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TABLE 1 Calculation of required absolute suction conditions ($\Omega_s = N_s/2733$)

Pump Type	Operating Speed—rpm	Specific Speed, N_s	NPSH Required ft (m)	Submergence ft (m)
Single Suction*	1800	2163	48 (14.6)	18 (5.5)
Single Suction*	1200	1529	30 (9.1)	0**
Double Suction	1800	1442	31 (9.5)	1 (0.3)***
Double Suction	1200	1019	18 (5.5)	0***

*Can be top or bottom inlet
**Barely acceptable with submergence sufficient to prevent vortexing at inlet
***Very safe

Figure 7 illustrates an interesting installation of a float-mounted pump of unique design. The pump is of cantilevered shaft design (no submerged bearings) with the pump utilizing the motor bearings. Pump construction is shown in Figure 8. This is ideal for barge mounting. It involves a special motor, but the lower center of gravity permits a smaller barge and readily compensates for the extra cost of the motor. No submerged shaft bearing is required, so dirty water can be handled with little maintenance. A double discharge design also provides balanced hydraulic side thrust and eliminates the need for a shaft seal.

If floats are to be used for pump mounting, considerations in addition to hydraulics must be examined. Floats must be designed for safe operation in all circumstances. Floats for individual pumps are simple and readily moved about a mining operation; however, there are other factors beside the weight of the pump and motor. A portion of the discharge line (usually a hose on floats) must be supported by the pump float. Consideration must be given to the possibility that a number of workmen might gather at one side of the float and overturn it. Stability calculations must be made by considering

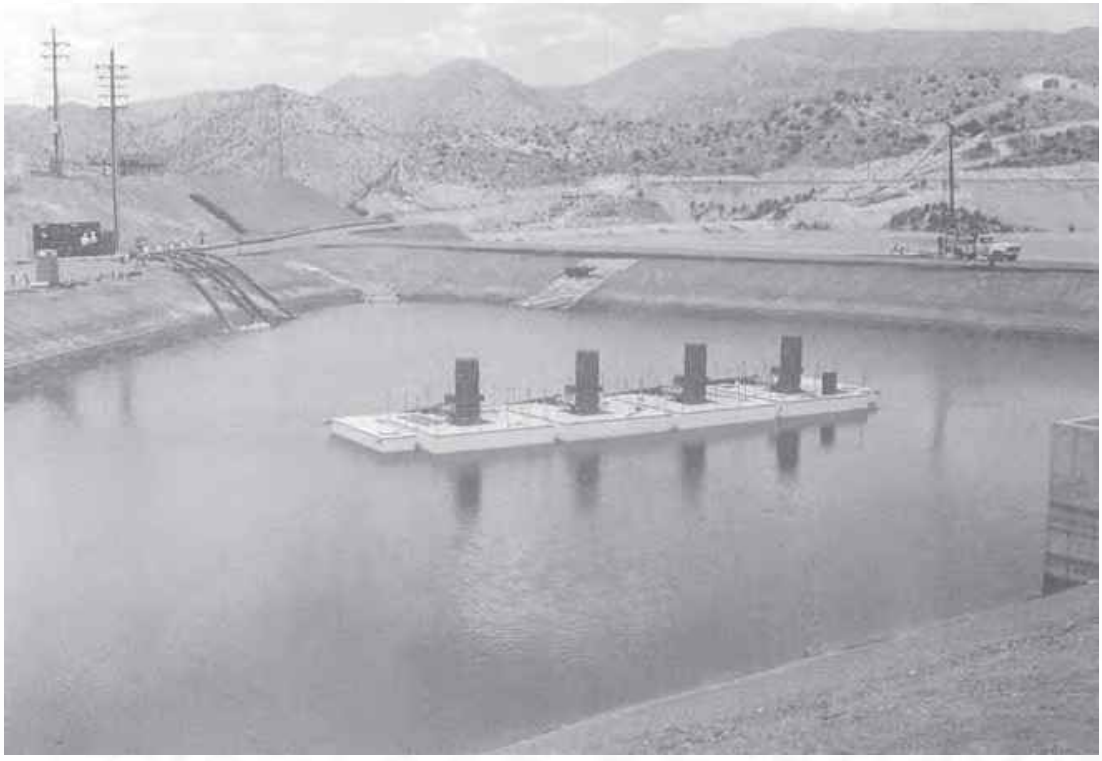


FIGURE 5 An installation of four floats with top inlet cantilevered shaft pumps (Hazleton Pumps, Inc.)

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FIGURE 6 Float-mounted pumps designed to skim the clean liquid in a settling pond (Hazleton Pumps, Inc.)

lateral rotation of the float and the uprighting forces due to the additional buoyancy on one side (or end). The center of gravity must be within the center of buoyancy. If the center of gravity moves beyond the center of buoyancy, the float will overturn. It must also be remembered that, after a pontoon is submerged, that is the limit of buoyancy. In addition to these factors, wind effects must be considered. If the climate is not too severe,

