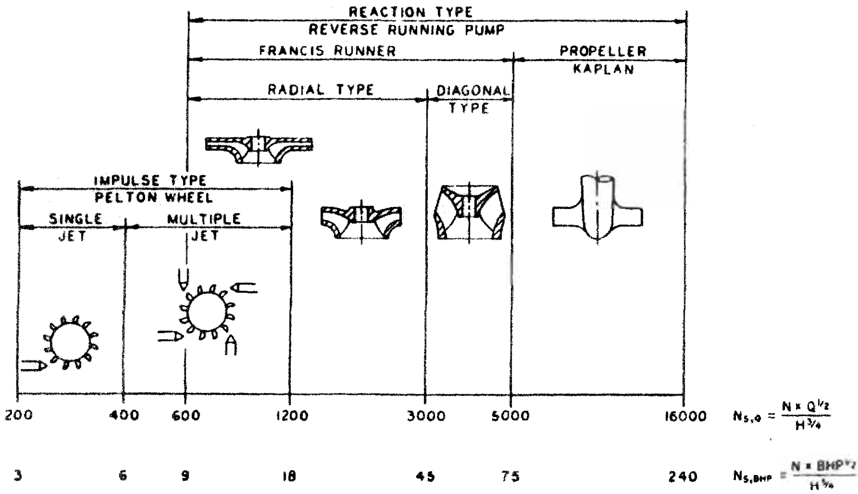


Figure 14-3. Turbine-type vs. specific speed. The ratio between $N_{S,Q}$ and $N_{S,BHP}$ is approximate only, since $N_{S,BHP}$ is a function of turbine efficiency.

Power Output and Affinity Laws

Power output is the rotational energy developed by the HPRT. Its value in BHP is calculated in a similar manner as for pumps except for the efficiency term.

$$\text{BHP} = \frac{Q \times H \times \text{sp gr} \times E_t}{3{,}960}$$

where
Q = Capacity GPM
H = Total head in feet
sp gr = Specific gravity
E_t = Overall Efficiency at the turbine mode

Variations in capacity, head, and BHP due to RPM (N) changes can be determined within reasonable limits by using the affinity laws, which normally are used for pumps but also apply to HPRT's (described in Chapter 2).

Configuration

The configuration of an HPRT is a function of the N_s, NPDHA or TAEH, BHP, RPM, installation requirements, and preferable vibration and noise levels. Similar to pumps, the specific speed, N_s, controls the number of stages for a given head capacity and RPM. The specific speed also specifies either single- or multi-stage HPRT's, which can be either the horizontal or vertical configurations.

The available net positive discharge head or total available exhaust head of the system for a given capacity will limit the RPM of the HPRT and determine whether the runner will be the single- or double-eye construction. Space requirements will specify the length of a horizontal unit or will call for the vertical installation. Energy-level requirements will limit the BHP per stage and may result in use of multi-stage units.

Turbine Performance Prediction

Prediction by Approximation

The performance characteristics of centrifugal pumps operating as hydraulic turbines may be approximated from pump performance characteristics. Typically, the capacity and head at the best efficiency point (BEP) will be greater for the turbine operation than for operation as a pump. The amount of shift from pump performance generally varies according to the specific speed. From tests, curves are developed that give ratios versus specific speed that are used to give the percent shift from pump to turbine performance.

Another procedure that is used to estimate the turbine performance from known pump performance characteristics is to simply divide the pump capacity and head values at the BEP by the pump efficiency at that point. This will give a rough approximation of the turbine head and capacity at the turbine BEP. Since there can be considerable error using these estimating procedures, they should only be used for preliminary selection of candidates for a particular application.

Prediction by Analysis

The performance characteristics of centrifugal pumps operating as hydraulic turbines and other turbomachines used exclusively as hydraulic turbines can be readily predicted with reasonable accuracy by use of a relatively simple analysis procedure. Only minor adjustments are to be expected to obtain the required performance on actual tests compared to the predicted performance by analysis. A modern computer program may be used to perform the calculations, print out the results, and also plot the performance curve.

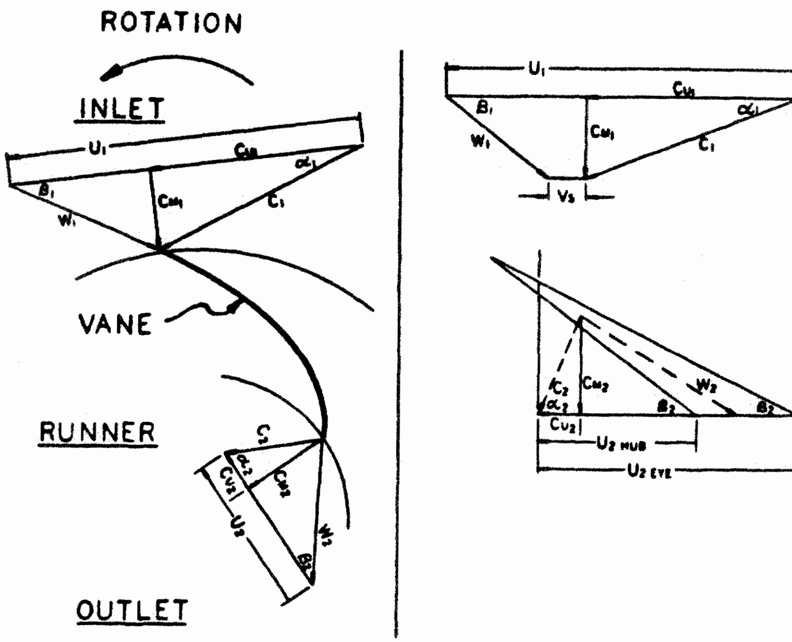
The prediction procedure generally consists of accounting for the various components that compose the total head characteristics. These are friction losses, absorbed head, shock loss, and outlet loss. Power losses due to internal leakage, disc friction, and mechanical losses are also calculated or estimated as appropriate. The calculations are made using the required turbine speed, flow capacities, viscosity, specific gravity of the fluid, and various combinations of mechanical data required for certain multi-stage turbines. There are many publications that cover basic theory and design of pumps that show how to calculate the head and loss components for pumps. These also apply to hydraulic turbines.

Friction losses. Certain components of the total dynamic head are attributed to friction losses. These are due to flow through the cases, volute nozzles, diffusers, guide vanes, and runners (impellers) as appropriate. These losses may be simply calculated as the resistance to the incompressible flow of fluid in a pipe, using appropriate friction factors, length to diameter (or hydraulic radius) ratios, and the velocity head.

Absorbed head. The absorbed head is derived from the well-known Euler's equations and velocity triangles, which have general validity for all conditions of flow through turbomachines. Refer to Figure 14-4 for illustration. In practice, the true velocities of flow and direction are never known. The idealized velocity triangles of the Euler head equation assume perfect guidance of the flow by the vanes. It is known that there is a deviation of the fluid from the vane direction, which is the phenomenon called "slip." This is a consequence of the nonuniform velocity distribution across the runner channels, boundary-layer accumulation, and any separation.

Actual prediction of "slip" cannot be predetermined in a practical manner. However, it has been found that "slip" factors used for pump design applied to the turbine outlet vectors produce good results. The absolute velocity at the runner inlet (C_1) is the average velocity of the liquid at the nozzles with a free vortex correction applied to account for the distance from the nozzles to the runner. The nozzles are the highest velocity throat areas of the volute cases, diffusers, or guide vanes as appropriate for the turbine construction.

Shock loss. The shock loss component of the total dynamic head is calculated as the velocity head ($V_s^2/2g$) due to the mismatch of the absolute



C = Absolute Velocity of Flow (Ft/Sec)

U = Peripheral Velocity of Runner (Ft/Sec)

W = Relative Velocity of Flow (Ft/Sec)

β = Vane Angle

V_s = Shock Velocity (Ft/Sec)

g = Acceleration Gravity

SG = Specific Gravity of Liquid

E_t = Overall Efficiency

Q = Total Flow (GPM)

H = Total Dynamic Head (Ft)

HP_L = Horsepower Losses (Leakage, Disc, Mechanical)

H_A = Absorbed Head $= \dfrac{U_1 C_{u_1} - U_2 C_{u_2}}{g} =$ Ft

$$\text{Output BHP} = \frac{H_A \times Q \times SG - HP_L}{3960} = \frac{Q \times H \times E_t \times SG}{3960}$$

Figure 14-4. Turbine theory.

inlet velocity (C_1) and the runner inlet vane angle ($\beta 1$). Refer to Figure 14-4 for illustration.

Outlet loss. The outlet loss component of the total dynamic head is calculated as the velocity head due to the absolute outlet velocity (C_2) times an appropriate loss coefficient ($KC_2^2/2g$). The loss coefficient may be taken as unity for many designs without serious error. Refer to Figure 14-4 for illustration.

Power loss. The power losses due to internal leakage, disc friction, bearings, and shaft seals applicable to pumps are also applicable to hydraulic turbines.

Turbine performance characteristics. The total dynamic head is the sum of the individual heads due to the friction losses, absorbed head, shock loss, and outlet loss. The hydraulic power is based on the total available energy to the turbine. The turbine output power is the absorbed head minus the power losses due to internal leakage, disc friction, bearings, and shaft seals. The turbine overall efficiency is the ratio of the output power to the hydraulic power. Refer to Figure 14-4 for illustration of terms.

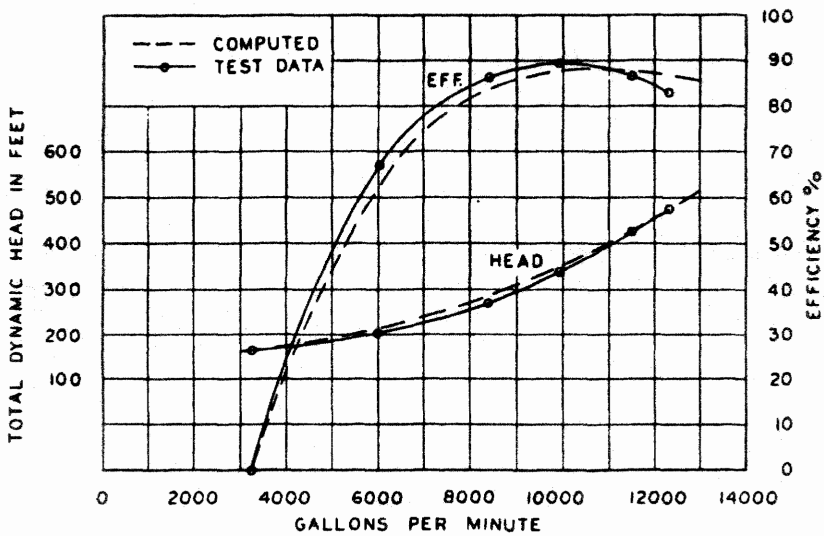
Predicted performance vs. test results. Figure 14-5 shows a typical comparison of the turbine performance characteristics determined by the preceding calculation procedure and the results obtained by actual test. Identical nozzle sizes were used.

Turbine Performance Prediction by Factoring

The performance characteristics of a hydraulic turbine may be quite accurately predicted by size factoring from a known performance at a specified specific speed. The rules that apply to pumps (described in Chapter 2) also apply to turbines.

Optimizing and Adjusting Performance Characteristics

The inlet and outlet velocity triangles as illustrated by Figure 14-4 are used to predict, adjust, and optimize the turbine performance. These give an instant picture of whether the turbine performance characteristics are expected to be optimum, satisfactory, marginal, or unsatisfactory. The optimum overall efficiency will generally be achieved when the shock loss at the inlet to the runner is near zero and the absolute velocity (C_2) at the outlet from the runner is near a minimum value.

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 INPUT DATA FILE NAME:
BOOKP
 14x16x26   HT HSA
 IMPELLER NO.  1   NO. OF STAGES  1
 VOLUTE DRG.  6685 IMPELLER DRG.   7131
 NOZZLE AREA 23.00 VORTEX DIST. 30.00
 TURBINE RPM=  1170
***************************************
```

GPM	HEAD	BHP	EFF.
TOTAL TURBINE PERFORMANCE			
3000	163.1	-8.8	-7.1
4000	173.7	38.6	22.0
6000	211.2	198.9	62.1
8000	270.7	446.0	81.5
10000	352.0	779.5	87.7
11000	400.9	978.5	87.9
11500	427.4	1086.0	87.5
12000	455.3	1198.9	86.9
13000	515.1	1440.6	85.2

Figure 14-5. A computed-vs.-test HPRT performance (courtesy of Bingham-Willamette Company).

The inlet velocity (C_1), which depends on the nozzle size, is critical to the turbine performance characteristics. It can produce a significant change to the turbine performance by its effect on the shock loss and the absorbed head. Therefore, the nozzle size is usually the main control for adjusting the turbine performance characteristics. A smaller-size nozzle area will generally shift the best efficiency point (BEP) to lower capacities and the larger size to higher capacities. The runner diameter controls the peripheral velocity (U_1), which theoretically could be adjusted to change the performance characteristics. Usually, however, only minor adjustments to the runner diameter can be made without distorting the hydraulic relationships.

Not all pump designs will make a good performing turbine without some modifications. Quite often the existing pump impeller vane angles at the outside diameter (runner inlet) and at the eye (runner outlet) are not a good combination for best performance. Also existing nozzle sizes and stationary passages may be too large or too small, which would require an alteration.

A pump operating as a hydraulic turbine will usually have an overall efficiency equal to or greater than the same machine operating as a pump, provided that the internal hydraulic parameters for turbine operation are good. This depends to a great extent on the runner vane angles and nozzle velocity considerations.

The overall efficiency of a turbine at capacities near the best efficiency point usually is improved by shaping the inlet ends of the runner vanes to a bullet-nose-type configuration and slightly rounding the inlet edges of the runner shrouds. The improvement is 1% to 2% at BEP capacity (100% capacity) and still achievable at $\pm 20\%$ of BEP capacity (80% to 120% capacity). The reason is the reduced turbulence of the runner inlet. The effect of surface finish on friction losses and the effect of leakage losses may be readily evaluated by the turbine performance prediction procedure.

Design Features (Hydraulic and Mechanical)

Reverse-Running Pump

Most centrifugal pumps in the low- to medium-specific speed range (N_s,Q = 600 to 5,000 or N_s,BHP = 9 to 75 (see Figure 14-3)) are suitable and capable of operating as HPRT's. Because of the reverse rotation, one has to check that the bearing lubrication system and threaded shaft components, such as impeller locking devices, cannot loosen. However, most pumps nowadays are designed to withstand reverse rotation.

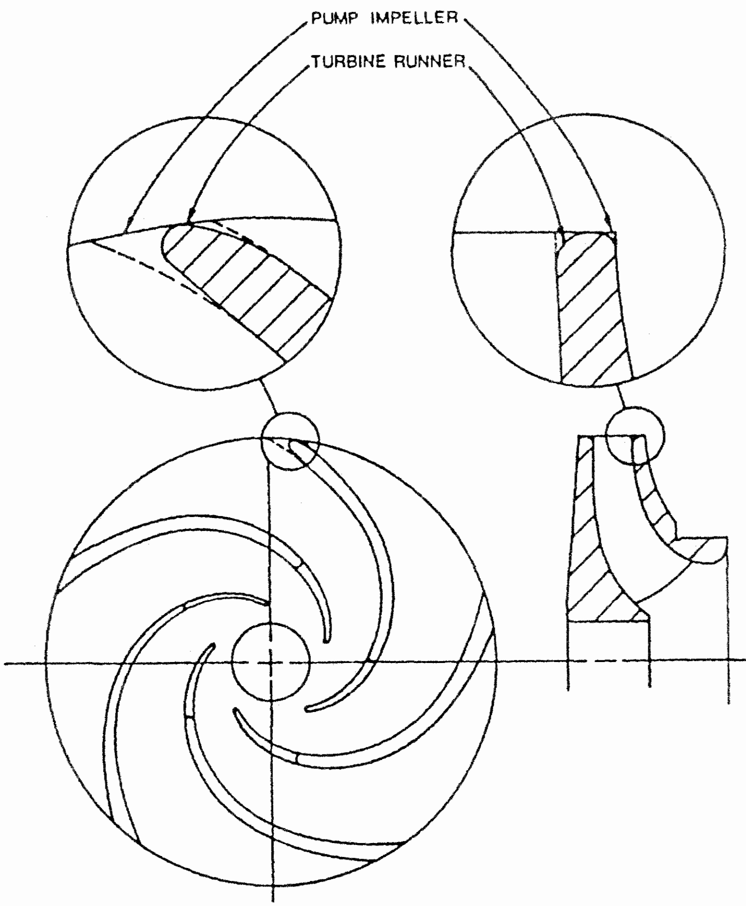

Figure 14-6. Rework of pump impeller for operation as an HPRT runner.

Trimming the runner (impeller) diameter, as is done for pumps, to shift the performance characteristics is not normally done for hydraulic turbines. The turbine runner diameter is selected for optimum running clearance.

The inlet ends of the turbine runner vanes are ground to a bullet-nose shape and the inlet edges of the runner shrouds are rounded slightly to preclude excessive turbulence for efficiency considerations (Figure 14-6). But before operating a pump as an HPRT at the same speed, one has to remember the change in performance characteristics. A comparison is shown in Figure 14-7.

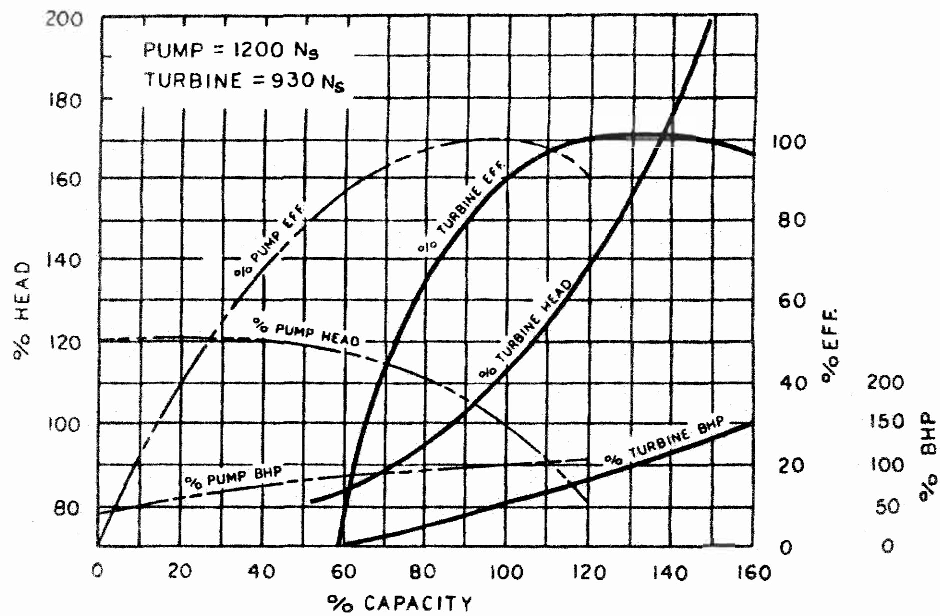

Figure 14-7. Comparison of pump and turbine characteristics at constant speed. Characteristics are the percent of pump best efficiency values taken as 100%.

Because of the higher capacity throughput and higher differential pressure, the HPRT power output is greater than the power requirements of the pump. Subsequently, the higher shaft stresses result in speed limitation unless the allowable stress or diameter of the shaft has been increased.

If a multi-stage pump is selected to operate as an HPRT, a heavier shaft can be installed, since most pump manufacturers build their multi-stage pump line with standard and heavy-duty shafts because of a larger number of stages, higher specific gravities, or high-speed applications. Single-case pumps are limited to the maximum pressure they can withstand. For high-pressure HPRT applications, either the allowable maximum working pressure of the pump case has to be increased or a double, or "barrel-type," case has to be selected. Barrel-type cases are also used for high-energy and low-specific-gravity-type HPRT's.

Another important check is evaluating the adequacy of the bearing design. Depending on specific speed, some pumps when operating as HPRT's at the same speed will have twice the differential pressure and

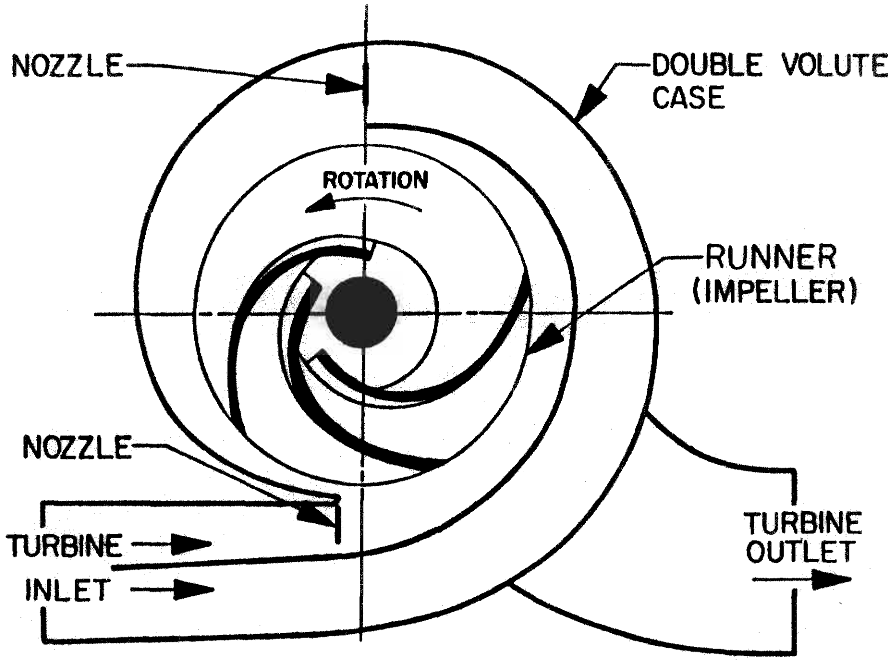

Figure 14-8. A typical double-volute-type pump used as a hydraulic turbine.

radial force across the impeller or runner. This results in increased radial loading of the bearing and could present a problem with end-suction pumps; especially, if the case is of the single-volute design. Stronger bearings will increase the bearing life; however, the resulting greater shaft deflection at the impeller wear rings and seal faces could decrease wear ring and seal life and increase the vibration level. A pump case in double-volute design (Figure 14-8), which results in radial balance, will solve these problems. Besides an increase in radial load there will be a change in axial thrust. If required, a change in wear-ring or balancing-device diameter will reduce the axial thrust to an acceptable level.

In general, pumps are built to customer specifications such as API 610. Therefore it is natural that the same specifications will apply to pumps operating as HPRT's. These customer requirements cover: running clearances, limitations for horizontal split-case pumps in relationship to operating temperature and specific gravity, nondestructive examination (NDE), shaft sealing and seal flushing, bearing life, lubrication and cooling system, baseplate design, material selection for corrosion and erosion protection as well as nozzle loading, and noise and vibration level to name a few. With all these considerations in mind, a well-designed pump will operate smoothly, quietly, and reliably as an HPRT.

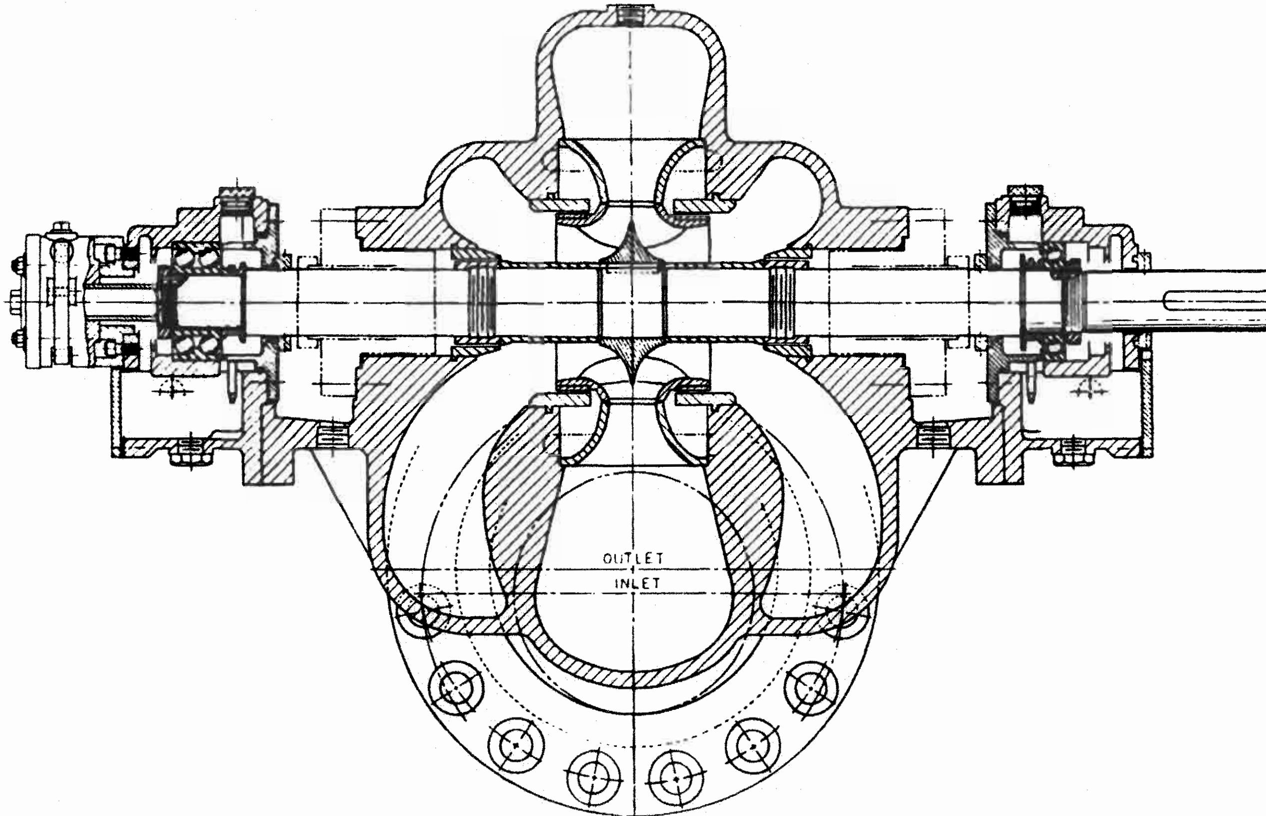

Figure 14-9. A single-stage double-eye Francis-type HPRT.

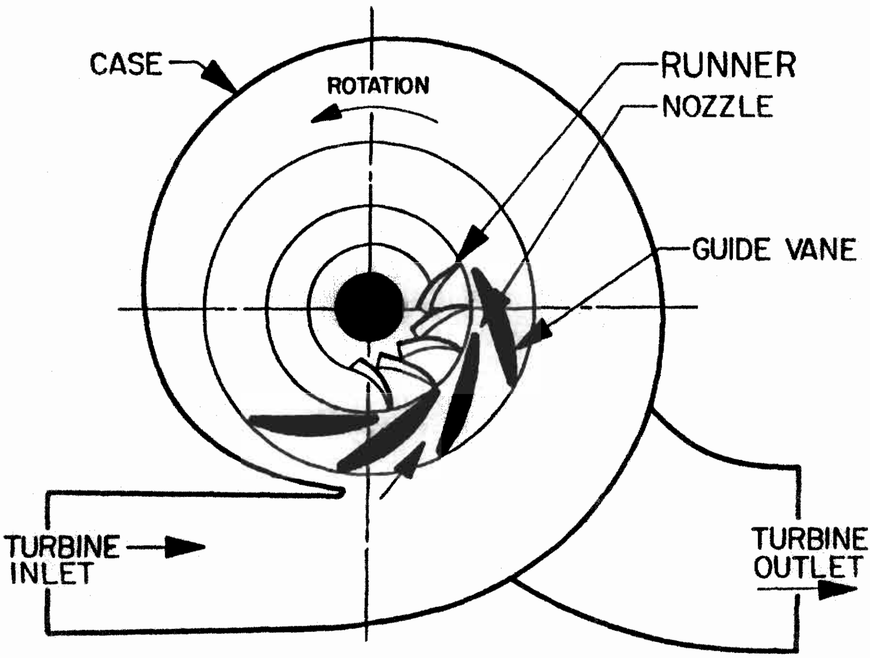

Figure 14-10. HPRT construction using single-volute-type case, turbine runner, and guide vanes.

Turbine Design with Fixed Guide Vanes

Most pump manufacturers offer, in addition, different lines of turbomachines designed specifically to be applied as HPRT's. They are basically developed to cover a range of performance generally not available with a reverse-running pump. One of these HPRT's is shown in Figure 14-9.

This HPRT features an axial-split single-volute-type scroll case, a single-stage double-eye Francis-type runner, and a removable guide vane component (Figure 14-10), that controls the inlet flow to the runner.

This type of HPRT has a fixed performance characteristic. However, the design can accommodate seasonal or "plant turn-down" flow-capacity conditions by changing to a different guide vane assembly or by using an adjustable guide vane assembly for optimum performance. The runners used in this type of HPRT are quite different from the conventional pump impeller in that it has a greater number of vanes and generally larger vane angles.

The eye area at the outlet of the runner is extra large and the eye vane angles are carefully selected to accommodate any potentially large amounts of vapor that may evolve out of solution by expansion through the turbine. The combination of the number of nozzles in the guide vane assembly and the number of vanes in the runner is selected to preclude in-phase torque pulses. For higher working pressures or temperatures or lower specific gravities, this type of HPRT can be supplied in a radially split and/or centerline-mounted volute case. For most applications, however, the axial-split-type is sufficient and preferred, basically because of the ease of maintenance and inspection of the rotating element.

The runner is essentially balanced in both radial and axial directions. The thrust bearing in the outboard bearing housing will take the axial thrust resulting from upset conditions such as unequal amounts of vapor in the two eye areas of the runner. For higher-speed applications, these HPRT's are furnished with Kingsbury thrust and sleeve radial bearings.

For higher differential pressure and lower capacity, multi-stage HPRT's with guide vane assemblies are available. These are generally lower-specific-speed turbomachines with single eye and narrow runners to avoid large bearing spans. In general, the mechanical design criteria are the same as used for centrifugal pumps running in reverse.

Turbine Design with Internally and Externally Adjustable Guide Vanes

Specially designed HPRT's include the feature of an adjustable guide vane assembly, which can be furnished for a single-stage or multi-stage HPRT. The method of adjusting the guide vane assembly is made possible by an internal or external design feature. The advantage of this variable vane assembly is the capability of operating more efficiently over an extended flow range compared to an HPRT or a reverse running pump with fixed inlet guide vanes.

The performance characteristics of an HPRT can be varied over a considerable range by changes to the velocity of the liquid passing through the guide vane assembly. For optimum performance, this is best accomplished by changes to the flow-cross-section area formed by the vanes and the side walls of the assembly when aligned at a proper angle to the runner. A decrease in the flow-cross-section area will generally shift the optimum efficiency to a lower flow range.

The typical performance characteristic curve for an HPRT with fixed guide vanes is illustrated in Figure 14-11. The hydraulic turbine is essentially like an orifice in a fixed-pressure-differential system. The operating point will be where the particular head, capacity, speed, and power relationship is satisfied.

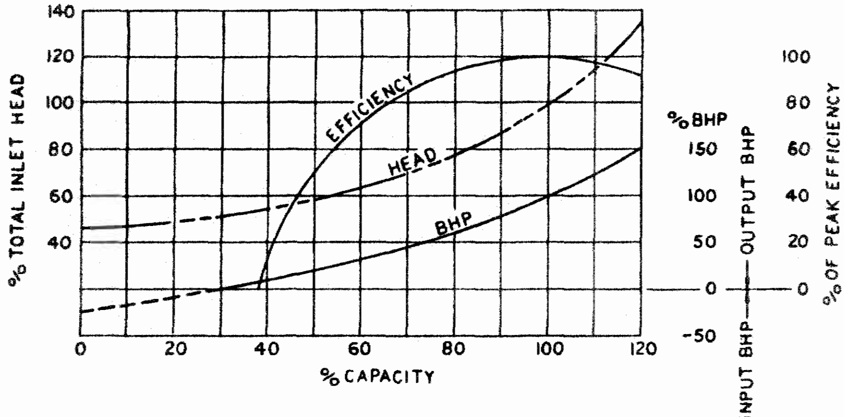

Figure 14-11. Typical performance characteristics for an HPRT with fixed guide vanes.

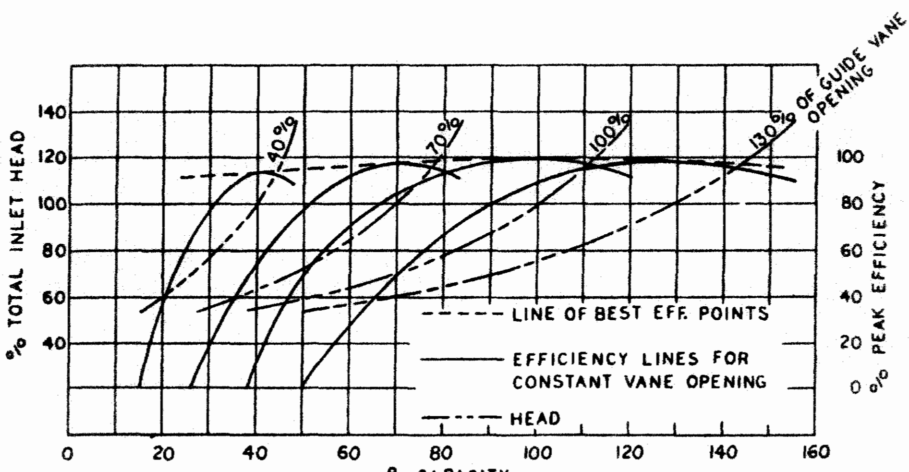

Figure 14-12. Typical performance characteristics for an HPRT with an adjustable guide vane assembly.

The typical performance characteristic curves for an HPRT with an adjustable guide vane assembly are shown in Figure 14-12. The adjustable guide vane assembly performs, as one can see, as a variable orifice in a large number of fixed-pressure-differential systems. The vane setting will control the flow at a certain differential pressure.

In Figure 14-13, the best efficiency points of each vane setting or opening are connected to a single line and compared with the efficiency curve of a hydraulic turbine with fixed inlet vanes. Because the runner is de-

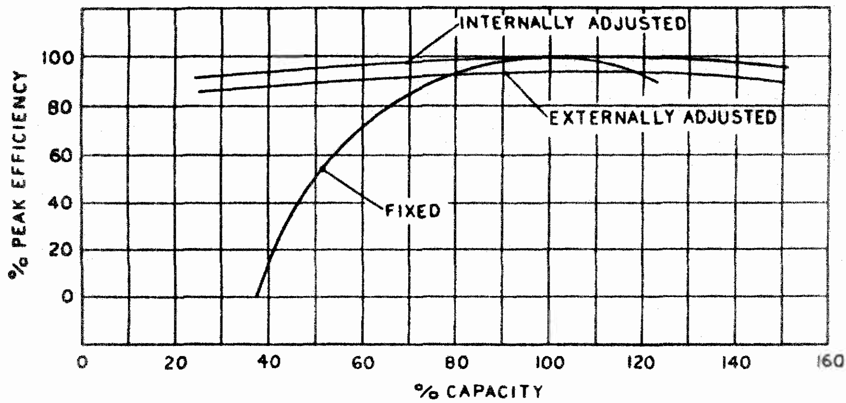

Figure 14-13. Efficiency comparison between HPRT's with fixed, internally adjustable guide vanes and those with externally adjustable guide vanes.

signed with fixed blade angles, for the 100% capacity only, there will be a small efficiency drop towards the higher capacities and a slightly larger drop for the lower capacities. The slightly larger efficiency drop towards the lower capacities is the result of the reduced specific speed of the turbine.

The slightly lower efficiency at 100% capacity for the HPRT with an externally adjustable guide vane asembly is primarily the result of the runner vane angle and possibly the slightly higher inner leakages due to the mechanical design.

The losses at 100% capacity for the HPRT with an internally adjustable guide vane assembly are equal to the one with fixed guide vanes. The internal adjusting feature is used where the capacity variations are not frequent, since the HPRT has to be disassembled to adjust and lock the guide vane assembly to a different configuration (Figure 14-14).

The external adjusting feature makes it possible to vary the guide vane setting during operation of the HPRT. According to the available capacity, a level controller or capacity indicator sends an air or electric signal to the turbine-actuator, which in turn changes the setting of the guide vane assembly until the flow through the openings equals the available capacity and the signal stops. Continuous capacity changes will result in continuous resetting of the guide vane assembly, thus making it possible to operate the HPRT always at its BEP (best efficiency point), if the differential pressure across the turbine remains about constant.

The externally adjustable guide vane assembly features are illustrated in Figure 14-15. The design incorporates the conventional principle of

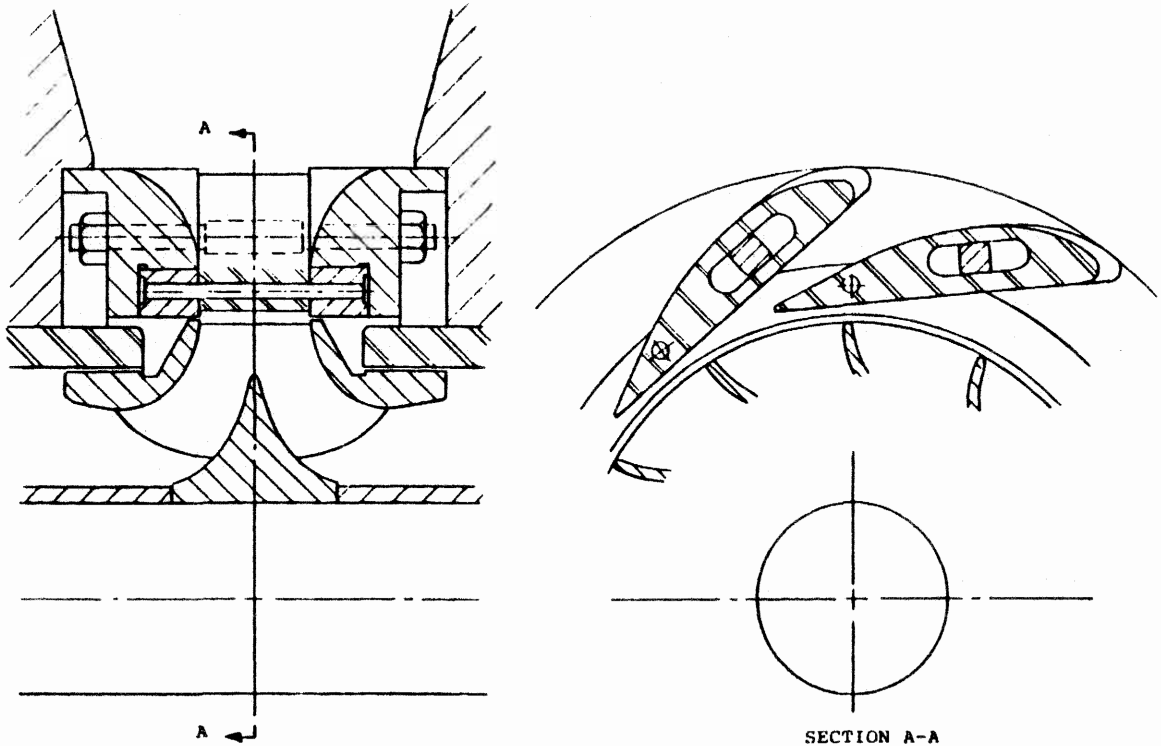

Figure 14-14. Internally adjustable guide vane assembly (courtesy of Bingham- Willamette Company).

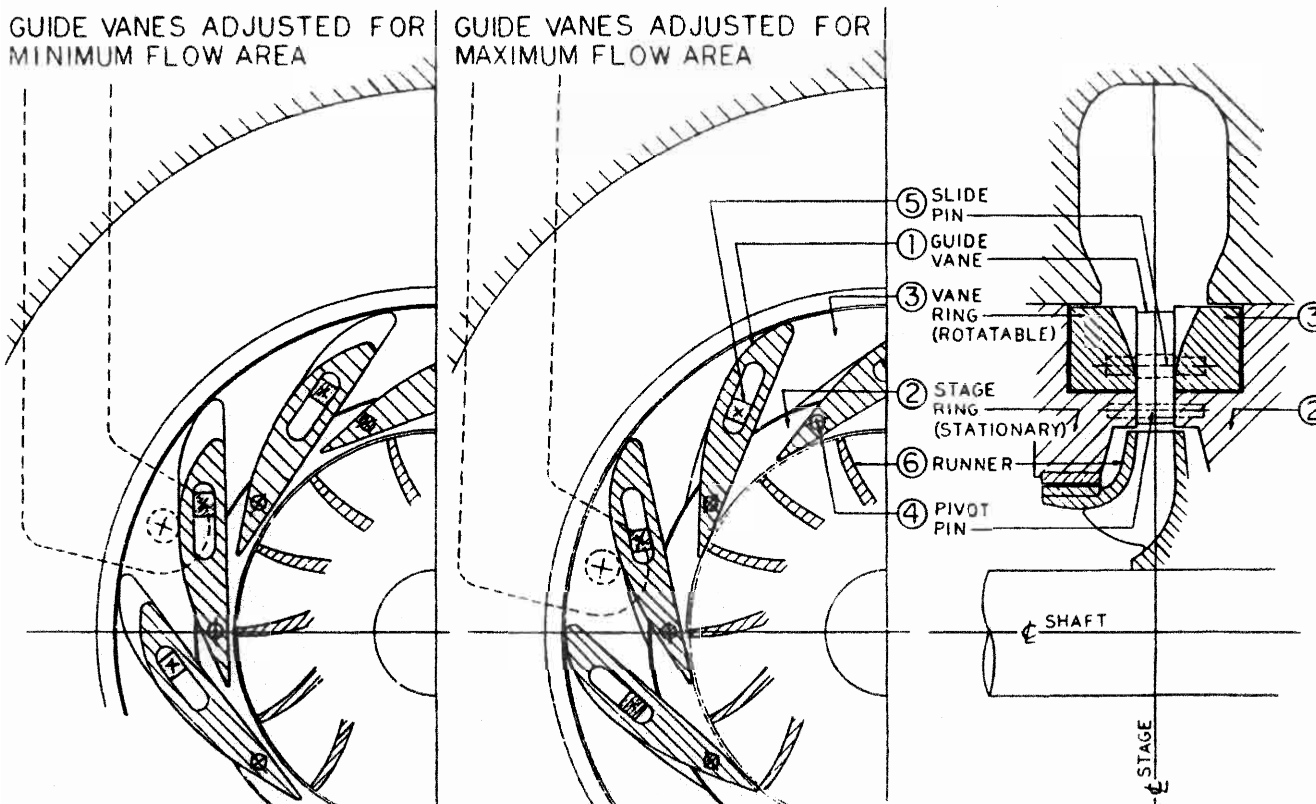

Figure 14-15. Externally adjustable guide vane assembly.

tilting the guide vanes (Part #1) about a pivot pin (Part #4) parallel to the runner (Part #6) shaft axis to vary the velocity of the liquid flowing through the assembly at a proper flow orientation angle relative to the runner. Each guide vane is held in position by the pivot pin and by a slide pin (Part #5), which moves the guide vane by its position in the slot through the vane.

The pivot pins are located in the two stationary vane rings or stage pieces (Part #2) and the slide pins are assembled to the two rotatable vane rings (Part #3). The stationary and rotatable rings establish the width of the inlet opening. They are the side walls of the vane assembly. The operating position of the vanes and the resultant through-flow cross-section area is dependent on the angular position of the rotatable vane ring in relation to the stationary rings.

Between the guide vanes, the rotatable vane rings are shaped in a manner to achieve the correct velocity increase for each through-flow cross-section area.

The rotatable vane rings perform the additional function of avoiding undesirable vane flutter, by a clamping action due to developed differential pressure. A reduction in pressure occurs in the flow passages due to the increase in velocity of the fluid, while the pressure acting on the outward side areas of the rotatable rings is essentially the same as at the entrance to the vane passages.

Because of the relatively large outward side areas of the rotatable rings, the clamping force is higher than the different hydraulic forces that act on the guide vanes and could cause vane flutter. However, the force is not restricting the adjustment of the guide vane position during operation.

The cross-section of this HPRT is shown in Figure 14-16. The turbine is built basically like a multi-stage pump with standard bearing housings. The runner eyes of each stage face all in the same direction and a drum takes care of balancing the axial thrust.

Figure 14-17 shows the crossunder in the bottom half. There are no crossovers. The top half contains the yoke assembly, which moves up and down and creates the rotational position of the rotatable rings and subsequently the resultant through-flow cross-section area of the guide vane openings.

A crossbeam as shown in Figure 14-18 connects the yokes for synchronous travel. Individual setting of through-flow areas for each stage is possible by adjusting the nuts on the crossbeam. If required, through-flow areas for each stage can be adjusted differently to allow for an increase in the specific volume for compressible liquids, when the pressure reduces from stage to stage. This is another feature to achieve optimum performance. The up-and-down movement of the beam can be achieved by an electric or pneumatic actuator that is mounted on top of the beam cover.

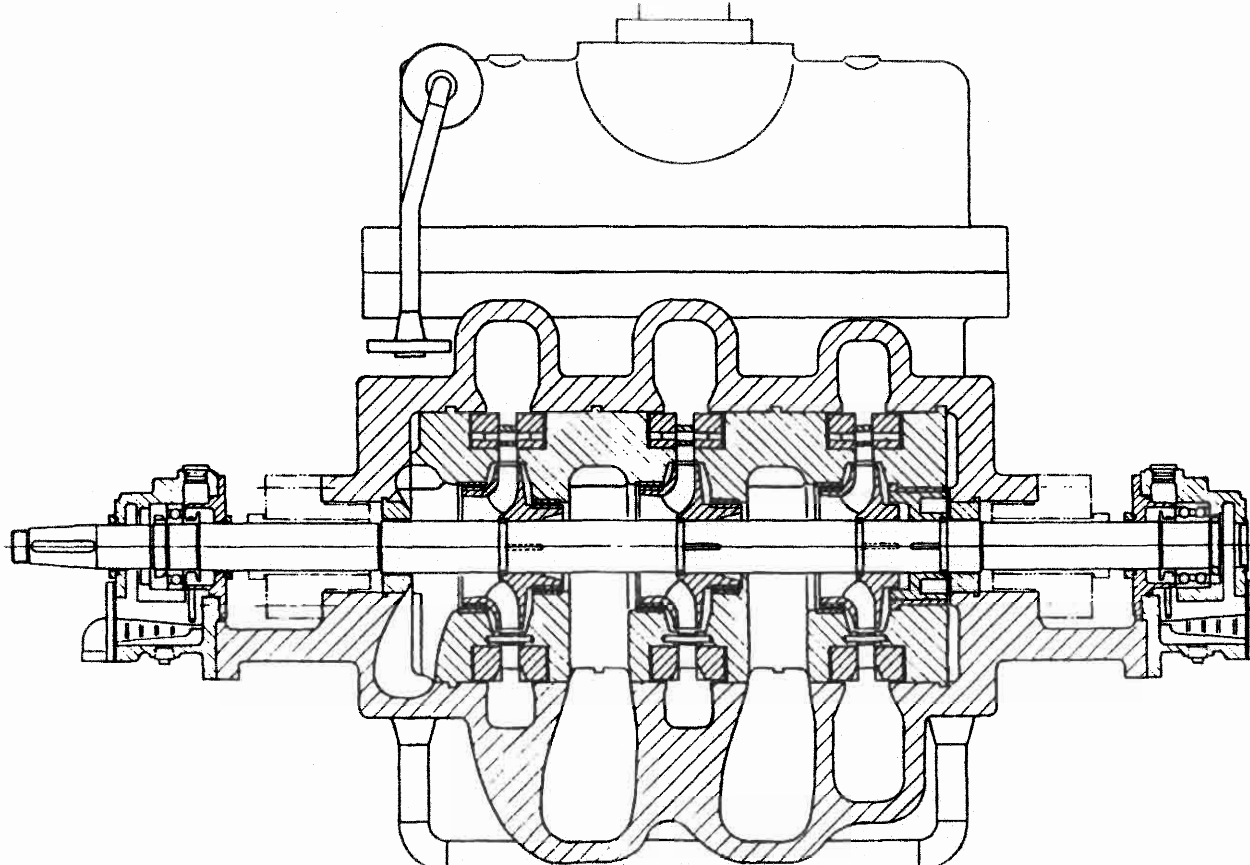

Figure 14-16. Cross section of a three-stage HPRT with externally adjustable guide vanes (courtesy of Bingham-Willamette Company).

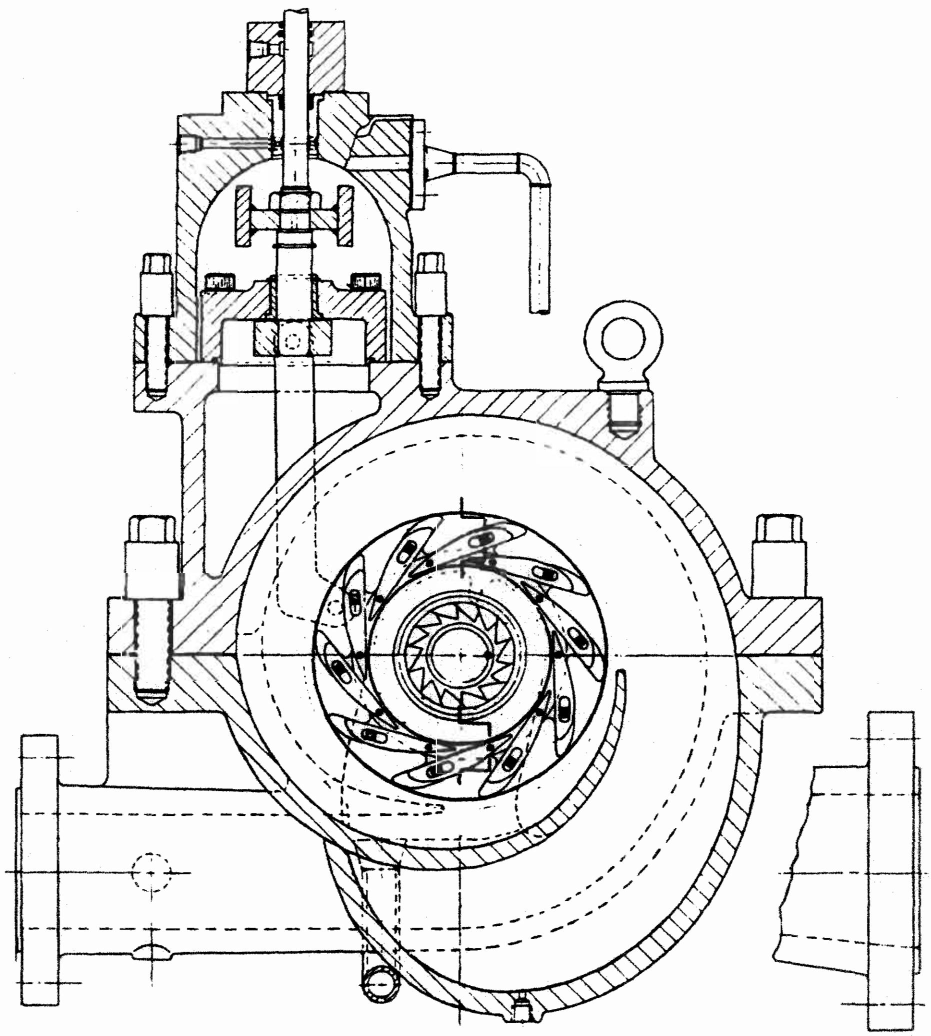

Figure 14-17. Cross section of a three-stage HPRT illustrating the guide vane adjusting mechanism *(courtesy of Bingham-Willamette Company)*.

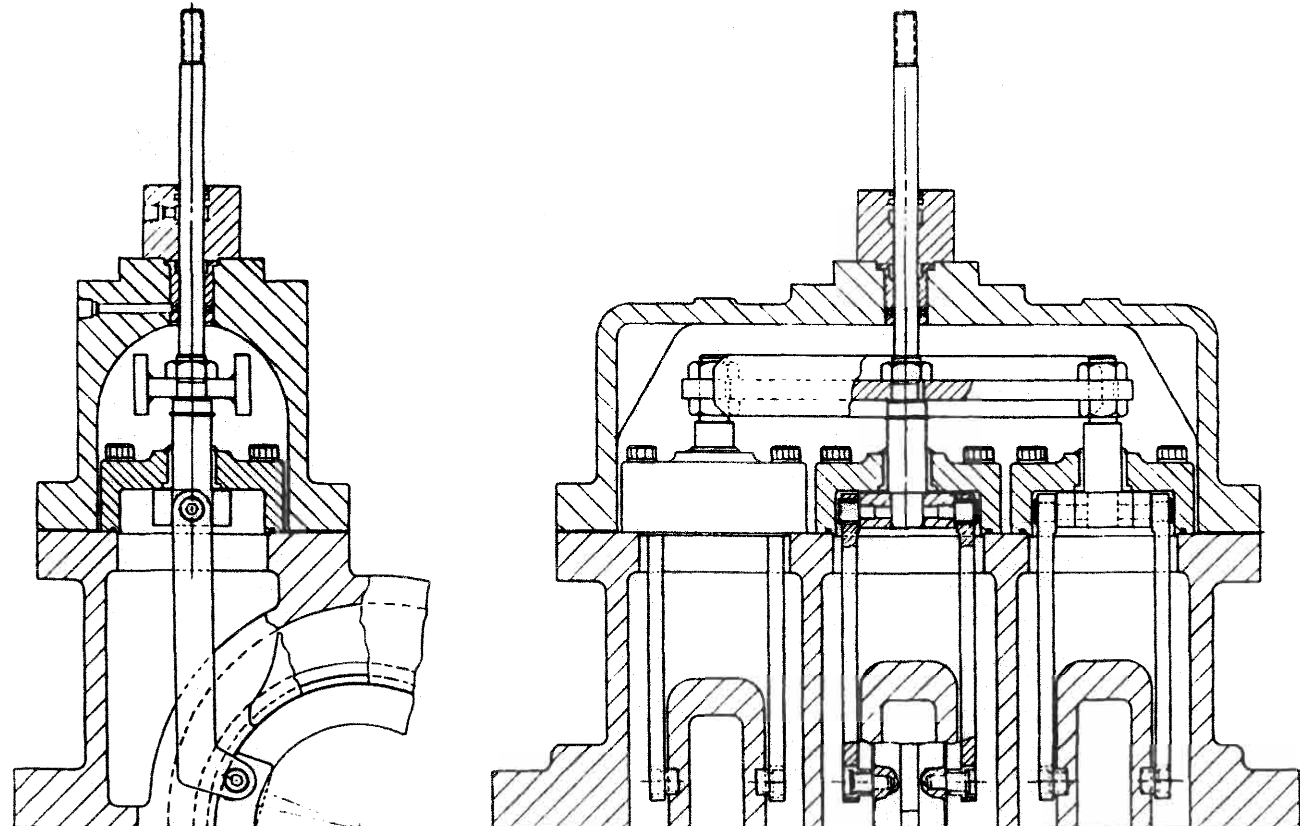

Figure 14-18. A crossbeam connects the three yokes to facilitate synchronous vane adjustment (courtesy of Bingham-Willamette Company).

Operating Considerations

The product handled by a hydraulic turbine may be a single-phase liquid, a multiphase liquid-gas mixture, or a slurry composition.

Hydraulic turbines have been extensively used for two-phase, liquid-gas flow streams where there is a potential for a substantial amount of gas released as the product passes through the turbine. There may also be small amounts of "free" gas at the turbine inlet. With a decrease in pressure, gas is subject to be released from the liquid with a resultant increase in volumetric flow. The effects of the potential vaporization at the various turbine stage pressures is evaluated to assure proper turbine performance. Generally, this may be accomplished by limiting the two-phase flow velocities at the runner (impeller) outlet eye to a reasonable value. It is also appropriate to give consideration to the runner (impeller) design to assure proper vane angles and eye sizes to accommodate any potential vapor release from the fluid stream. Actual field experience known to the author has shown that calculated two-phase flow velocities at the turbine outlet runner eye up to 150 ft/sec can be accommodated with no adverse effects. This velocity is suggested as a guideline for HPRT's whether they be single- or multi-stage types. Using this limit, the two-phase flow rate by volume can be at least three or four times the single-phase flow rate for many applications.

Theoretically higher output horsepower should be achieved by gas expansion through the turbine since the increase in volume means more work done. However, many reports have indicated that the expected additional power has not been realized. One explanation may be that the product passes through the turbine too fast for vapor-equilibrium to be obtained. For example, consider the time it takes for the carbon dioxide to escape from a bottle of carbonated beverage when the cap is removed. It does not all escape instantly. Another reason may be the fact that as the gas expands, the product velocity increases, and causes additional losses to occur.

For multi-stage hydraulic turbines, the nozzles may be sized differently from stage to stage to accommodate any theoretical increase of the volumetric flow as the pressure is reduced.

Performance Testing

Performance tests for hydraulic turbines may be accomplished by use of a centrifugal pump to furnish the head and flow capacity necessary to drive the turbine and to verify the turbine performance throughout its operating range. An induction motor excited by AC power from the utility system is used as an induction generator to absorb the output from the hydraulic turbine when gain over synchronous speed is achieved and to drive the hydraulic turbine at low flow capacities where input HP is required.

The output and input HP is determined by use of a wattmeter with efficiency curves for the induction machine. A torque meter may also be used to measure the power. A venturi meter or an orifice is used to measure the flow capacity. Dead weight testers and Bourdon-type gauges are used to measure the head. The RPM change from synchronous speed is counted by use of a strobotac and stop watch.

The turbine test may be performed at a reasonable reduced speed to facilitate testing. The performance at normal speed is then determined by applying the "affinity laws". The availability of a drive pump with sufficient head and flow capacity is a determining factor for the test speed. Cavitation tests are needed for hydraulic turbine performance characteristics. This is best determined by reducing the turbine outlet pressure and observing any resulting changes in the total dynamic head of the turbine, the power, the capacity, or efficiency. Measurements of noise, pulsation, and vibration accompanying the operation of the turbine during the cavitation test should be recorded.

Applications

Any continuous process where high pressure liquid or partially gas-saturated media is let down to a lower pressure across a reducing device is a potential application for an HPRT. Such potentials are:

- In pipeline service on the downside of high mountain ranges to keep the pipeline full and avoid excessive pressures.
- In bleeding products from a high-pressure point in the pipeline to storage.
- In geopressured-geothermal zones where high-temperature water is at a very high pressure. The formation pressure may exceed 10,000 psig, while pressure at the surface may approximate 2,000 to 6,00 psig depending on flow rates.

The early HPRT applications were basically in the noncorrosive and nonerosive service. Modern plants use HPTR's nowadays in mildly severe services such as:

- In hydrocracking operations, where boosting pressure of charge stocks to the 1,500–2,000 psi operating pressures used in modern hydrocracking processes requires large quantities of energy. Effluent from the reactor is still at high pressures, however, so that much of this energy can be recovered if an HPRT is incorporated in the drive train.

- In the gas processing industry where crude gas is scrubbed by a high pressure fluid medium such as potassium carbonate or amine in order to remove unwanted components. For the purpose of regeneration and recycling, the pressure has to be reduced; in other words, possible energy recovery has been made available.

The pressure can be reduced by using pressure breakdown valves; however, the differential pressure will be converted into thermal energy, which is either wasted or very uneconomical to recover. A relatively efficient method for pressure reduction and energy recovery is by the use of HPRT's.

HPRT's will convert the differential pressure into rotational energy, which can be utilized in helping to drive the centrifugal pump that returns the regenerated medium to the absorber. Both major types, namely, the reaction and impulse types, are used in the gas processing industry. Figure 14-19 shows the operational system using a reverse-running pump with fixed guide vanes. Since in a recycle system the recovered energy is smaller than the required energy to drive the pump, an electric motor or steam turbine on the other side of the pump is used to cover the energy difference and to maintain as a second function a constant RPM of the entire train.

The desired flow can be obtained either by changing speed of the assembly (steam turbine drive) or by throttling the pump output (motor drive), which means loss of energy. Unfortunately, the operating behavior of the standard reverse-running pump with fixed guide vanes requires a controllable throttling inlet valve for reduced capacity and a bypass line for increased capacity. Both represent additional energy losses (see Curve "A" and "B" in Figure 14-20).

Figure 14-21 illustrates the system using an HPRT with variable guide vanes.

The losses of the system in Figure 14-19 are avoided. The function of the inlet throttling valve (reduced capacity) and the bypass (increased capacity) are served by the variable inlet guide vanes installed in the HPRT, which satisfy the following purposes:

- Regulation of the capacity by varying the cross-section area of the guide vanes depending on the level in the absorber.
- Feeding the medium to the runner in a definite direction.
- Complete or partial conversion of the differential pressure into kinetic energy.

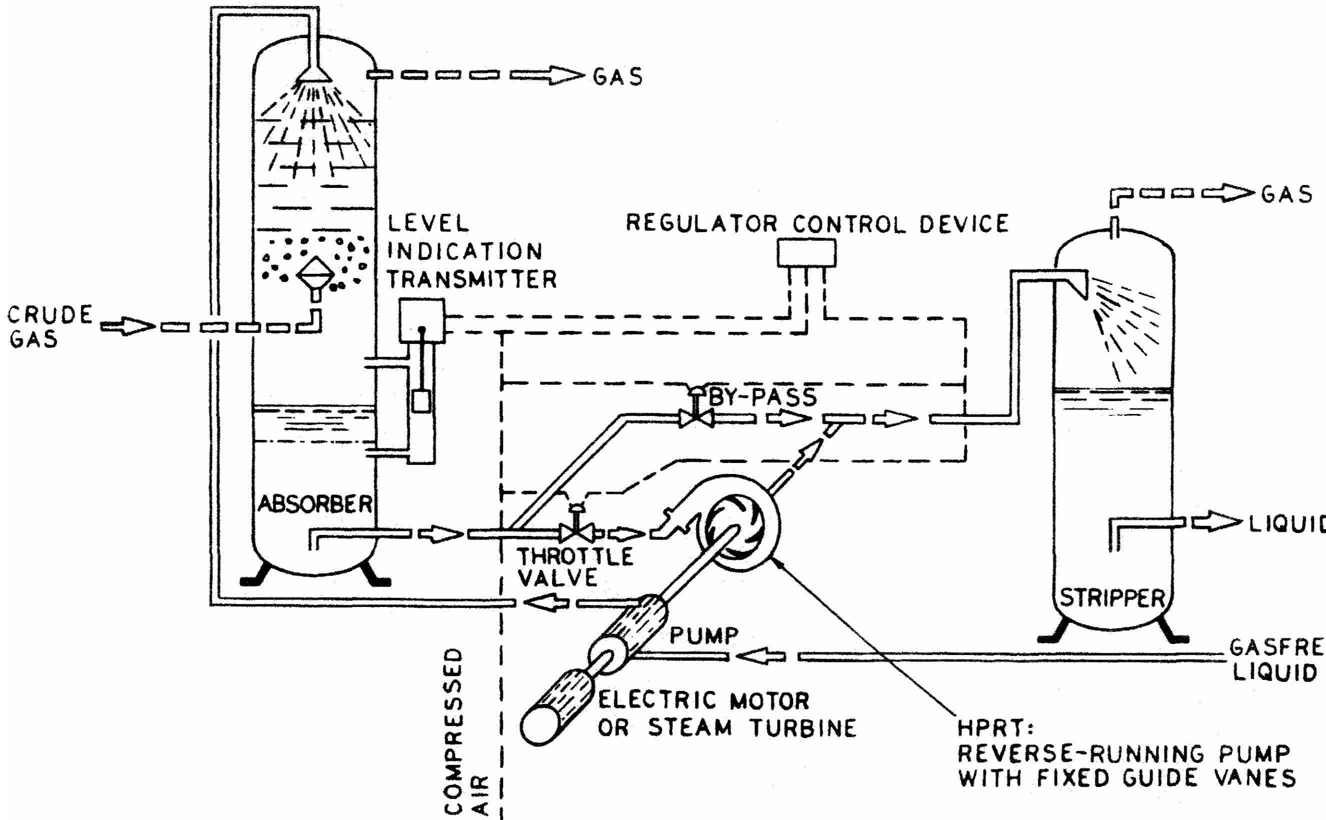

Figure 14-19. Flow diagram of hydraulic energy recovery using a reverse-running pump with fixed guide vanes (from Franzke).

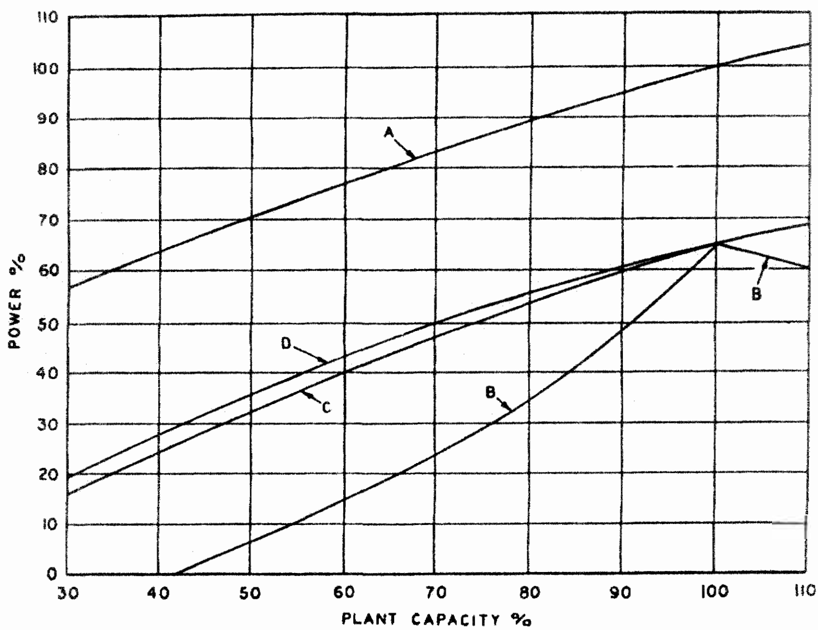

Figure 14-20. Power vs. plant capacity.

- Tripping the HPRT in the event of failure of compressed air, oil pressure, or power.

As a result of the improved supply of medium, the HPRT will still generate energy when the plant capacity drops below 40%, whereas the reverse running pump with fixed guide vanes will start to consume energy. (See Curve "A" and "C" in Figure 14-20).

Installation of a Pelton-type HPRT is shown in Figure 14-22. The pump-driver train arrangement is identical to the ones with the reaction-type HPRT's. However, a horizontal-level controlled tank has to be placed immediately below the turbine outlet to avoid paddling of the wheel in fluid flowing from the wheel. Paddling of the wheel in the medium results in high energy losses and strain being placed on the turbine. The efficiency of a properly installed Pelton-type HPRT is slightly higher at lower flows than the one of the HPRT with adjustable guide vanes, since the effect of the lower specific speed is not as significant (see Curve "A" and "D" in Figure 14-20).

HPRT's find many more applications such as services in hydropower stations and cooling towers and as reversible machines for pumped stor-

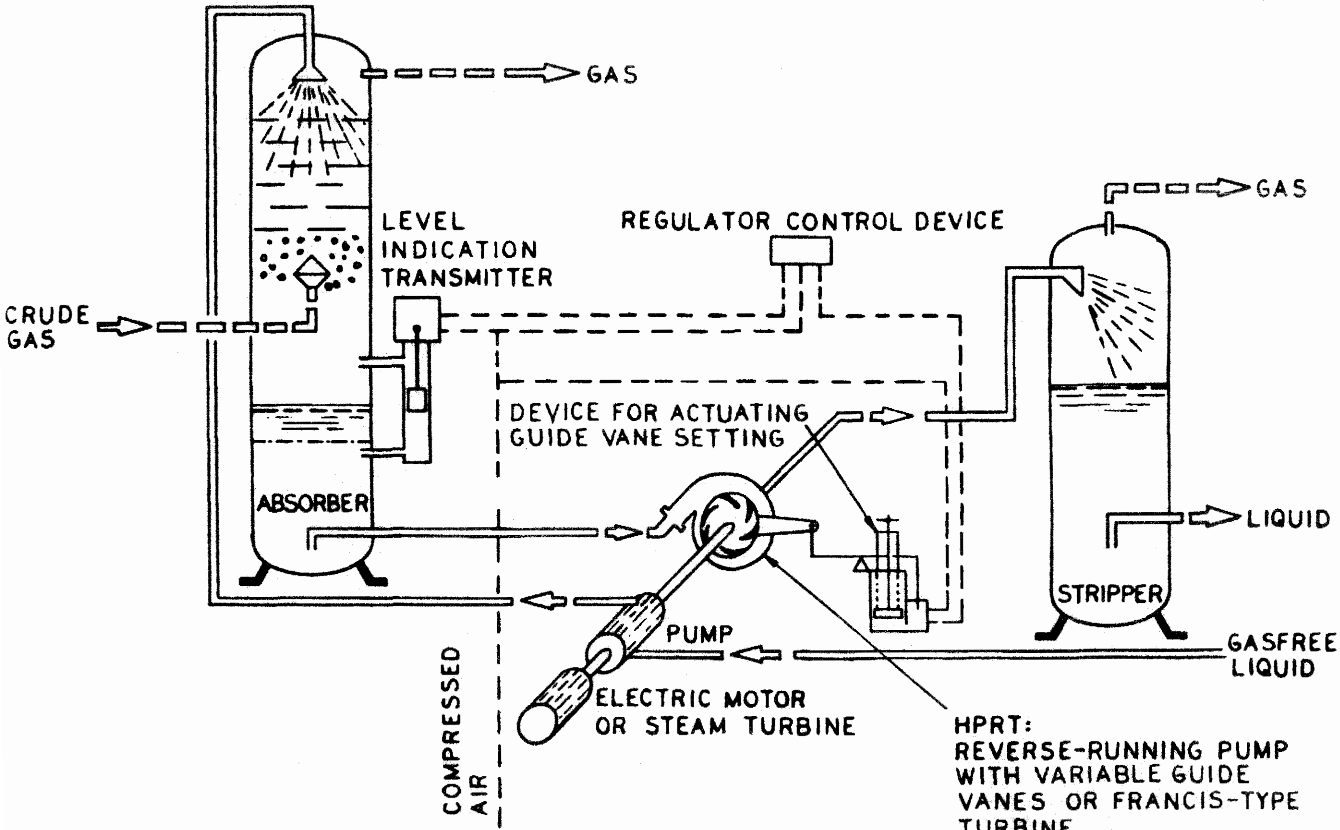

Figure 14-21. Flow diagram of hydraulic energy recovery using an HPRT with variable guide vanes or Francis-type turbine (from Franzke).

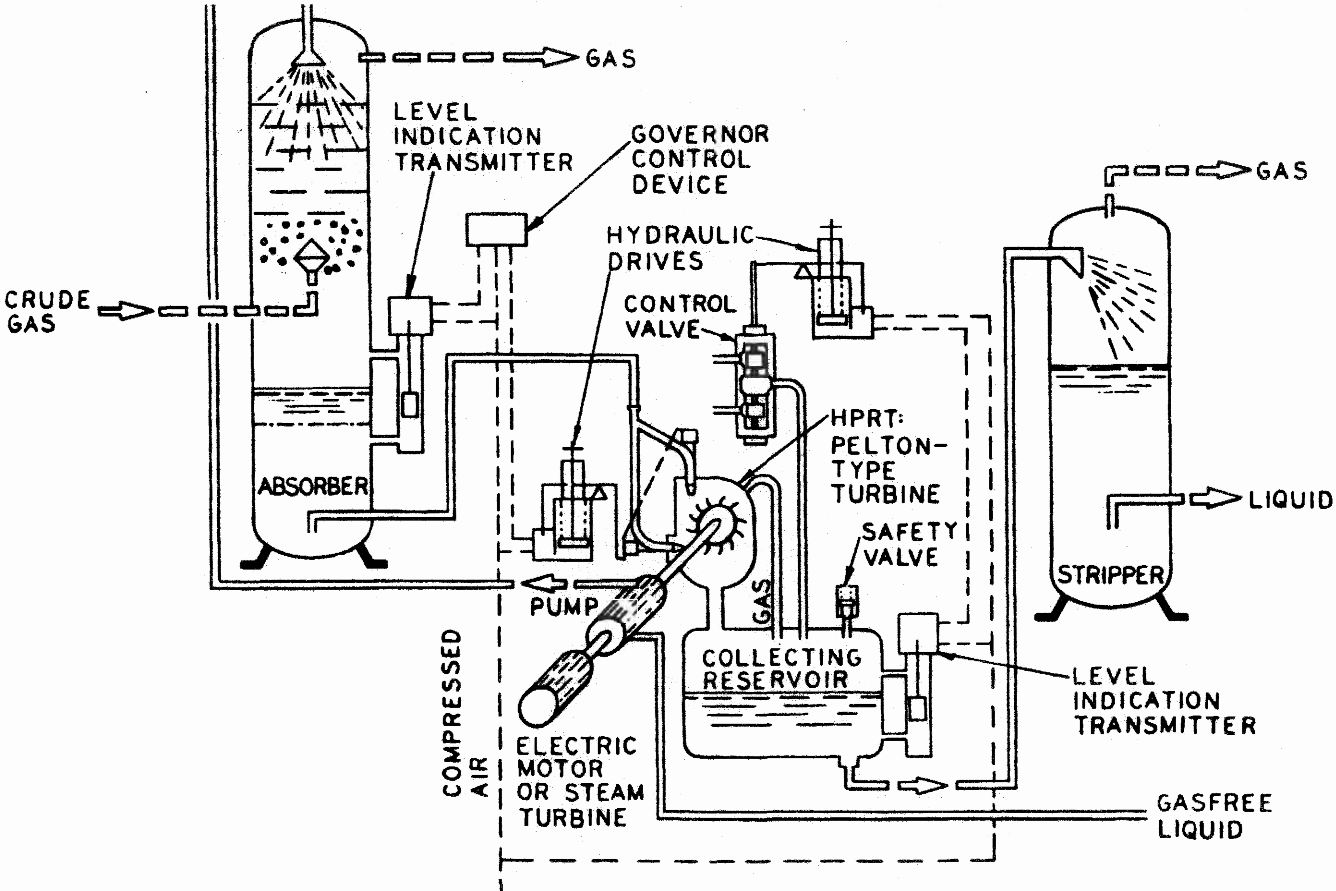

Figure 14-22. Flow diagram of hydraulic energy recovery using a Pelton-type HPRT (from Franzke).

age systems. Hydraulic turbines in power recovery applications may be used to drive a pump, compressor, or other types of rotating equipment either as a sole driver or as a helper driver in tandem with another driver such as an electric motor or steam turbine. Hydraulic turbines may also be used to drive electric generators.

When the hydraulic turbine is used in tandem with another driver to drive a pump, consideration must be given to the available starting load requirements and operating load conditions. If the hydraulic turbine is able to bring the pump up to a speed with a reduced flow capacity through the pump, such as at pump minimum flow where the required HP is less, it is possible to use a reduced size electric motor or steam turbine driver to make up the horsepower difference required for normal pump operating conditions. This is not usually done, however, because plant operating conditions may cause an upset in the flow capacity to the hydraulic turbine with a resultant potential overload on the partial-sized drivers; the pump system would malfunction. The driver used in conjunction with the hydraulic turbine is usually full sized to run the pump by itself and in addition to accommodate the low flow input horsepower requirements for the hydraulic turbine.

On tandem-drive pump units, an over-running automatic free-wheeling clutch is often used that will permit the hydraulic turbine to be disengaged from the drive operation for simplified start-up procedures, system operating upsets, and maintenance. The use of the over-running clutch will also permit a lower flow capacity to the hydraulic turbine when it is operating at minimum flow conditions.

The arrangements of the drive train components for tandem-drive units depend on the disassembly requirements for the components.

When an electric motor is used in conjunction with a hydraulic turbine in tandem-drive arrangements, a double-extended motor shaft with the pump on one end and the turbine on the other end, is most common. An over-running clutch may be used between the motor and the hydraulic turbine when desired. The full-sized motor acts as an excellent speed governor for the hydraulic turbine. The motor may be essentially idle or it may even function as an electric generator with no adverse effects on the electric utility system should the RPM reach or slightly exceed synchronous speed.

When a steam turbine is used in conjunction with a hydraulic turbine in tandem arrangements, the pump is typically installed between the steam turbine and the hydraulic turbine since the steam turbine is usually not available with a double extended shaft.

A steam turbine is capable of acting as a good governor for speed regulation, provided the hydraulic turbine power rating does not significantly exceed that required by the pump (or other driven equipment). Power recovery may be realized by using a hydraulic turbine to drive an electric generator of either the synchronous or induction type. For the smaller systems, the induction-type generator is attractive for economic considerations.

A squirrel-cage induction machine becomes an excellent power generator when it is excited by AC power while the shaft is rotated above synchronous speed. Frequency of the generated power is that of the excitation; shaft speed determines only the amount of power consumed or delivered. If the shaft is rotated much faster than synchronous speed, the machine can burn out. But the system tends to be self-regulating because the shaft becomes increasingly harder to rotate as speed increases above synchronous.

When the induction machine is excited by AC power from a utility system, power is fed back into the power grid as the speed reaches and surpasses rated synchronous speed. The power grid provides the excitation voltage needed by the induction machine for both motor and generating action.

When an induction generator must work without a source of AC power, excitation can be supplied by residual magnetism and capacitors connected phase to phase. A storage battery can be used to provide a current pulse through one of the windings and thus leave sufficient flux to start generation.

Generation occurs when the capacitor current exceeds the excitation current of the windings. Generation stops when the shaft speed is lowered to the point where capacitive reactance exceeds that of the winding or when the load absorbs too large a portion of the capacitor current.

Operation and Control Equipment

As the flow through the HPRT increases from the no-flow condition, the fluid velocity through the runner gradually imparts to the runner not only enough energy to overcome internal friction but also to permit some net power output. This point usually occurs at about 40% of design flow or capacity. As in any turbine driver, the machine will speed up until the load imposed on the shaft coupling equals the power developed by the turbine. The hydraulic turbine must operate to satisfy its own head-capacity-speed-horsepower relationship within the available head and imposed speed limits.

Consider a power recovery turbine operating as the only driver. If more liquid is allowed to flow to the power recovery turbine than is needed to produce the horsepower required, the turbine will speed up and try to handle the liquid; at the same time the driven pump or compressor will speed up. In speeding up, the turbine will produce more shaft horsepower, which the driven pump or compressor must absorb at the new speed. Finally, the horsepower will be balanced, but the speed of the driven unit may be off design.

If speed control is necessary, throttling some of the turbine's driving fluid across a valve bypassing the turbine allows it to satisfy the horsepower-capacity-speed requirements of the driven unit. If the amount of fluid available to the turbine is less than that needed for the design conditions of the driven unit, the turbine will slow down and try to shed some of the load. Here speed control can be achieved by throttling the available pressure so that the turbine sees only that portion of the available head needed to satisfy its head-capacity-speed relationship at the desired speed.

When a power recovery turbine is combined with a makeup driver, except at a single point, the recovery turbine always requires either flow bypassing or inlet pressure throttling. The balance point is always determined by the power-speed characteristics of the driven unit. If the driven unit can use all the generated horsepower, such as a floating electric generator would, capacity control and pressure throttling may not be needed. When a speed-controlling, variable-horsepower helper driver such as an electric motor or steam turbine is used it will hold the speed constant and make up just enough horsepower to permit the power recovery pump turbine to satisfy its head-capacity curve at virtually any flow rate.

Split-range liquid-level controllers are typically used to regulate the available flow to HPRT's. The split-range liquid-level controllers and pressure-control valves are usually furnished and installed by the purchaser. The pressure-control valve is usually located at the inlet side of the hydraulic turbine to prevent an excess pressure condition from occurring at the turbine shaft seals by a closed valve. Also, the low shaft sealing pressure usually results in a lower initial cost and reduced maintenance. The signal from the controller is used to adjust the pressure-control valve when too much head is available for the capacity and speed and to bypass excess capacity from the system when more liquid is available to the HPRT than needed to satisfy the relationship. When an HPRT is provided with adjustable guide vane nozzles for performance variation, a proportional range controller will provide the operator signal to appropriately adjust the guide vane setting for optimum conditions.

An over-speed trip device is often furnished with the hydraulic turbine. This device is typically used to provide a signal to operate an over-speed alarm or to close the pressure-control valve for minimum flow turbine operation. The sensing device may be a pneumatic or electronic transmitter or a mechanical trip mechanism installed to sense the turbine shaft speed.

If the hydraulic turbine should operate at runaway conditions (zero torque) due to no load, the turbine shaft speed will generally increase to within the range of 120% to 155% of the normal design speed with 100% normal design head. The overspeed amount depends on the specific speed characteristics of the machine. Should an upset condition occur where there is a large amount of vapor present with a loss of liquid level and with full differential pressure across the turbine, a very high runaway speed could occur. This is due to the low-density vapor producing a high differential head and a high-volume flow.

HPRT's should be brought up to full operating speed as rapidly as possible, because they not only fail to generate power but actually consume power until they attain about 40% of the design capacity.

The installation of the previously mentioned over-running automatic free-wheeling clutch between turbine and the driven pump or compressor is a good solution. The to-be-driven machine does not have to turn until fluid is available to the HPRT, which is not connected to the to-be-driven unit until it tries to run faster and puts out power. Using this arrangement, the start-up sequence can be selected so that the HPRT goes from zero speed to full operating speed along the zero torque curve.

Conclusion

In view of the significant power savings possible by use of power recovery turbines, energy users should take advantage of every opportunity to investigate the economics involved. Justification is based on the value of the energy saved during a projected life of the turbine versus the projected cost of purchasing, installing, and maintaining the machine for the same period of time.

The effects of changes to the operating conditions, such as available flow capacities and differential pressures for the HPRT's and driven machines need to be considered. Since the most commonly used turbine types have fixed performances, changes to the operating conditions may cause a significant change to the power output from the turbine unless modifications to the turbine internal nozzle sizes are made. HPRT's with internally or externally adjustable guide vane assemblies are desirable when changes to performance characteristics are expected.

Another consideration for selecting a hydraulic turbine as a driver in place of an electric motor or steam turbine is the fact that the hydraulic turbine does not have the incremental costs in energy. Experience with HPRT's in actual operating installations shows that these machines are very reliable, they perform the design requirements, and the operating costs are minimal. The hydraulic and mechanical performances are readily predictable.

Current inquiries show that there is a significant potential for hydraulic turbines during the eighties and surely beyond. Indications are that sizes much larger than those currently in use will be needed. Also, the types in demand will include the adjustable guide vane nozzles, diagonal flow types, vertical types, and possibly the combination turbomachine with the turbine and pump unit in the same case.

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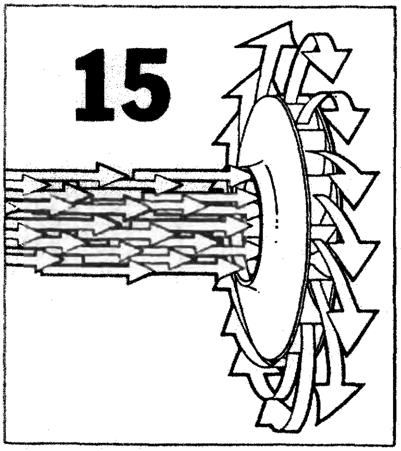


15 Chemical Pumps Metallic and Nonmetallic

by **Frederic W. Buse**
Ingersoll-Rand Company

Chemical pumps are designed for many processes and products that are not normally handled by pumps designed for a single product or process such as general water pumps, boiler feed pumps, cooling water pumps, or petroleum industry pumps. The chemical processes vary from acids, alkalies, toxics, reducing agents, oxides, slurries, organics, or inorganics causing corrosion, erosion, galvanic action, or leaching to occur on the pumps and piping system and any other product in the process.

To handle this variety of conditions, the pumps employ various materials such as 316 stainless steel, ductile iron, alloy 20, titanium, Hastelloy B and C. The continuous development of nonmetallics also make these pumps available in vinyl esters, epoxies, PVC, or with linings of teflon. Some pumps employ carbon, ceramic, and glass bodies or linings.

ANSI Pumps

Specifications

Most chemical pumps in the United States in the past 25 years have been developed according to the ANSI B73.1M and .2M specifications for horizontal and vertical pumps respectively (Figures 15-1 and 15-2). These specifications were initially developed in 1955 and were published in 1962. The current specifications were published in 1991. Besides safety criteria, the main objective for these specifications was to establish dimensional standards and interchangeability of various size pumps

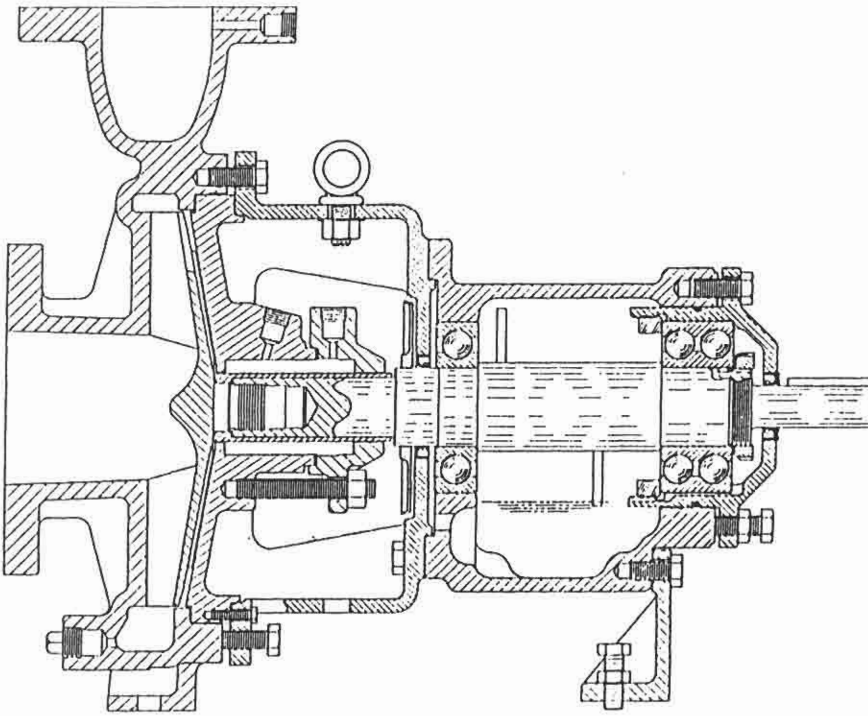


Figure 15-1. ANSI overhung single-stage pump (courtesy of Ingersoll-Rand Company).

within a given envelope (Figure 15-3A). The ANSI B73.1M specification has a dimensional designation of AA to A120 that covers 19 various size pumps. Its dimensional standards not only cover the pump itself, but also cover the pumps on bedplates. ANSI B73.2M has a dimensional designation of 2015 to 6040 that covers 15 various size pumps (Figure 15-3B). This dimensional standard became an important criterion for chemical plant designers because they could rely on the pump envelopes for dimensional accuracy when laying out the piping and foundations for the pumps. This eliminated the need for certified drawings of the pump assembly or pump bedplate combination from the pump suppliers. It also eliminated the need for extra inventory for spare parts because spare pumps could be purchased from various pump manufacturers with the assurance that they would fit into an existing piping system.

The hydraulic range of these pumps at a synchronous speed of 3600 RPM is 2000 gallons per minute and over 800 feet. At 1800 synchronous speed, the range is from 3500 gallons per minute to 250 feet (Figures 15-4 and 15-5).

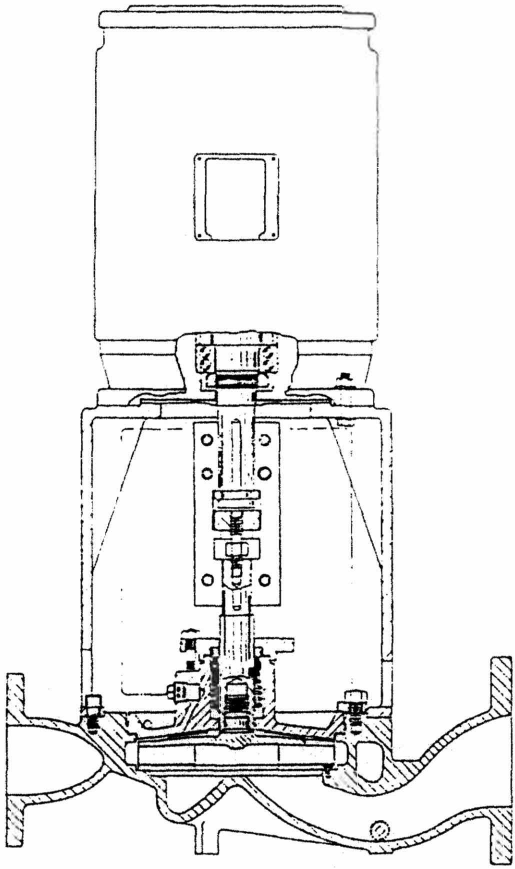
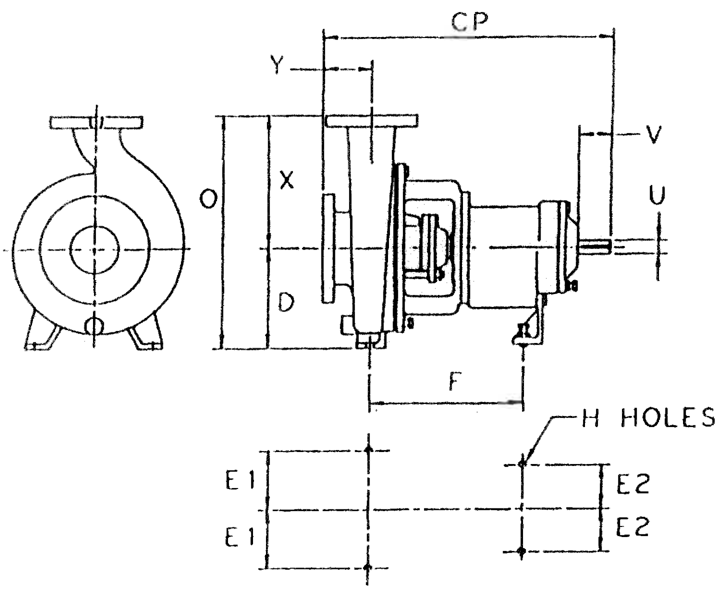


Figure 15-2. ANSI vertical in-line overhung single-stage pump with rigid coupling (courtesy of Ingersoll-Rand Company).

The specifications also stipulate that the pumps have centerline discharge casings and should be pulled from the rear rotor design that allows disassembly without disconnecting the suction or discharge nozzles. To maximize mechanical seal life, the specifications require a .005-inch shaft deflection limit at the impeller centerline due to dynamic deflection and a maximum full indicator run out at the stuffing box face of .002 inches.

The specifications also require that there be a minimum bearing life of 17,500 hours due to the defined maximum imposed hydraulic loads and that the suction and discharge flange pressure-temperature limits comply to a minimum of ANSI B16.5 Class 150 (Figures 15-6 and 15-7).



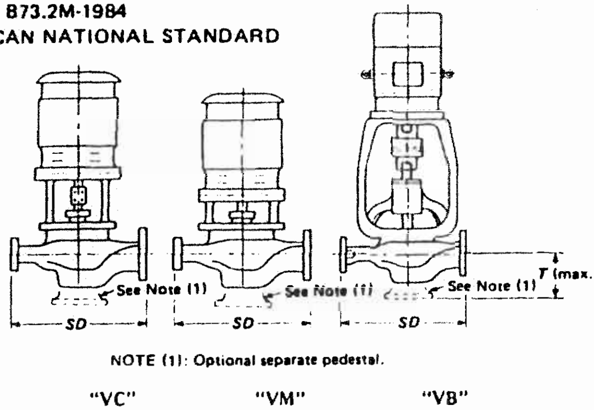
(Dimensions in Inches)

Dimension Designation	Size, Suction × Discharge × Nominal Impeller Diameter	CP	D	2E₁	2E₂	F	H	O	U [Note (1)] Diameter	U [Note (1)] Keyway	V Minimum	X	Y
AA	1½ × 1 × 6	17½	5¼	6	0	7¼	⅝	11¾	⅞	3/16 × 3/32	2	6½	4
AB	3 × 1½ × 6	17½	5¼	6	0	7¼	⅝	11¾	⅞	3/16 × 3/32	2	6½	4
A10	3 × 2 × 6	23½	8¼	9¾	7¼	12½	⅝	16½	1⅛	¼ × ⅛	2⅝	8¼	4
AA	1½ × 1 × 8	17½	5¼	6	0	7¼	⅝	11¾	⅞	3/16 × 3/32	2	6½	4
A50	3 × 1½ × 8	23½	8¼	9¾	7¼	12½	⅝	16¾	1⅛	¼ × ⅛	2⅝	8½	4
A60	3 × 2 × 8	23½	8¼	9¾	7¼	12½	⅝	17¾	1⅛	¼ × ⅛	2⅝	9½	4
A70	4 × 3 × 8	23½	8¼	9¾	7¼	12½	⅝	19¼	1⅛	¼ × ⅛	2⅝	11	4
A05	2 × 1 × 10	23½	8¼	9¾	7¼	12½	⅝	16¾	1⅛	¼ × ⅛	2⅝	8½	4
A50	3 × 1½ × 10	23½	8¼	9¾	7¼	12½	⅝	16¾	1⅛	¼ × ⅛	2⅝	8½	4
A60	3 × 2 × 10	23½	8¼	9¾	7¼	12½	⅝	17¾	1⅛	¼ × ⅛	2⅝	9½	4
A70	4 × 3 × 10	23½	8¼	9¾	7¼	12½	⅝	19¼	1⅛	¼ × ⅛	2⅝	11	4
A80	6 × 4 × 10	23½	10	9¾	7¼	12½	⅝	23½	1⅛	¼ × ⅛	2⅝	13½	4
A20	3 × 1½ × 13	23½	10	9¾	7¼	12½	⅝	20½	1⅛	¼ × ⅛	2⅝	10½	4
A30	3 × 2 × 13	23½	10	9¾	7¼	12½	⅝	21½	1⅛	¼ × ⅛	2⅝	11½	4
A40	4 × 3 × 13	23½	10	9¾	7¼	12½	⅝	22½	1⅛	¼ × ⅛	2⅝	12½	4
A80 (2)	6 × 4 × 13	23½	10	9¾	7¼	12½	⅝	23½	1⅛	¼ × ⅛	2⅝	13½	4
A90 (2)	8 × 6 × 13	33⅞	14½	16	9	18¾	⅞	30½	2⅜	⅝ × 5/16	4	16	6
A100 (2)	10 × 8 × 13	33⅞	14½	16	9	18¾	⅞	32½	2⅜	⅝ × 5/16	4	18	6
A110 (2)	8 × 6 × 15	33⅞	14½	16	9	18¾	⅞	32½	2⅜	⅝ × 5/16	4	18	6
A120 (2)	10 × 8 × 15	33⅞	14½	16	9	18¾	⅞	33½	2⅜	⅝ × 5/16	4	19	6

NOTES:
(1) *U* may be 1⅜ in. diameter in A05 through A80 sizes to accommodate high torque values.
(2) Suction connection may have tapped bolt holes.

Figure 15-3A. ANSI pump dimensions (from ASME B73.1M-1991 by permission of the American Society of Mechanical Engineers).

ANSI/ASME B73.2M-1984
AN AMERICAN NATIONAL STANDARD



VERTICAL IN-LINE
CENTRIFUGAL PUMPS FOR CHEMICAL PROCESS

DIMENSIONS, in.

Standard Pump Designation[1]	ANSI 125, 150, 250, or 300 Flange Sizes		SD +0.10 −0.08	T (maximum)
	Suction	Discharge		
VC, VB, VM				
2015/15	2	1½	14.96	
2015/17	2	1½	16.93	6.89
2015/19	2	1½	18.90	
3015/15	3	1½	14.96	
3015/19	3	1½	18.90	7.87
3015/24	3	1½	24.02	
3020/17	3	2	16.93	
3020/20	3	2	20.08	7.87
3020/24	3	2	24.02	
4030/22	4	3	22.05	
4030/25	4	3	25.00	8.86
4030/28	4	3	27.95	
6040/24	6	4	24.02	
6040/28	6	4	27.95	9.84
6040/30	6	4	29.92	

NOTE:
(1) Pump Designation: defines design, flange sizes, and *SD* dimension (e.g., VC, VB 50-40-380).

Figure 15-3B. ANSI pump dimensions (from ASME B73.2M-1991 by permission of the American Society of Mechanical Engineers).

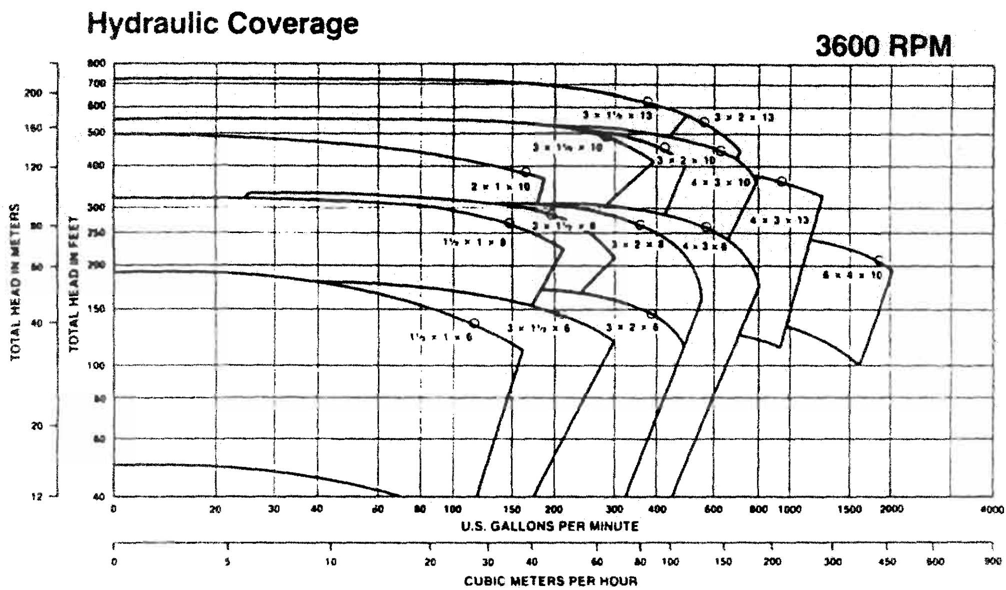


Figure 15-4. Typical 60 cycle-3600 rpm performance chart for ANSI B73.1 pumps (courtesy of Ingersoll-Rand Company).

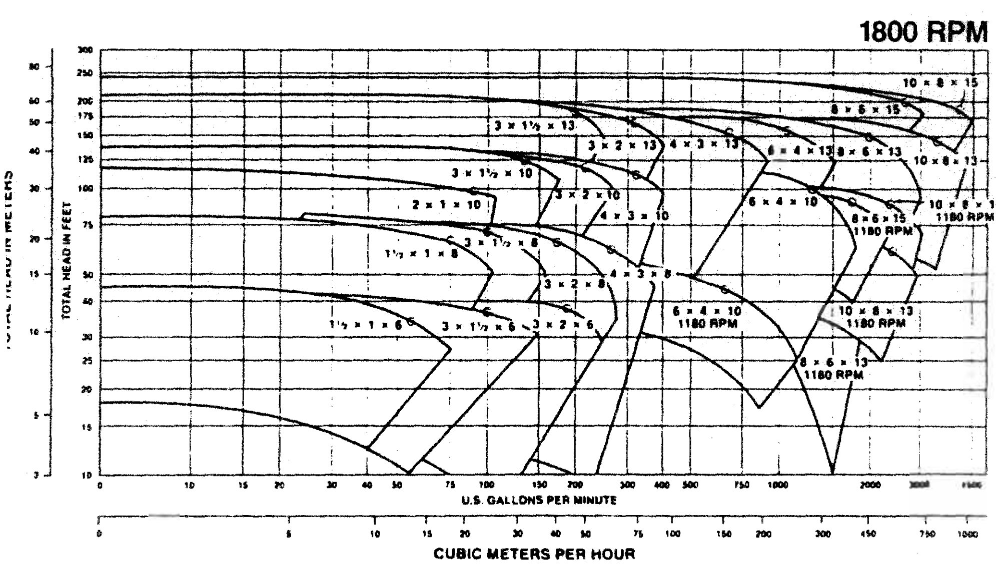


Figure 15-5. Typical 60 cycle-1800 rpm performance chart for ANSI B73.1 pumps (courtesy of Ingersoll-Rand Company).

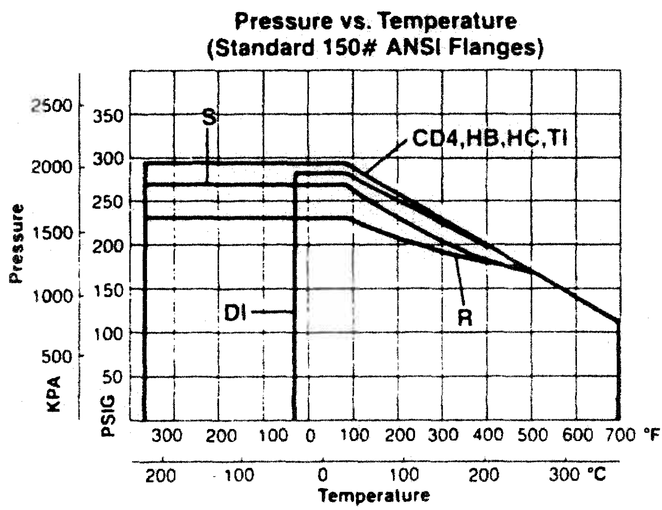


Figure 15-6. Pressure versus temperature for 150 pound ANSI flange.

Material Specifications	
DI	Ductile Iron
S	316 Stainless Steel
R	Alloy 20
CD4	CD4MCU
HB	Hastelloy B
HC	Hastelloy C
TI	Titanium

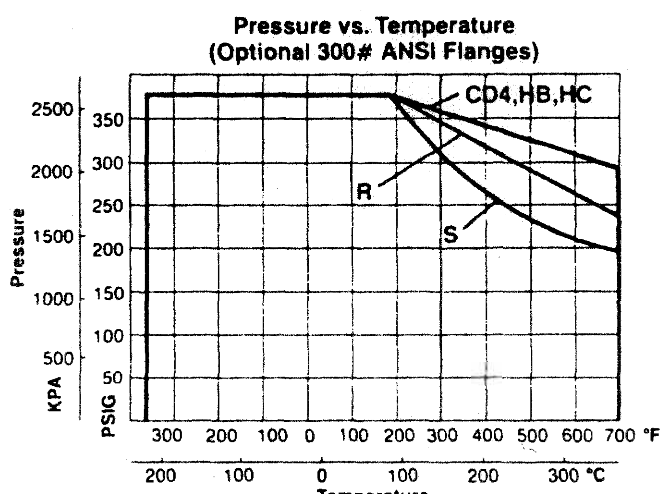


Figure 15-7. Pressure versus temperature for 300 pound ANSI flange.

The impellers employed by most manufacturers are semi-open even though the specifications allow for both semi-open and closed impellers. The specifications call for the wetted-end materials to be manufactured from alloy steels, carbon steel, ductile iron, or cast iron. However, most manufacturers stock ASTM A744 (similar to Type 316) or ASME A395 (cast ductile iron).

ANSI B.73.1M is for horizontal cradle pumps. Most pump manufacturers divide the 19 pump sizes into three groups. Many of the parts are interchangeable.

ANSI B.73.2M covers the same hydraulic range up to the 6 in. suction × 4 in. discharge nozzle size and consists of a total of 15 pump sizes. The vertical in-line pumps are designed so a user can obtain access to the impeller and stuffing box area without disassembly of the pump from the line nor disassembly of the motor. This is accomplished by use of a rigid coupling and/or a separate bearing housing that fits between the casing and the motor.

These pumps have the same deflection and total indicator runouts as the horizontal pumps. When using a rigid coupling, these pumps employ a P-face motor. This motor is called an in-line motor (NEMA MGI-18.620). It was developed in conjunction with NEMA and employs double-row or back-to-back deep groove thrust bearings to absorb the radial and axial thrust developed by the pump. The construction of this motor is such that for a radial load of 25 lbs. at the end of the motor shaft the radial deflection shall be no more than .001 in. with an axial load of 50 lbs., the shaft movement is limited to .0015 in.

When a separate bearing housing design is employed for the vertical pumps, a C-face motor is used (Figure 15-8). Because the hydraulic thrust is absorbed by the bearing housing's bearings, the standard C-face motors do not require special thrust bearings. A flexible coupling is used between the bearing housing and the motor. The advantage of the bearing housing design on the larger size impellers (over 10 in.) and the larger size motors (used on 3 in. discharge nozzle and above) is that the pump shaft system is more rigid and deflection is less because of the smaller overhang. A disadvantage of the design is the problem of removing the extra weight of the assembly from the casing and out of the support head area without causing damage to the parts being removed.

General Construction

Impeller

Semi-open impellers develop higher axial thrust loads than do closed impellers. However, with chemical pumps, the semi-open type impellers are normally employed to facilitate cleaning of fibers or particles often contained in the process liquid.

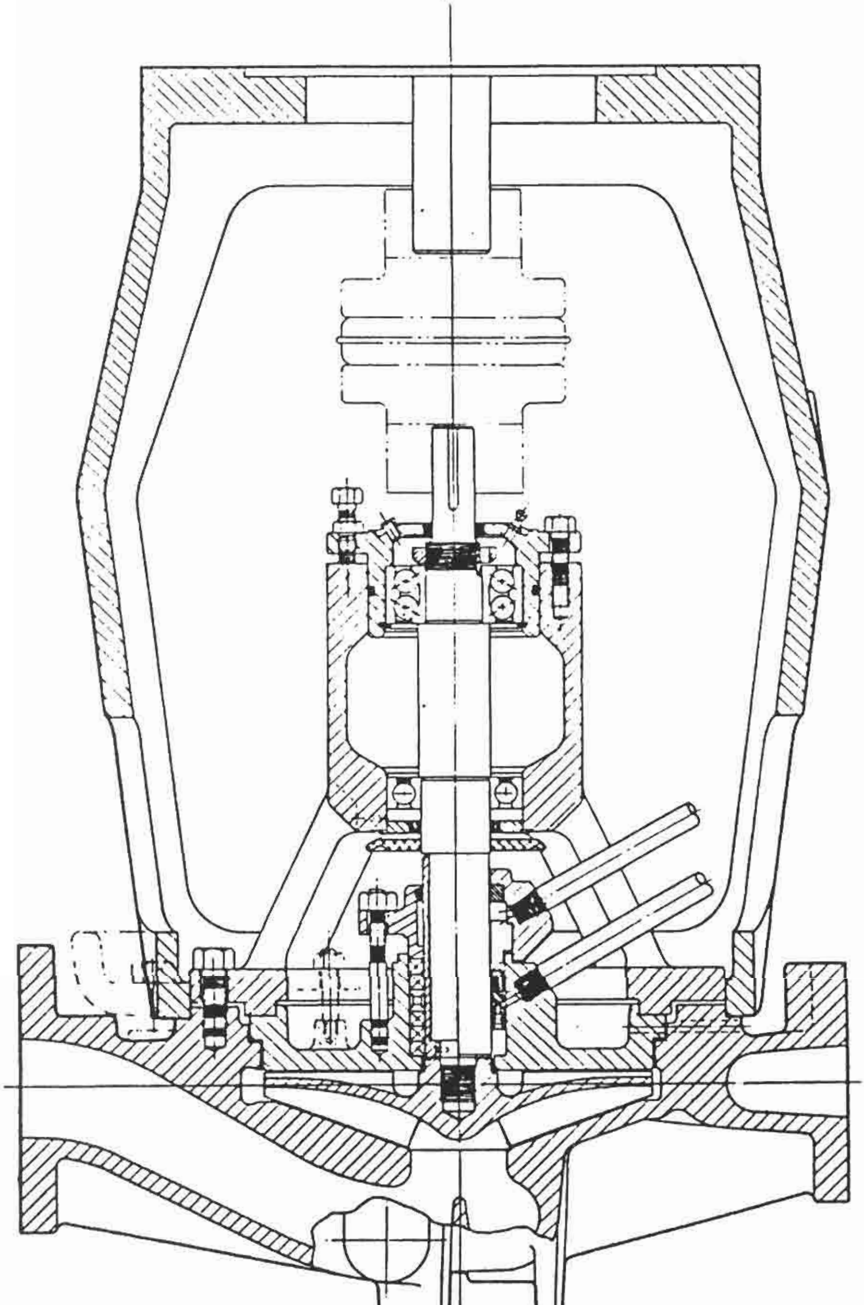


Figure 15-8. ANSI vertical in-line overhung single-stage pump with bearing housing (courtesy of Goulds Pumps, Inc.).

There are three basic types of back shroud configurations. One is a full-open impeller where the back shroud is almost completely scalloped out to reduce the area on which the hydraulic pressure can react thereby almost eliminating axial thrust (Figure 15-9A). The second is a semi-open impeller that has a partially scalloped back shroud (Figure 15-9B) that has greater axial thrust than the full-open impeller but has better efficiency and head characteristics. The third is the full back shroud (Figure 15-9C) that normally has about five points higher efficiency than the scalloped impeller but has less head than the scalloped impeller because of the regenerative action of the scallop. Most open-impeller designs are of the scallop or full shroud variety. Full-open impellers are rarely used in this industry because of low efficiency and the bending loads on the vanes. If it is found that impellers with plain back surfaces produce inadequate bearing life due to excess axial thrust, then pump-out vanes are usually employed on the back of the shroud to reduce the thrust. (Refer to Chapter 18).

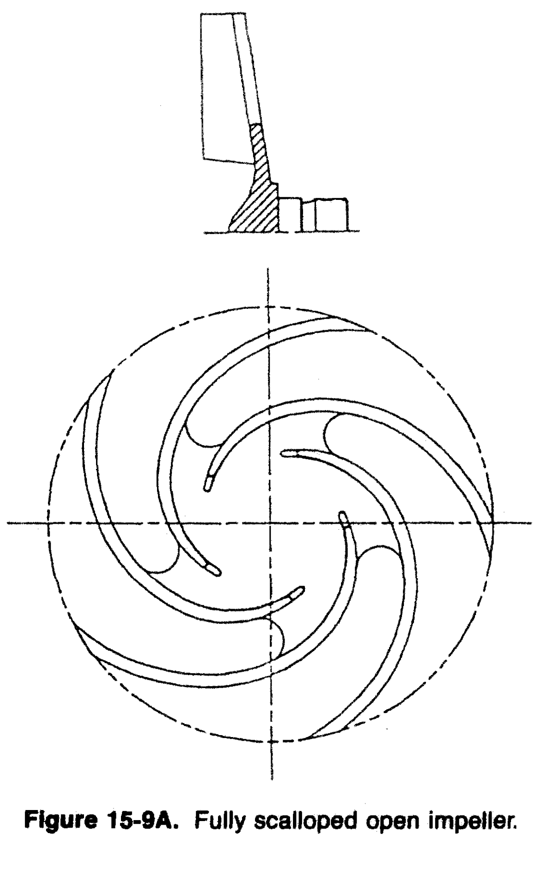


Figure 15-9A. Fully scalloped open impeller.

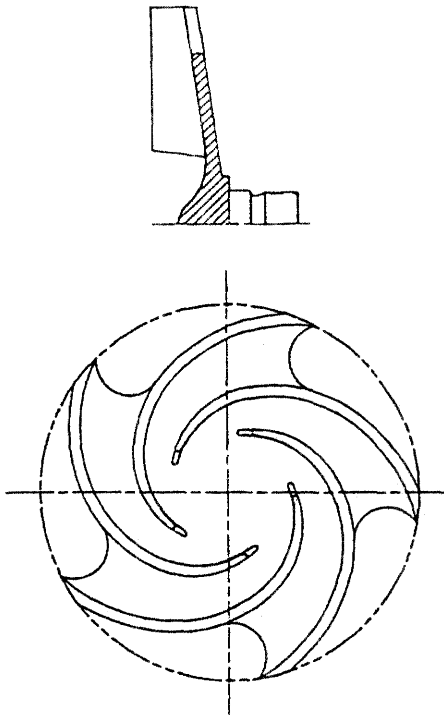


Figure 15-9B. Partially scalloped open impeller.

Casings

Casings for the ANSI chemical pumps have centerline discharge and suction both in the horizontal and vertical pumps. This makes it easier for laying out the piping in a system as well as reducing the nozzle loading. This is because the centerline nozzle eliminates the moment arm from the centerline of the casing to the centerline of the nozzle that exists with tangential discharge (Figures 15-10A and 15-10B).

On the horizontal casing, the centerline discharge results in a cutwater being approximately 30° off the centerline allowing the casing to be self venting. The casings are designed so that the rotor can be removed from the back without disturbing the suction and discharge piping. The gaskets of the casing are atmospheric confined so that the internal pressure cannot push the gasket out, as could occur with a full, flatface gasket. The flanges are 150 ASME flatface with an option of a raised face for steel and alloy material casings. There is the option of 300 lb. flanges for both the suction and the discharge on the steel and alloy casings. In the chemi-

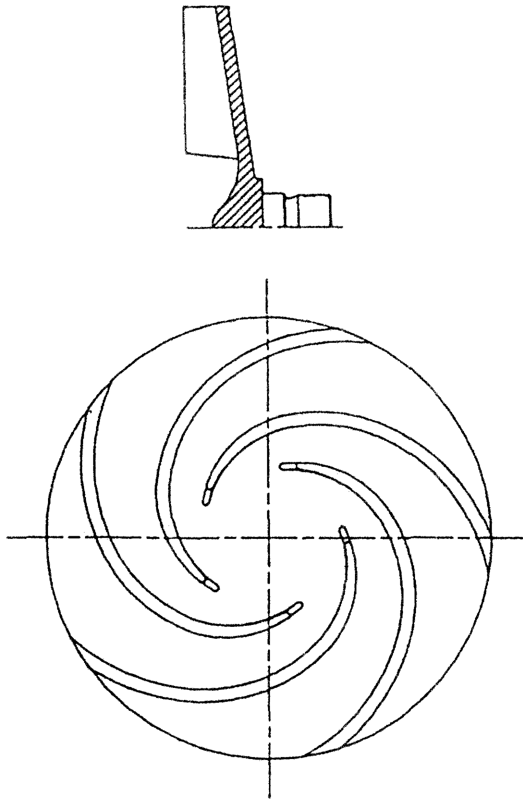


Figure 15-9C. Full back shroud open impeller.

cal industry, to prevent localized erosion-corrosion many customers do not want any holes in the casings; therefore, vents and drains are offered as options.

The running surface of the casing that is adjacent to the front face of a semi-open impeller is designed so that the clearance between the impeller and case ranges from 0.010 to 0.020 inches depending on the manufacturer, pump size, and material. When this surface is machined on an angle relative to the centerline of the pump, the surface has to be concentric with the centerline within .0010 inches. If quality control is not adhered to, the wear surface will have wider clearance on one side relative to the other resulting in inconsistent performance. The surface has to be machined on an angle of plus or minus 3 minutes of a degree to maintain performance. Instead of putting this surface on an angle, some manufacturers machine it so it is perpendicular to the centerline of the casing.

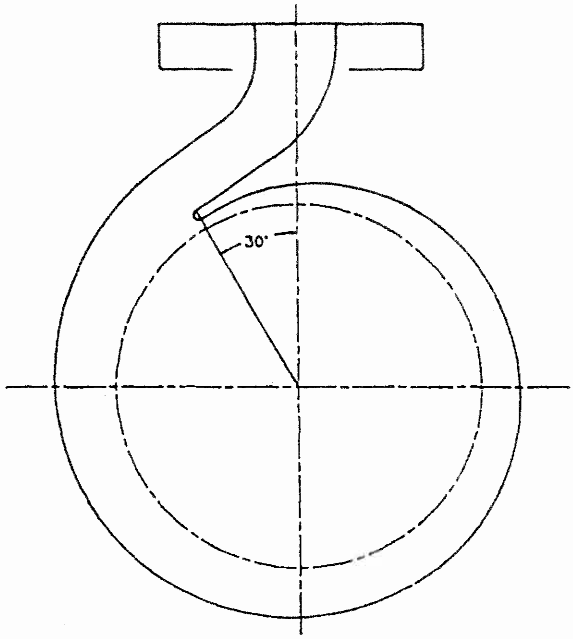


Figure 15-10A. Casing volute with centerline discharge nozzle.

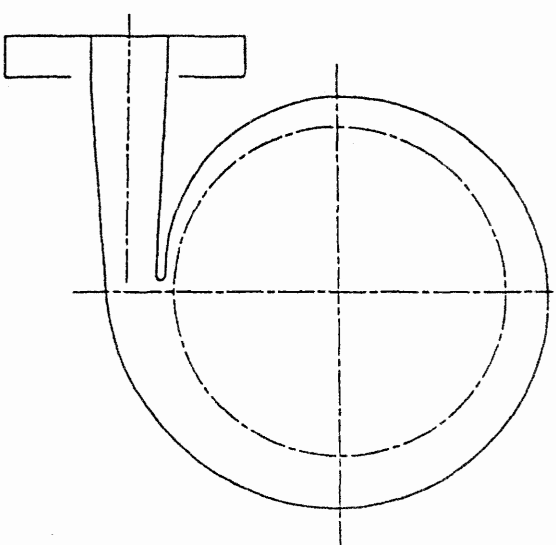


Figure 15-10B. Casing volute with tangential discharge nozzle.

This makes it easier to manufacture and eliminates the problem of the angle machining. This usually results in the back wall of the impeller being cast on an angle. Studies have been made comparing angles of 0° to 8° in the forward position relative to the back position to determine if there was any difference in performance, efficiency, and NPSH. The investigation showed no difference in the various performance criteria.

Volute

The design of the volute for hydraulics depends upon the stiffness of the shaft system. Most of the systems can absorb the radial thrust developed by a single volute. When the bearing loading becomes excessive, a full double volute can be employed. The radial thrust developed by the double volute is about 16% that of a single volute resulting in longer bearing life. A partial splitter, which is between a single and double volute, reduces the thrust to approximately 33% of the single volute and results in bearing life and deflection within the parameters of ANSI specifications. The length of the splitter is dependent on the specific speed. A partial splitter is easier to cast because its core support is not as long and is easier to remove after pouring. On specific speeds of 500 or less, circular volutes are sometimes employed. Typical volute designs and the method used to calculate radial load are described in Chapter 5.

Gasketing

The gasketing is usually a flat semi-confined design or an O-ring design. Flat gaskets take more bolting because the bolt load must compress the gasket as well as resist the hydraulic force. With the O-ring only the hydraulic force acts at the centerline of the O-ring.

Flat Gasket

The flat uncompressed gasket is 1/32 inch to 1/16 inch in thickness and has 27% compression due to the bolt force. For many years it was made from asbestos which was universal for most chemicals; however, with the changeover to nitrile synthetic material, two different types of base materials are required to handle the spectrum to which chemical pumps are applied. The new material has the same compression rate and hardness as the asbestos materials.

O-Rings

O-rings can be fitted radially, axially, or in a corner. Radial O-rings require more control of the machining of the concentricity of the casing and the casing cover relative to the axial O-ring. Because the axial O-ring has a greater sealing diameter, it requires additional bolting. Corner O-rings take more tolerancing but are a good compromise between radial and axial machining. The O-rings are made out of EP (ethylene propylene) for hot water, buna for hydrocarbons, viton for general chemicals, and Kalrez for highly corrosive chemicals. They are usually color coded to designate materials.

Casing Covers

Stuffing Box

The casing cover, sometimes called a stuffing box extension, encloses the back end of the casing. The casing cover also includes the stuffing box or seal chamber. Originally, the ANSI standards required that the minimum stuffing box packing size be 5/16 in., 3/8 in., or 7/16 in. depending on pump size. The stuffing box was designed to handle both mechanical seals and packing; however, through years of experience, it was found that the mechanical seal's outside diameter had too small clearance between it and the bore of the box. This limited the amount of cooling that a seal could obtain, especially in double seals. So even though there was cooling injection into the gland and out of the box for double mechanical seals, there were frequent failures at the outboard seal. As a result, the specification includes optional large bores that only accommodate mechanical seals. This should give adequate cooling of the box to increase the life of the seal. If a customer requires a box for both seals and packing, he will require a box to the original specification. The taps into the stuffing box may be 1/4 in. minimum, but 3/8 in. is the preferred NPT size.

Depending on fluid temperature, the option of a cooled or heated stuffing box is usually offered. The type of mechanical seals offered on these pumps are single seals, double seals, and tandem seals. Mechanical seals used in ANSI pump applications are discussed in detail in Chapter 17.

Frame

The frame for the horizontal pump is composed of the support head and bearing housing. Depending on pump size, this can be one integral component or two separate pieces. Some manufacturers refer to the bearing frame housing as the bearing housing. The bearing frame or housing consists of the housing, the shaft, bearings, bearing end cover, flinger, and feet. On pump sizes AA and AB, the feet are usually cast integral with the bearing housing support head combination.

Support Head

The support head is the member that aligns and fixes the casing to the bearing housing. ANSI requires the support head to be made of ductile iron or carbon steel. This requirement stems from concern that a system upset could subject the pump to excessive pressure and result in catastrophic failure of the cast iron support head. Support heads are also offered in stainless steel as an optional feature. This is done for three reasons:

- To reduce corrosion due to leakage from packing or mechanical seals.
- To reduce thermal conductivity in very high or very low temperature applications.
- To have a material that has high impact properties for temperatures below 40°F.

On vertical in-line pumps, the support heads are larger than horizontal pumps because they must allow the rotor to be passed out of the support head during disassembly and also must adequately support the weight of the vertical motor. When a bearing housing vertical in-line is employed, the support head is at least 50% higher in height than with the rigid coupling design. Depending on size and motor horsepower, the support heads are made out of cast iron, ductile iron, or fabricated steel. When the vertical support heads are made out of ductile iron or carbon steel, extra care has to be taken in machining the toleranced dimensions because of the release of residual stresses in these materials.

Bearing Housing

This is usually made out of cast iron. After machining, it is protected internally with a rust preventative such as a paint or clear material to prevent rust particles from forming internally in the housing during storage or shut down. The housing is designed to hold a reservoir of oil that is approximately a half pint on the small pumps, and 3 to 4½ pints on the large pumps. The housings have vents, drains, and a tap for an oiler. The vent and drain should be a minimum of ½ in. so the oil can readily flow during filling or draining. The vent should be designed in a way that water cannot enter into the housing. Sometimes the vent is made up of a pipe that comes up through the bottom of the housing; other times it is composed of a nipple and cap with a sixteen-hole or commercial vent. The oiler is located so that movement of the oil from the rotation of the shaft does not prevent the oil from entering into the housing. Oilers are supplied with a glass or plastic bubble; glass is specified in a refinery type of service.

Shaft

The shaft has to be designed to take the radial, axial, cyclic, and torsional loads. The axial load, depending on suction pressure, can be in either direction. Shaft diameters are rated by horsepower per 100 RPM. The shaft material can be 1020, heat treated 4140 or 316. In each case the material should be reviewed for imposed stresses. Refer to Chapter 16 for methods of calculations.

Impeller Attachment

On chemical service pumps, impellers are usually attached with a male or female threaded connection. Threaded connections are used between the impeller and shaft because they can be more readily sealed than when a key design is used, which inherently has an additional joint. Specifications do allow both types of connection.

Bearings

Specifications require that when the maximum hydraulic load from the largest impeller at a given speed is applied to the bearings, they will have a minimum L10 life of 17,500 hours. The designer and user should carefully select original or replacement bearings because the interchangeable dimensional envelope of an AFBMA bearing does not ensure that the size or number of balls are the same from one manufacturer to another, thereby resulting in a possible change in the rating of the bearing. It should also be remembered that the life of a particular design will change with suction pressure.

Most chemical pumps use semi-open impellers, and operate with a .010 to .020 inch gap between the impeller and casing wall. To maintain this clearance, the bearings should have .0015 to .002 of an inch axial input end play. Double-row bearings have an assembly end play of .002 to .003 of an inch. Back-to-back bearings have an assembly end play of .005. However, the back-to-back bearings are more forgiving to misalignment when radial loads are applied to the bearings. The life of a back-to-back bearing is about twice that of the comparable double-row bearing. Back-to-back bearings are usually offered as an optional feature on chemical pumps. This applies to both horizontal and vertical pumps.

Lubrication of the Bearings

The bearings are normally lubricated with either oil bath or grease. The oil level is usually at mid-ball. If it is higher than mid-ball, churning usually occurs resulting in foam in the bearing housing and an increase in temperature. Disk flingers are used to splash oil within the housing. Roll pins are also used on the shafts to splash oil. In this case, the initial oil level is below the balls.

With oil lube, the temperature on the outside skin of the housing is about 20° cooler than the temperature of the outer race. Skin temperature of the small pumps ranges between 110°F and 130°F, and on the large pumps between 140°F and 165°F. At skin temperatures above 185°F, the unit should be shut down and inspected to determine the cause of high temperature. It takes approximately 45 minutes to one hour for the temperature of a cradle to stabilize. With roll pin splash, the temperature is 20°F less than the oil bath.

Grease bearings are either replacement grease or seal-for-life bearings. The problem with the replaceable grease is that excess grease is usually put into the bearing's cavity causing sharp increase in temperature and drop in life.

The seal-for-life bearings come with either shields or seals. Shield bearings are adequate for the majority of applications. In general, the temperature of grease bearings is approximately 20°F less than the oil lubrication. The subject of bearing lubrication is discussed in detail in Chapter 20.

Mounting the Bearing

Finish and dimensions of the shaft should meet the bearing manufacturer's recommendations. Typically, the bearing is .000 to .0005 inches tight on the inner race and .0005 to .001 inch loose on the outer race. Refer to the bearing manufacturer's catalog for recommended fit. The bearings can be pressed on to the shaft, but it is usually better to heat them to prevent excess stress. They can be heated in an oven or put in an oil bath up to 240°F. When put in an oven, the bearings should be laid flat and should not touch each other. For field installation, the bearing can be set over a light bulb to expand the inner race. If the bearing is too loose on the outer race, the race will spin within its housing. The bearings are secured to the shaft with a snap ring or a bearing lock nut. The bearing lock nut is secured with a tab washer. This is preferred to a lock nut with a nylon type pellet. The outer race axial movement is restricted in two ways: either within a separate end cover or by having a snap ring in the outer race clamped between the bearing housing and the end cover.

Clamping Between the Housing and End Cover

With this type of a mount (Figure 15-11), there are fewer tolerances and fits to cause misalignment than when used in a separate end cover. However, to obtain axial adjustment, the end cover has to be removed to add additional shims to either side of the snap ring to obtain the impeller clearance. This will require a complete shut down of the pump and disassembly of the end cover.

End Cover Mount

The end cover is a separate piece that slides within the bearing housing. The end cover has an O-ring to prevent oil leakage from the housing. The fit between the end cover and the bearing housing bore is .001 to .002 inches. Expertise is required in machining of the end cover with automatic machines that can exert excessive jaw pressure. This will cause distortion resulting in incorrect bearing fits. The end cover has an oil return passage line that can be either cast or machined so oil can flow back into the reservoir.

The bearing is held in the end cover by a snap ring, a lock ring, or a solid ring (Figure 15-12A). When using snap or spiral flex rings, care has to be taken that the radius on the outer race of the bearing is maintained within the tolerance. If this radius is too large, the axial load will concentrate on the inner diameter of the snap or circle ring causing it to deflect in a cantilever action reducing the clearance between the impeller and the casing.

A solid ring is used to prevent this problem. The solid ring is either threaded into the bearing end cover or screwed on with a series of small bolts (Figure 15-12B). Axial adjustment of the impeller to casing is obtained by two sets of three bolts. One set of bolts is threaded into the end cover so that when they are turned they go against the bearing housing causing the end cover to move back toward the coupling. The other set is screwed into the bearing housing so when they are tightened and the others are loosened the end cover goes toward the impeller.

End Sealing

Ends of the bearing housings have to be sealed to prevent external liquids from entering into the housing as well as oil from leaking out. This

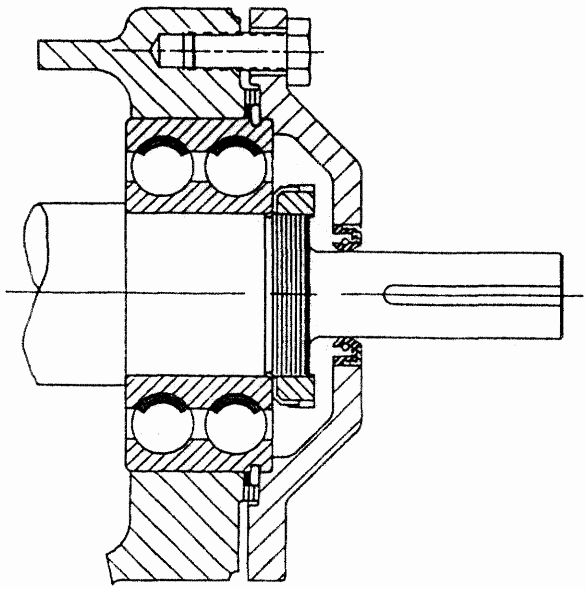


Figure 15-11. Thrust bearing positioned by a snap ring in the outer race.

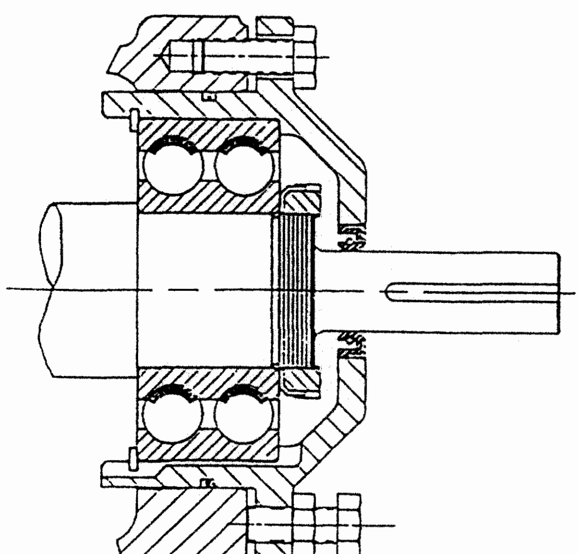


Figure 15-12A. Thrust bearing positioned with snap ring in the bore of an end cover.

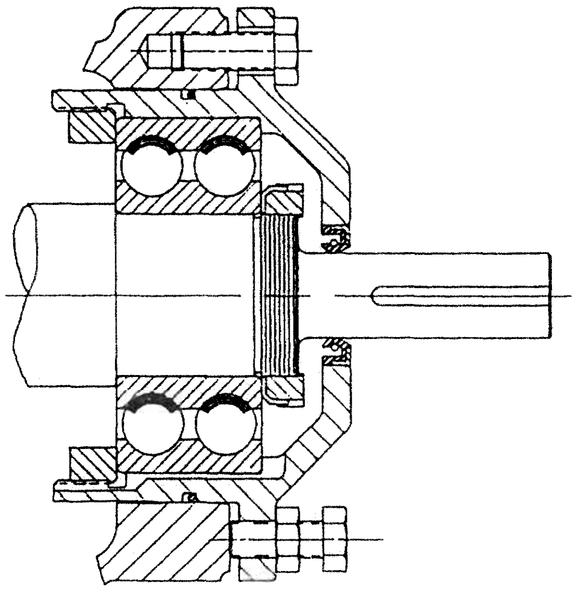


Figure 15-12B. Thrust bearing positioned with a solid ring in bore of an end cover.

is usually done with closures. Closure lips usually point out to prevent liquid from entering the housing especially during wash down. Closures can be supplied with garter-type springs or leaf-type springs. The garter-type spring is usually easier to assemble with this type of design. It is important that the finish of the shaft be within the manufacturer's recommendation, usually 16 RMS or less. A light film of castor oil should be applied to the closure for ease of assembly. There should be a 30° chamfer on the shaft for assembly of the closures over the shaft.

On the outer diameter of the closure, even though boring is correct and stamped casement is correct, the pieces are not always completely round. Therefore, it is recommended that some type of adhesive be put between the outer diameter of the closure and the housing to prevent leakage. When it is known that there is excess environmental water or vapor or contamination, magnetic-type closures or labyrinth-type closures are used. They can fit in place of the clipper-type of closure.

Bedplates

Standard Beds

Bedplate dimensions are dictated by ANSI specifications. Various types are used in the chemical industry, namely structural channel, bent plate, and castings. Bedplates are generally grouted for applications above 25 horsepower. For certain applications, drain rim channels or cast channels are offered.

Stilt-Mounted Beds

The stilt-mounted bed is a standard bed mounted on stilts 6 in. to 8 in. above the surface. In this way they can be washed off to obtain clean drainage in applications such as food and paper mills. The stilts should be a minimum of 1 inch in diameter and made of a stainless material. On some channel beds, in order to obtain secure footing (especially on a rough surface), three stilts are used instead of four—ending up with a milk stool type of construction.

Spring-Mounted Beds

This is a stilt-mounted bed with springs attached to the stilts that enable the entire assembly of pump, motor, and bed to move when external loads are applied to the nozzles. This is done in lieu of using piping expansion joints and loops.

Noncorrosive Beds

Noncorrosive beds are used in a corrosive atmosphere where it is known that steel beds will corrode in a short period of time. These beds are offered in the same sizes of the ANSI beds. They are made by epoxy coating steel beds or from nonmetallic. The nonmetallics are made from a form using resin transfer molding or are made as a solid mass, the thickness being that of the height of the bed. When grouting is required, the grout hole is usually cut with a saber saw or hole saw.

Flinger

The flinger is installed on the shaft with a press fit in front of the support head or cradle wall. The flinger is usually made out of elastomer or polymer. Its function is to prevent excess packing or seal leakage from entering into the bearing housing.

Other Types of Chemical Pumps

With continuous development of structural composite materials, pump manufacturers are offering various types of nonmetallic pumps in the ANSI envelope. ANSI specifications do not encompass nonmetallic pumps and there are no national standards for pressure, temperature, or limitations. These pumps can be horizontal cradle pumps, self-priming pumps, or submersible sump pumps. The design of these pumps will be discussed later in this chapter.

Sealless Pumps

Another type of chemical pump is referred to as a sealless pump, but more properly should be called a vapor tight or leakproof pump. These pumps have no mechanical seals; therefore, there are minimum risks due to seal failure or pump liquid being exposed to the atmosphere. Some manufacturers offer pumps up to 500 horsepower; however, the majority of vapor tight pumps are offered below 10 horsepower. Some companies manufacture the casing dimensional envelopes to be the same as the B73.1.

The two popular drives are magnetic drive and canned motor pumps (Figures 15-13 and 15-14). In both cases the normal limiting factors are the size of particles that can be pumped through their mechanism. The length of life of these types of designs depends on the type of bearings that are used. Excessive wear or seizure of the bearing results in downtime of the whole unit. Surveys of failures show that most are caused by flashing of the bearing lubricant rather than to the presence of particles. Bearings are sleeve journal and flat plate thrust or conical combination for journal and thrust. Monitors indicating bearing wear can be supplied to prevent catastrophic failures. The magnetic drive usually uses permanent magnets and can operate at higher temperature limits before cooling is required. The magnetic drive allows the use of a standard type motor to drive the magnets.

The canned motor pump has the advantage of being one complete unit for the pump and motor; thus, it is shorter than the magnetic drive. The outside liner of a canned motor pump is reinforced by the stator of the motor resulting in high allowable pressures. Maximum viscosity for this type of pump is 150 centipoise. This value can be much less depending on the size, torque, and speed of the equipment.

Initial capital expenditure of sealless pumps is higher than standard horizontal pumps with mechanical seals; however, manufacturers of this type of equipment suggest that this can be amortized in a short period because mechanical seal maintenance cost will be eliminated.

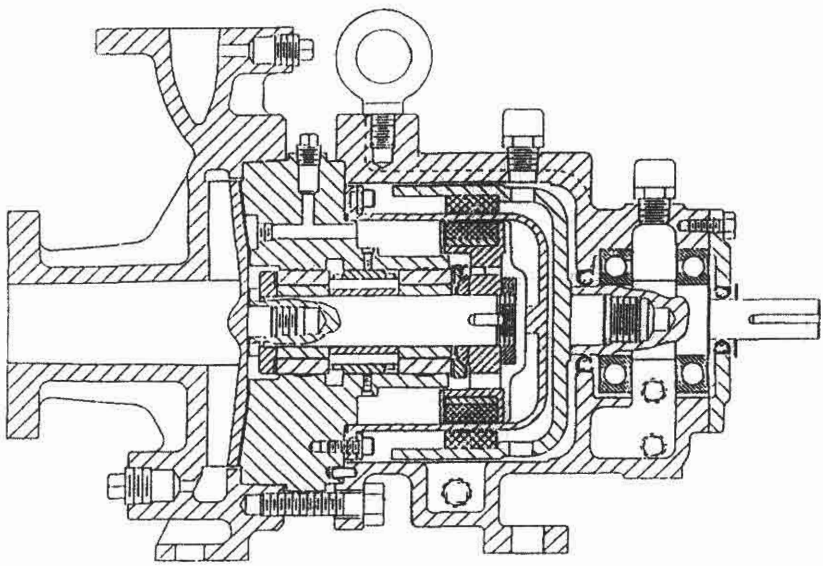


Figure 15-13. Overhung single-stage magnetic drive pump (courtesy of Ingersoll-Rand Company).

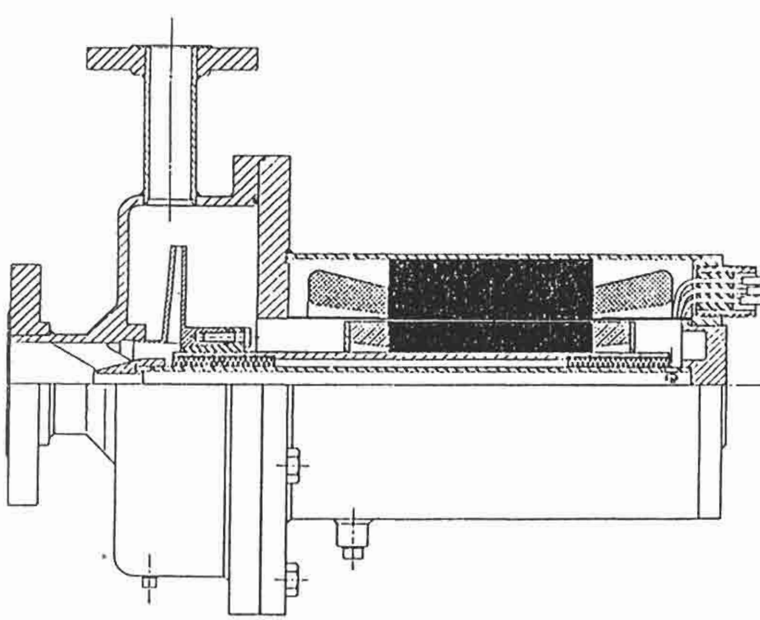


Figure 15-14. Overhung single-stage close-coupled canned motor pump (courtesy of Pacific Pumps Division of Dresser Industries).

Sump Pumps

Submersible or immersible sump pumps are a line of chemical pumps that have been derived from the combination of parts of the horizontal and vertical pumps (Figure 15-15). These pumps fit into wet sumps that may be 3 ft to 20 ft deep.

These pumps employ the casing and impeller of a horizontal pump and the support head, motors, and sometimes a casing cover of the vertical pump. To prevent critical speed frequencies, the shaft is supported by line bearings usually having a centerline to centerline distance of 5 ft for 1750 RPM and 3 ft for 3550 RPM. The column supporting the bearings and the discharge pipe is made of compatible material for the liquid in the sump. The bearings are either product lube, grease lube, or external lube which has been centrifuged or filtered. Typical applications are shown in Table 15-1.

The depth of the pump relative to the sump is called setting. Setting goes from the bottom of the strainer to the bottom of the mounting plate.

The sump pump is usually required at the mounting plate; therefore, when the impeller is selected, the frictional loss through the discharge pipe as well as the static head above the minimum liquid level has to be added to what is required at the mounting flange. These additional hydraulic losses may require a motor larger than would normally be used for the same selection as a horizontal pump. These pumps also require a minimum submergence to prevent vortexing or entrained air from entering into the suction (Figure 15-16).

There are no standards for the location in the sump of pumps of this type. Many users employ the suggested applications shown in the Hydraulic Institute Standards for sump design. The mounting plates for these pumps are usually plain carbon steel or carbon steel with an epoxy coating on one side. Sometimes stainless steel is used.

Sometimes it is desirable to pump the liquid to a level below the suction of the pump. A tailpipe is used to achieve this; however, the liquid level has to be above the impeller centerline when the pump is started (Figure 15-17). The tailpipe allows the liquid level to be pumped down to as much as 10 ft below the end of the flange of the suction pipe. The use of a tailpipe reduces the cost of the initial pump. The disadvantage, however, is that air can be pulled into the back of the casing thus reducing the overall performance of the pump.

Self Priming

The self-priming chemical pumps are also an offshoot of the horizontal pumps (Figure 15-18). These are usually available in 316 or ductile iron.

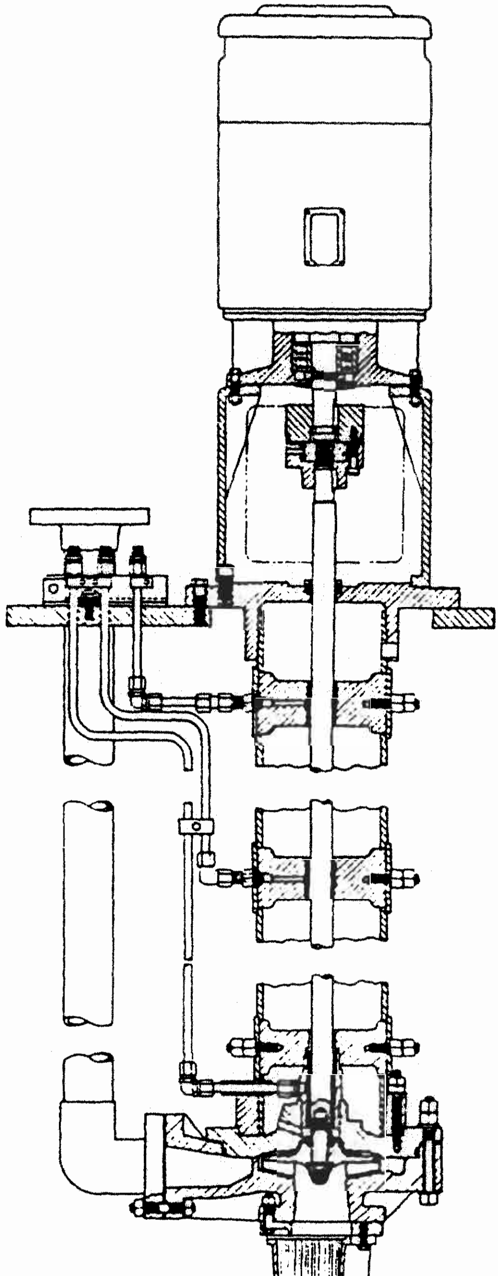


Figure 15-15. Vertical nonmetallic sump pump (courtesy of Ingersoll-Rand Company).

Table 15-1
Vertical Metallic Sump Pump Bearings

Bearing Material	Max Temp °F	Min Temp °F	Liquid	Shaft Mat'l
Bronze	180	−20	Water & compatible liquids	Carbon steel
Iron	180	−39	Water & compatible liquids	Carbon steel
Rubber	160	39	Abrasive with liquids compatible to rubber	316
Carbon	350	−65	Acids, chemicals hydrocarbons	316
Teflon grease lube	180	0	Chemicals	316
Teflon product lube	350	−100	Not compatible with teflon	316

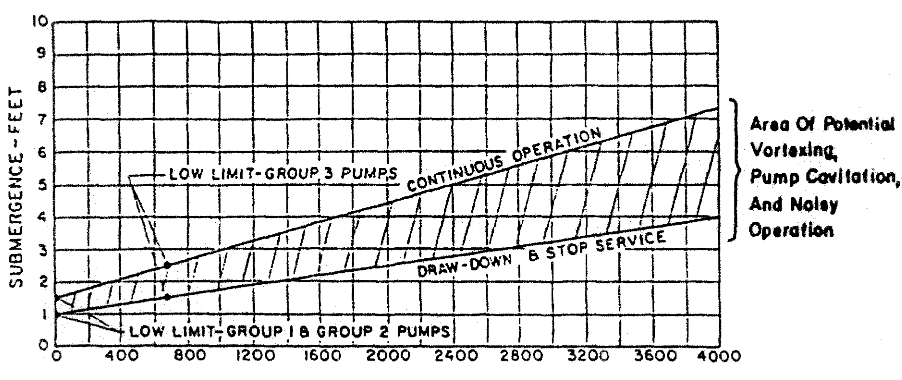


Figure 15-16. Minimum submergence versus g.p.m. for vertical sump pumps.

Most of these pumps are designed like a horizontal pump except they have a self-priming casing. These pumps are used in mine dewatering (which is usually acidic), or in refinery service where a vertical immersion pump may not be used because of the space limitations.

Nonmetallic Pumps

In the past 30 years, the demand for pumps with greater corrosion resistance than is offered by the high alloy stainless steels and nickel-based

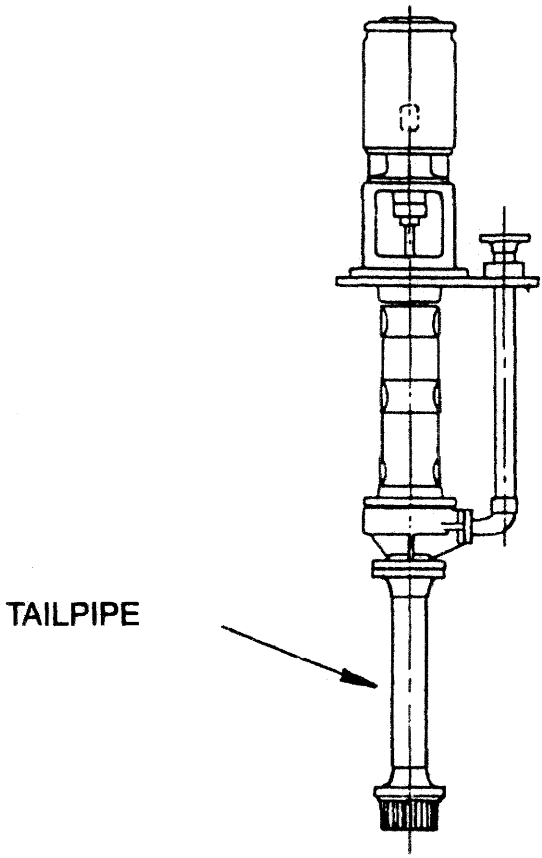


Figure 15-17. Vertical sump pump with tail pipe. (Courtesy of Ingersoll-Rand Company)

alloys has been continuously increasing. This has been evident not only in the chemical industry, but in other industries that use chemicals in their processes. This demand has been met with various expensive alloys such as titanium, alloy 20, zirconium, and many types of hastelloys. Although these materials improved corrosion resistance, they caused other problems.

- Foundries experienced difficulties in castings.
- Existing patterns did not compensate for different shrinkage.
- Machinability was reduced.
- Delivery was longer due to foundry problems.
- The alloys were more expensive.

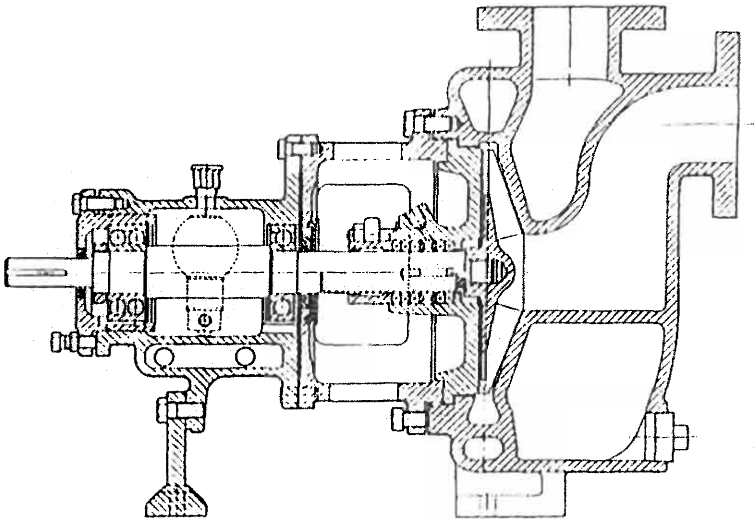


Figure 15-18. Self-primer overhung single-stage (courtesy of Goulds Pumps, Inc.).

- Quality was more difficult to obtain, and it was difficult and costly to comply with NDE (nondestructive examination) requirements.
- Alloy 20 and Hastelloy-C pump case defects exposed during hydrotest required excessive weld repair.

The extensive efforts made to overcome these problems included:

- Creation of new patterns to satisfy metal shrinkage.
- Development of Teflon-lined pumps.
- Development of other linings, such as polypropylene, epoxy, glass, and Kynar.

Although lining the inside of pumps solved many of the earlier problems, new ones were created because linings:

- Were difficult to produce and apply to the complicated pump casing areas.
- Would not properly adhere to metals.
- Would buckle, cold flow, or fail for other strength reasons.

Armored Pump

To overcome the difficulties experienced with pump linings, the armored pump was developed with complete pump casing and other wetted parts produced from carbon, Teflon, CPVS, Kynar, Ryton, etc. These were protected by outside metal plates to hold the required pressure. The various resins were produced with fillers that improved moldability but did not substantially improve strength.

Reinforced Composite Material Pumps

The next development in nonmetallic pumps led to improved manufacturing techniques using thermo resin without armor. Successful resins include glass-reinforced thermoset composites. These have strengths equivalent to the metallic chemical pumps and are suitable for applications of acids, alkalies, oxidizing agents, solvents, and salts with temperature ranges up to 250°F as normal with peak temperatures up to 400°F (Figure 15-19). These pumps were originally called FRP (fiber reinforced polymer) pumps but the term *composites* has basically replaced that label.

Proper selection of composite materials offers many combinations to improve corrosion resistance, lightweight, flame retardation, low-cost magnetic transparency, and complexity of art design. The terms *reinforced plastics* or *composites* generally include two large groups of organic compounds that differ in their make-up. These are thermosetting polymers and thermoplastics.

How does a designer choose between thermoplastics and thermosets? With the present state of the art, the chemical compatibility, maximum applicable temperatures, and consistent quality are about the same for both processes. The differences are listed in Table 15-2.

Thermosetting Polymers

Thermosetting polymers for pump use are reinforced with fiberglass or carbon fibers. During the molding cycle, these materials undergo a chemical change that is irreversible. The resulting material will not soften or become pliable with heat. They have four basic chemistries: polyesters, phenolics, vinyl esters, and epoxies. Each has its own set of advantages, manufacturing processes, and mechanical and chemical properties. The fibers are either continuous or short fibers and are the key in developing the temperature range and corrosion resistance of the final part. There are many manufacturing processes for thermosets and they are often every bit as critical to the final part performance as the selection of the proper polymer and reinforcement combination. Compression molding, transfer molding, resin transfer molding, cold molding, and extrusions are among the most commonly used processes.

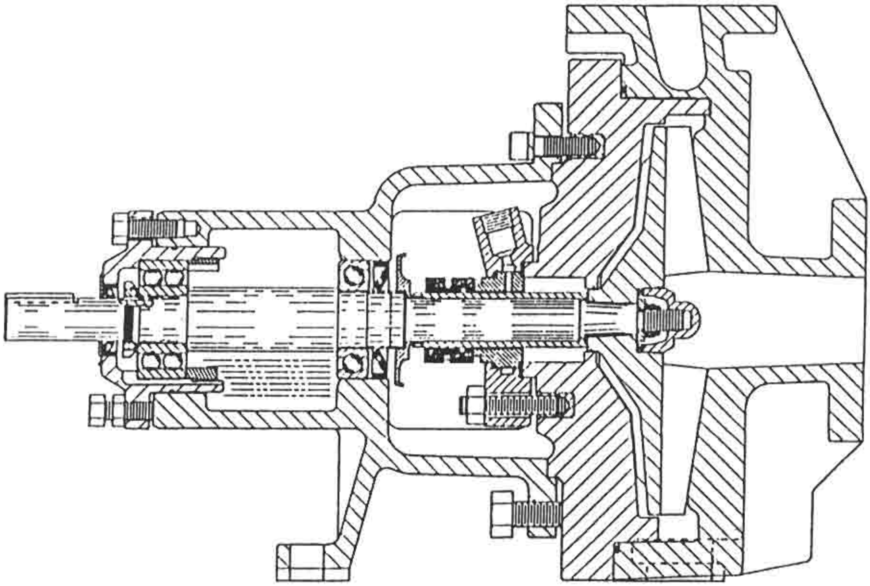


Figure 15-19. Nonmetallic overhung single-stage pump built to ANSI dimensions (courtesy of Ingersoll-Rand Company).

Table 15-2
Comparison of Thermosets with Thermoplastics

Process	Thermoset	Thermoplastics
Average range of molded thickness	.030 to 2.0	.06 to .38
Weight range of material per molded piece	1 to 500 pounds	.5 to 5 pounds
Glass content range by volume	50%–60%	30%–40% (glass degrades when put through the auger of the injection machine)
Length of glass fiber	.25 inch	.060 inch
Strength	Not uniform throughout (anisotropic)	Basically uniform
Minimum annual quantities for design criteria	1,000	10,000

Table 15-2 continued
Comparison of Thermosets with Thermoplastics

Process	Thermoset	Thermoplastics
Obtain additional strength with ribs	Not necessarily	Yes
Tooling	Depends on complexity and size	Generally 15% to 20% higher than compression molding but offset by volume of quantities
Process comment	Compression	Injection. Cannot use compression molding because not enough heat to obtain proper melt flow

Thermoplastics

Thermoplastics do not undergo a chemical change in their processing and will become pliable upon reheating above their yield temperature. Thermoplastic materials are available in a wide range of strengths and application envelopes. They can be divided into fluoropolymers (PFA-PTFE), engineering plastics (LCP-PPS), and general plastics (ABS acrylics, polyethylene, PVC, and polypropylene). Thermoplastic processes such as injection molding, vacuum forming, extrusion, and blow molding offer the design engineer many selections for optimum cost considerations. Selecting a suitable composite requires a complete understanding of the end use application as well as a familiarity with the polymer's physical, chemical, and processing properties. Although direct replacement without design changes is feasible, more often the use of a nonmetallic is optimized by a well-informed specialist.

Table 15-3 shows a general comparison of various resins applications.

Manufacturing Techniques

Two methods used in manufacturing the casing, casing cover, and impeller of nonmetallic pumps are compression molding and resin transfer.

Compression Molding

This process uses matched metal dies that have cored heat transfer passages to control the temperature of the process. The base resin is mixed with appropriate amounts of chopped glass, fillers, and chemical catalysts, inhibitors, and release agents to make a batch. This batch can be set aside in plastic containers for a shelf life of approximately 30 days.

Table 15-3
Resin Performance

Resin Type	Strong Acids	Alkalies (Caustic)	Oxidizing Agents (Bleaches)	Organics (Solvents)	Temp. Limit
General Purpose Polyester (Fiberglass Boats & Bathtubs)	Poor	Very Poor	Very Poor	Poor	160°F
Isophthalic Polyester (Structural Applications)	Fair	Poor	Poor	Fair	190°F
Anhydride Polyester	Excellent	Poor	Poor	Good	275°F
Bisphenol A Polyester	Good	Good	Fair	Poor	250°F
Epoxy	Poor	Excellent	Poor	Excellent	190–250°F
Conventional Vinyl Ester	Good	Good	Fair	Fair	210°F
High-Performance Vinyl Ester Dow Derakane 470 IR GRP Materials	Excellent	Good	Good	Excellent	300°F

When a piece is to be made, a portion of the batch for the piece is measured within an ounce of what is required. If there is too little, the die will not be completely filled; if there is too much, the piece will have an extra thick parting line and will not meet specifications. When the portion of the batch is put into the die, the die closes and compresses the batch at a temperature of approximately >300°F for 10 minutes.

The atmospheric condition to which the whole molding machine is subjected should be controlled for temperature and humidity to obtain the proper quality of the piece. The design engineer has to work closely with the tooling engineer to make sure there is proper flow of material and the path of glass or reinforcement is in the proper location. Experience has shown that reinforcing ribs on casings can be detrimental to the strength because the glass will form a continuous path within the rib producing a knit line. (Knit lines are a result of material coming from two directions and meeting.) This is usually a weak point. In many cases the piece will be stronger by eliminating ribs where it was thought they would be beneficial.

The pieces made from the compression molding process are consistent from one piece to another, both in dimensions and in quality. Poor quality from this process can be a result of (1) a bad mix of batch, (2) batch that is too old, (3) temperature within the die that was not controlled, (4) temperature of the atmospheric conditions that were not controlled, (5) excess humidity within the atmospheric conditions, (6) too little or too much batch, (7) time under compression that was not held as specified.

Resin Transfer

The tooling for resin transfer can be less costly than that of compression modeling, but the number of pieces that can be obtained from the die will be less. In this process, the two halves of the die are separated and reinforced cloth is cut to shape and put into the upper and lower envelope portions of the die. Core made of beeswax is then set within the die. The die is closed and vacuumed and brought up to temperature. A valve is then opened to allow the resin to flow into the die. When the die is filled, it is allowed to cool from 12 to 24 hours. The piece is then removed and set into an oven. The beeswax is then melted and is recovered. The resulting cavity gives the desired shape of the core.

The disadvantage of the resin transfer pieces is that there is a knit line where the two halves of the die meet; therefore, the way the reinforced cloth is put into the die is extremely important to obtain the proper strength of the piece. Pieces made from this process usually do not have the strength of a comparable compression molded piece. Pieces also do not have the consistency of the compression molded piece due to the hand lay up of the cloth. This process is usually used for larger types of pieces where the allowable tolerances are greater than with the compression molded piece. The internal finish for hydraulic passages with this process is not as smooth as obtained with the compression molded piece.

The problems of quality in this process are:

- The quality of tooling that is used to substantiate life of the part for consistency.
- The type of reinforcing glass.
- The method in which the glass is put in the die.
- The amount of glass cloth that is put in the die.
- The quality of resin.
- The control of vacuum to allow the resin to come into the die.
- The quality of the beeswax that is reused from one piece to another.
- The length of time that the piece is allowed to solidify.
- The temperature that is used to melt out the core.

Design Stresses

Designers should be made aware that the stresses advertised in the sample ASTM bars will not necessarily be equivalent to the stresses of the molded piece. This will be verified by the molder as well as the material supplier. When designing with metals, a designer can use the same stress throughout the piece; however, this is not the case with nonmetallic parts. When designing the casing, it is advisable to use different stress levels for the suction nozzle, discharge nozzle, volute, and the wall of the casing. This also applies to the tensile, compression, and hoop stress. Likewise, the modulus of elasticity that is used for the design will change from one process to another. Experience has shown that design values of the actual piece or structural part may be 1/5 to 1/10 that of the test bar. The modulus of elasticity is between 1 to 2 million.

Pressure vs. Temperature

Unlike metallic pumps, there are no standards for pressure-temperature ratings of the flanges. Presently, the ratings change from manufacturer to manufacturer and material to material. Good design practices have demonstrated that the pressure reinforced vinyl ester capability at ambient temperature of the flanges can be equivalent to that of the metal flanges using the same dimensions as the metal flanges. The pressure-temperature gradient is a linear factor and basically degrades above 100°F.

Because heavy wall vinyl ester material is a good insulator, the temperature gradient from the liquid side of the pump case to atmosphere is basically 100°F. This is based on tests of heat soaking the vessel for 24 hours. This allows the manufacturer to pump higher temperatures without excess bearing temperatures. This is also good for the user since there will be little loss of heat from the fluid while passing through the pump.

NPSHR

As with metallic pumps, NPSHR is established by using 3% head loss as a criterion. However, it should be recognized that pumps operating at 3% drop in head or relatively close to this mode of operation in incipient cavitation. Where damage might not be apparent on metallic pumps, it will be observed after a period of time on nonmetallic pumps. It is suggested that the NPSHR offered by the manufacturer be 3 to 5 feet higher than metallic pumps in order to give equivalent life to the nonmetallic counterparts.

General Construction of Nonmetallic Pumps

Most nonmetallic chemical pumps presently being offered for the same hydraulic range as the ANSI pumps are being built to the ANSI dimensional standards envelope. Consequently, most manufacturers are using the same support head and bearing housing construction as on the metallic pumps. This allows the user to be able to interchange the bearing housing parts from metallic pumps to nonmetallic pumps. To obtain additional strength, some manufacturers employ back-up rings that are either separate pieces bolted to the support head or they have support heads that include a back-up ring. The nonmetallic pumps were initially designed with integral nozzles, but there were many molding problems. Some manufacturers resorted to molding separate nozzles and then either molded them to the casing or adhered them to the casing. This was found to be a problem when applying external nozzle loads. With the advancement of materials and dies, many manufacturers now mold the nozzle integral with the casing without incurring nozzle loading problems. Because the materials have moduli that are between 1/15 and 1/30 that of standard metallic materials, it is advisable not to put excess nozzle loadings on composite casings.

Nozzle Loading

There are no standards for the nozzle loads on ANSI pumps, and the manufacturer's specifications are usually referred to for the maximum load. The criterion used by the manufacturer for maximum nozzle loads is usually the movement on the coupling end of the shaft. This may be .0050 to .0100 depending on the size of the pump. This deflection can be caused by:

- The movement of the entire assembly when load is put on.
- Movement of the feet of the casing relative to the bedplate due to the friction force between the two.
- The movement of the bearing housing relative to the bedplate.
- Internal movements causing rubbing of the impeller against the casing.
- Deflection of the bedplate surface relative to the driver shaft.

With nonmetallic pumps the allowable nozzle loads are much less than with the metallic pumps because the casing feet move or deflect under a much lighter load. When nonmetallic beds are employed with either a metallic or nonmetallic pump, the movement of the top surface of the bed is the weak member of the assembly resulting in low allowable nozzle loads. This will occur if the bed is grouted or ungrouted, especially if the force is along the X axis or a moment around the Z axis.

Bolting

As threaded studs will impose tension in the composite case during assembly, it is preferable to use through bolting. This leaves the casing in compression rather than in tension. When through bolting cannot be used, the bolts or studs are fastened into stainless steel or alloy inserts and are molded into the piece. The inserts are gnarled and grooved on the outside diameter to prevent twisting or pulling within the piece when torque is applied to the fastener. The inserts are usually a class 3 fit on the inside diameter for the fasteners. A blind end insert is used to give a positive stop for the studs. When inserts are used, it is best to mold them within the piece rather than post insert them. When they are molded in the piece they should be located at least 1/8 in. below the finished surface so that when machining is being done, the cutting tool does not have an interrupt cut against the insert resulting in weakening of the mounting of the insert. When inserts are used, care has to be taken by the designer that there is proper flow of the composite material to avoid a path for leakage during hydrostatic testing.

Gaskets

With nonmetallic pumps, most main gaskets are O-rings. These can be either round or square cross sectional O-rings. O-rings result in less bolt loading on the main bolts. If gasket surface requires final machining, then it is recommended that the surfaces be coated with the base resin to prevent wicking of the pump fluid through the exposed ends of the glass reinforcement resulting in leakage of the gasket.

Back-up Support for Bolting

To reduce the bolt head or nut loading, it is recommended that when washers are used their diameter should be at least three times the diameter of the bolt. Casing covers usually have inserts for the gland studs as well as inserts for jacking bolts to aid in the disassembly of casing covers. Inserts require optimum strength to absorb radial and axial forces and must be compatible with the atmospheric conditions and in many cases with the liquid being pumped.

Stuffing Box Area

If the glands are made of a composite material, they must be capable of withstanding the torque that is applied without creeping. Depending whether an inside or outside seal is used, the gland may need additional reinforcing with either a metallic back up or extra strength reinforcing cloth. When designing the stuffing box area, the heat transfer of the injection fluid around the seal should be considered. The larger this area, the better the life of the mechanical seal.

The shaft sleeve can be a separate piece that is usually made by injection molding or it can be made integral with the impeller. There are advantages and disadvantages to both. When integral with the impeller, the entire impeller sleeve mechanism needs to be replaced if something goes wrong with the sleeve. When a separate shaft sleeve is used, there is an additional sealing surface between the impeller and the sleeve to prevent fluid from coming in contact with the shaft. When using nonmetallic sleeves, mechanical seals with teflon wedges should not be employed because of the excess fretting. Also, the designer has to be concerned with the extrusion from holding force of set screws on soft nonmetallic sleeves. Split clamping rings using a radial type of set screw are sometimes used to prevent damage to the shaft sleeve.

Mechanical Seals

Because of the corrosive properties of the fluids being used within nonmetallic pumps, many pumps use outside mechanical seals. As a result, the only wetted pieces are the stationary seat and the compatible rotating surface. The remaining springs and secondary seals are external to the stuffing box. However, care should be taken that if an outside seal fails it could be catastrophic. It is recommended that a seal guard be employed when outside mechanical seals are used. This subject is discussed in detail in Chapter 17.

Impellers

Many of the materials that have to be used for the liquids being pumped cannot be readily adhered or mechanically attached to themselves. Therefore, it is difficult to obtain closed impellers and consequently, most impellers are open vane design. Another basic problem with the nonmetallic pumps is the attachment of the impeller to the shaft. Depending on the speed and horsepower, most nonmetallic pumps use more than one key for attachment due to the stress levels of the material. Many impellers are attached by using threaded inserts that are molded within the impeller. The problem here is that care has to be taken that excess stress doesn't occur around the surface of the molded insert that would result in a weak surface between the two materials. Another method of attachment is a multi-keyed or polygon shape that does not require an internal insert because the stresses of the material are distributed throughout its circumference relative to the shaft surface. The disadvantage of the polygon attachment is that there are more surfaces that have to be sealed to prevent external fluid from attacking the shaft.

The sealing of the impellers with either the insert or a polygon fit is similar to that used in the metallic pumps. Most manufacturers will employ the same sealing mechanisms for the two types.

Nonmetallic Immersion Sump Pumps

Typical applications include wet pit chemical waste handling, effluent handling, and liquid transfer operations where broad corrosion resistance is required. These pumps are made of the same basic materials as horizontal nonmetallic pumps, either vinyl ester or epoxy. The hydraulics cover the same basic range as the horizontal pumps and in many cases, the casing impeller, and casing cover are the same parts as used in the horizontal pumps.

The shaft material is 316, alloy 20, Hastelloy B or C, or titanium, depending on the liquid being pumped. Optional shafts of 316 coated with various materials such as kynar are also available. The use of pultruded nonmetallic shafts is being investigated to eliminate all metallic parts for this type of application.

The column supporting the wet end to the mounting plate is a one-piece construction with inserted bearings or a multi-construction of short columns with flanges and the bearing support sandwiched between the flanges of the column. The column material is usually the same base material as the pump and impeller. The bolting of the casing and the columns can be of a nonmetallic material compatible with the fluid.

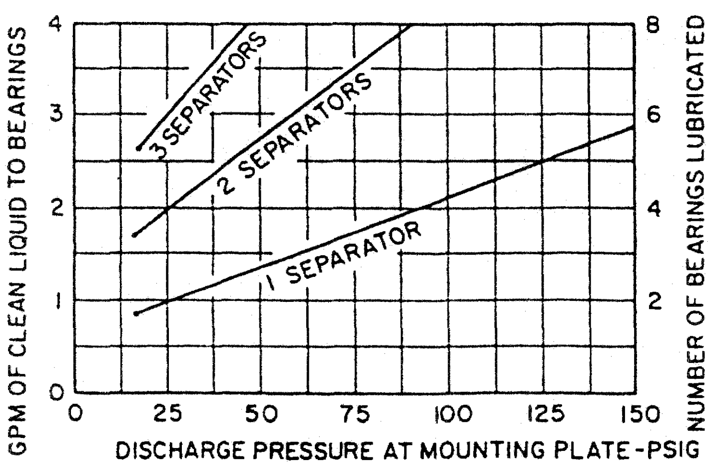
Bearings are made out of teflon or carbon with spiral flutes. The lubrication is either external or clean product lube. Clean liquid for lubrication is one that has less than 5 micron particle size. The lubrication to each bearing should be at least one half GPM at 160°F temperature or less and at a pressure of approximately 25 psig. Carbon bearings are furnished when external lubrication or injection pressure is not adequate.

Figure 15-20 shows when to supply carbon or teflon bearings based on particle size in the fluid and the flush pressure available to these bearings. It also shows when cyclone separators are required and what flow for a given flush pressure is obtainable from the separators. The lower bearings are usually twice as long as the line bearings to absorb the radial thrust developed by the impeller. A gap or relief hole is placed between the throat bushing of the casing cover and the bearing itself so that dirty liquid under pressure will be relieved of pressure and not be forced into the bearing clearances resulting in short life.

BEARING APPLICATION CHART — PRODUCT LUBE

	Flush Pressure (1)		
Particle Size (2)	>10 PSIG	>15 PSIG	>25 PSIG
< 10 Micron (Clean)	Carbon	Teflon	Teflon
> 10 Micron < 400 Mesh (Fine)	Carbon	Carbon	Teflon
> 400 Mesh < 20 Mesh (Coarse)	Note 3	Carbon (4)	Teflon (4)

1. Discharge pressure at mounting plate. Min flow of ¼ GPM per bearing is required. ½ GPM is recommended.
2. Particle sizes are as follows:
   10 micron = .0004 in.
   400 mesh (fine) = .0015 in.
   20 mesh (coarse) = .0328 in.
3. External flush only at 25 PSIG (Teflon bearings)
4. Cyclone separators required. Refer to chart below.



Note: Add 2 gpm per separator to total flow requirements of pump to allow for flow taken by separators.

Figure 15-20. Bearing and lubrication for nonmetallic vertical sump pumps.

A lip seal or closure is installed in the mounting plate where the shaft passes through preventing gases and vapors from escaping out of the sump. If the sump is under pressure or has toxic fluid, a mechanical seal is employed. Likewise, a gasket is placed between the mounting plate and pit cover. A strainer is placed at the bottom of the casing. It is made out of a polypropylene material, the net area of which should be three times the entrance area of the suction nozzle.

Driver

The pump shaft is connected directly to the motor by a rigid adjustable coupling, and in-line motors are used to absorb the axial thrust. If a normal thrust motor is used, then a separate thrust bearing is used within the support head to absorb the axial thrust.

Level Controls

With chemical sump pumps, the level control is usually encased to prevent the fluid from coming in direct contact with the switching mechanism.

Stilling Tubes

If it is anticipated that swirl or vortexing will exist within the sump, the level controls are mounted within a stilling tube that indicates the true level of the liquid within the sump.

Mounting Plates and Pit Covers

These plates are made out of the same base material as the pumps. Depending on the size, the pit covers may require reinforcement of steel angles or channels. These steel pieces are encapsulated so they are not exposed to the atmosphere.

Processes

Chemical pumps handle a variety of liquids that could be concentrated, diluted, or just a trace. The concentration could be from 5% to 50% or the liquid could be in parts per million. Temperature could be hot or cold, and a pump can be sold to handle maximum, minimum, or normal temperature. The range of temperature could cause thermal shock and can vary as much as 150°F from the cold to hot application. The rate of activity of the fluid changes approximately two to three times for each 18°F change in temperature. The liquids could have either an alkaline or acidic pH level. Liquids may contain solids that might cause erosion, corrosion, or settling problems that would result in clogging. The liquid may have entrained air that would make a reducing solution into oxidation or it could have inhibitors to reduce corrosion or accelerators to increase corrosion. The impurities could lead to something called discoloration or solution breakdown.

Final pump material selection is a collective decision based on input from the pump designer, plant operator, material supplier, and available technical literature.

Pump Corrosion

The types of corrosion encountered in a chemical environment fall into eight typical categories.

- General uniform corrosion at a uniform rate over entire surface, either very slow or very rapid.
- Crevice corrosion that is a localized form from small stagnant solutions in areas such as threads, gasket surfaces, or drain holes. Crevice corrosion is caused by a differential in concentration of metal ions and oxygen added to the main body. This causes an electrical current to flow, causing the damage.
- Pitting is localized. It is manifested as small or large holes usually produced by chlorines.
- Stress—corrosion occurs at cyclic stress on shafts.
- Intergranular corrosion—usually occurs in the presence of heat.
- Galvanic action.
- Erosion-corrosion—corrosion plus mechanical wear such as cavitation.
- Selective—that is, leaching corrosion or degraphitization usually not found in chemical pumps.

Pump Materials

The typical material of construction to combat corrosion is either a 304 or 316 stainless steel that is superior to austenitic or ferritic steels. Other materials would be composite plastic such as PTFE and FEP. Usually fiber reinforced plastic is used for strength and chemical resistance. This includes vinyl esters, epoxies, polypropylenes, and phenolics. Ceramic or glass is avoided because of low mechanical properties.

Some of the process liquids found in chemical plants are listed along with pump materials used in the various environments.

Chlorine

- 65% of the chlorines are used for organic chemicals such as vinyl chloride, pesticides, fluorcarbons.
- 15% of the chlorines are used for producing pump and paper.
- 10% is for inorganic chemicals.
- The remaining 5% is for sanitation, potable water, and waste water that are used in municipal water works and sewage plants.
- 50% of caustic soda is used for the chemical industry.

- 15% is pulp and paper.
- The remaining is in aluminum, rayon, cellophane, petroleum, soaps, and foods.

The elecrolytic plants produce chlorine and caustic soda using a range of 250 to 1000 GPM and about 20 to 30 pumps. They also have 15 to 20 peripheral transfer pumps for hydrochloric acid, diluted H_2SO_4, and sodium hypochlorite. These plants use pumps of titanium, nickel alloys, cast iron alloy 20, CD-4MCu, and nonmetallics such as vinyl ester.

Sodium-Hypochlorite

NaOCl is a byproduct of the chlorine and caustic process. It is found in the bleach plants of paper mills with a concentration of 12% to 20%, in commercial bleach such as household Chlorox, and OEMs for swimming pool chlorination. The electrolysis of sea water (brine) with a concentration of 1% to 3% is used for bleach plants, pulp mills, bleach plants for textile mills, and municipal waste treatment to kill bacteria. These plants use large amounts of sea water for cooling to prevent pipes from fouling with algae. It is also used in the pretreatment of desalinization intake and the pretreatment of secondary recovery brine in oil production. Pumps are alloy 20, titanium, and nonmetallic.

Hydrochloric Acid—HCl

Also known as muriatic acid, it usually requires Hastelloy B or titanium pumps. The primary consumption is pickling of steel for use with oil well acidizing where acid increases the permeability of wells by dissolving part of the limestone and dolomite formations. High purity aluminum chloride produced by aluminum hydroxide and HCl is used in pharmaceutical and cosmetic usage.

In food processing, HCl is used in the manufacture of sodium slutamate and gelatin for conversion of cornstarch to syrup or adjusting the pH value in breweries.

Sulfuric Acid—H_2SO_4

With a 97% concentration, cast iron pumps are used. Eighty percent or less concentration is either alloy 20 or vinyl ester nonmetallic pumps. It has a very high boiling point, therefore a minimum loss is incurred at elevated temperatures. It is an excellent drying agent in the manufacture of chlorine gas. It is used for making fertilizers and explosives.

Ferrous and Ferric Chloride

This is a byproduct from acid pickling of iron and steel. It is used for etching reagent in copper clad printed circuit boards, for electronics, or as a chemical coagulant for water treatment on waste water. The pumps are either titanium or vinyl ester.

Chlorinated Hydrocarbons

Chlorinated hydrocarbons are used to produce PVC, chloroform, carbon tetrachloride, solvents, flame retardants, insecticides, adhesives, pharmaceuticals, metal cleaning, and dry cleaning. The pumps are usually 316 or ductile iron.

Ethylene and Propylene Glycol

Ethylene glycol is permanent antifreeze. It is also used in alkaline resins for coating in brakes, shock absorbers, and latex paints. Propylene glycol is used in the manufacture of unsaturated polyester resins. It is used to make cellophane, tobacco moisture retention material, brake fluids, and food additives. These pumps are either 316 or ductile iron.

Synthetic Glycerine

It is used for making resins, cosmetics, cellophane, tobacco, food beverages, and explosives. The pumps can be nonmetallic material.

Corn Syrup

Used in corn oil process. Usually 316 or nonmetallic materials.

Dyes

Used for such things as the ink in ball point pens and dying silk and wools. Usually nonmetallic pumps.

Pesticides

Insecticides such as malathion for controlling insects; herbicides for controlling plant growth. These materials are solvents and nonmetallic pumps are usually not suitable for this application.

Sodium Chlorite—Na ClO_3

This is used for bleaches, herbacides, explosives, and rocket fuels. Pumps can be in 316 or nonmetallic.

Pulp and Paper

Sulfite process: usually acids and bleaches. Sulfate (which is Kraft) is used in making strong cardboard containers and wrappings. Usually use titanium or vinyl ester pumps.

Metal Finishing

Electronic plating, cleanings such as oil, rust, and scale. Usually use ductile iron pumps. Plating usually use nonmetallic pumps to prevent stray currents. Waste treating of the vent plating baths are usually 316 or nonmetallic pumps.

Carbon Steel Pickling

Sulfuric acid was replaced by hydrochloric acid to give a better finish. Use 316, alloy 20, or nonmetallic pumps.

Stainless Steel Pickling

Nitric hydrofluoric acid solutions process is not suitable for nonmetallic pump application.

Desalinization and Water Purification

Desalinization and water purification are done through distillation or reverse osmosis. Usually a simple plant used for marine or power plants is used for distillation. The size of the plant can be scaled up or down. It can tolerate a wide latitude of feeder water quality, but requires high temperatures and results in more corrosion and more maintenance. It is usually only efficient using low pressure steam. The temperature is from 70° to 250°F with a vacuum as low as 25 inches of mercury. The pumps are usually 316 or alloy 20.

The reverse osmosis process using a membrane is simple, compact, more efficient, and uses ambient temperature. The membranes are sensitive to metal pick ups. It can purify dirty well water, brackish water with 200 to 600 psi. If it is used for desalinization of salt water, the pressures are from 800 to 1000 psi. Miscellaneous pumps can either be 316 or nonmetallic.

Secondary Oil Recovery (Waterflood)

When no convenient surface water is available, deep well turbine pumps are used to pump the water containing calcium sulfite to the surface. This is filtered or chemically treated before reinjection to prevent clogging of the pores in oil sand. The oil and brine mixture then comes to the surface where it is separated. The water is a corrosive brine with hydrogen sulfide from the contact with oil. The brine is chemically treated and filtered for reinjection or disposed of as waste. Chemical pumps of 316 or nonmetallic can be used to transfer the brine.

Mining—Copper Leaching and Uranium Solvent Extraction

Copper ore is normally less than 1% copper. Flotation or leaching or solvent extraction is used to upgrade the ore. Copper emerges as a copper sulfate and then is pumped to an electrolytic cell where it is plated out. Pump material, either 316 or nonmetallic, is used at ambient temperature.

For uranium, a solvent extraction is settled and filtered. The clarified solution is mixed with kerosene and organic amines. The solution is stripped and uranium is precipitated as uranium oxide. Nonmetallic pumps or 316 for low head or ambient temperature are used.

Industrial Waste Treatment

These vary with great quantities due to the nature of the product and process that they drain. The range of fluids is from a discharge of great volumes of cooling to small but concentrated baths of inorganic or organic substances. The pump material is of ductile iron, 316, or nonmetallic. Some of these waste treatments are wastes containing mineral impurities, steel pickling, copper bearing wastes (where very small amounts of copper—less than 1 mg per liter—will interfere with life in a stream or biological sewage treatment works). Wastes containing chromates or cyanides are used for electroplating and electrolytic operations where the maximum is less than 1 mg per liter. They are also used for gas and coke plant wastes, oil-field brines (which are petroleum refinery wastes, mining wastes, or wastes containing organic impurities). They are also used for milk processing, meat packing, brewery and distillery, vegetable and fruit processing, textiles (such as wool, cotton, silk, linen, and dyes), laundries (which have soap, bleaches, dirt and grease), tanneries, and paper mills (which have black, green, and white liquors). The pumps could be ductile iron, stainless, alloy 20, titanium, or nonmetallic.

Material Selection

A selection of corrosion resistant materials is shown on Table 15-4.

Table 15-4
Thermoplastics Corrosion Resistance Guide

Key: A Acceptable Q Questionable NR Not Recommended

Media	200°F Poly-Phenylene Sulfide	200°F Penton	200°F Kynar	200°F Teflon	200°F Poly-sulfone	200°F Noryl	200°F Nylon	200°F Polycar-bonate	316 SS	Carbon Steel	Alu-minum 3003
Media	200°F Poly-Phenylene Sulfide	200°F Penton	200°F Kynar	200°F Teflon	200°F Poly-sulfone	200°F Noryl	200°F Nylon	200°F Polycar-bonate	316 SS	Carbon Steel	Alu-minum 3003

* *Polyphenylene sulfide grades containing glass fiber and/or mineral fillers will be less chemical resistant than indicated.*

Information taken from Phillips Petroleum Corrosion Resistance Guide

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