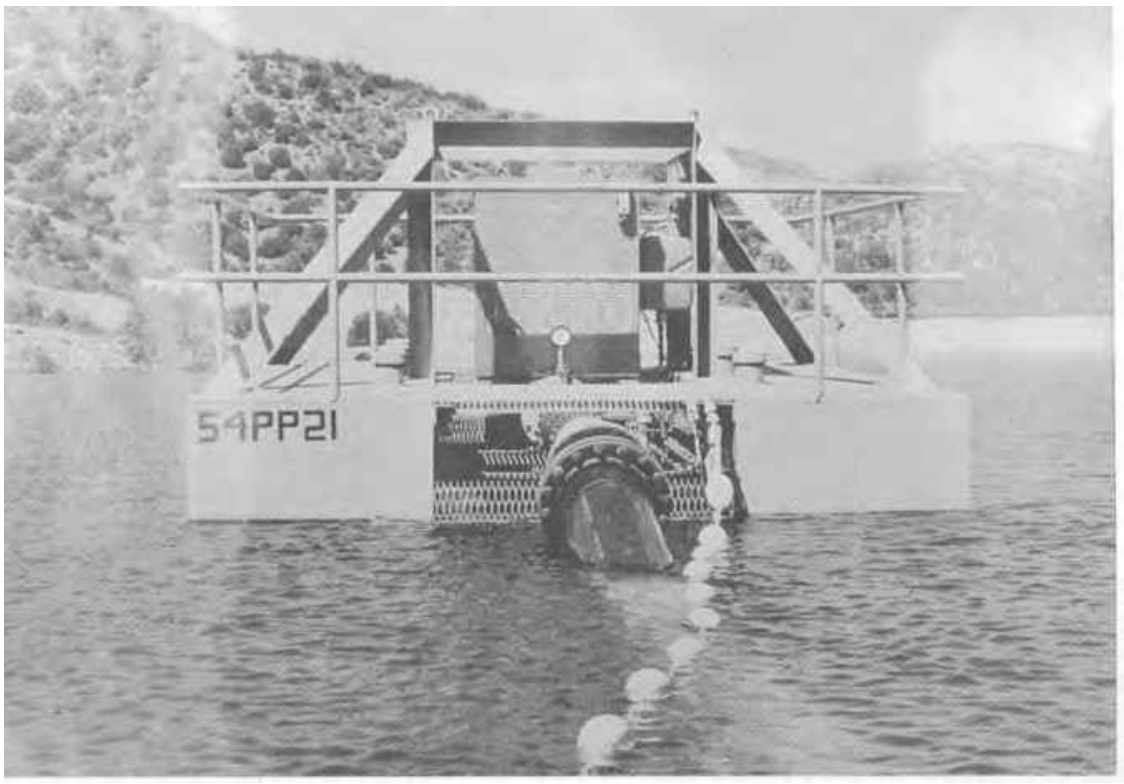


FIGURE 7 Unique design float-mounted pump (Hazleton Pumps, Inc.)

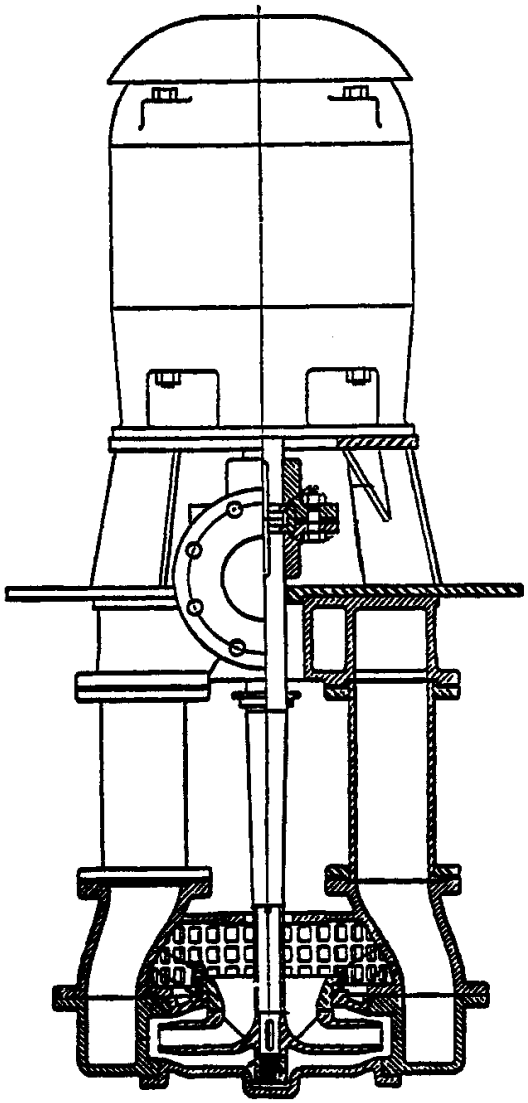


FIGURE 8 Cantilevered shaft design utilizing motor bearings (Hazleton Pumps, Inc.)

pumps on simple floats can be protected from ice damage if the float is surrounded with a simple air bubble system.

As the pump flow rates increase, fixed installations become more manageable. Figure 9 shows four 24-in (610 mm) discharge, 1250 HP (750 kW) vertical pumps. The pumps are of stainless steel in a mining operation. This is an ideal installation as a structure is in place for pump maintenance.

The selection of the discharge line will have a significant effect on the total pumping output. If selected for normal pumping conditions, the pipeline may not add significant capacity to the system by the use of an additional pump. As an example, examine an installation utilizing a 14-in (355 mm) Schedule 40 steel pipe discharge line for a normal flow of about 5000 gpm (1135 m^3/h). Assume a friction head of 70 ft (21.3 m) and a static head of 180 ft (54.9 m), for a total pumping head of 250 ft (76.2 m). Figure 10 shows that adding a second pump to the system will not double the pumping capacity as the two pump characteristics are added to the system head curve. In this exaggerated example, adding a second pump will increase the total flow by less than 1000 gpm (227 m^3/h). Note that each pump now will discharge less than 3000 gpm (680 m^3/h). The obvious answer is to either use a larger discharge line and a different pump selection, or the use of a separate discharge line for each pump. In that case, the total capacity would be 10,000 gpm

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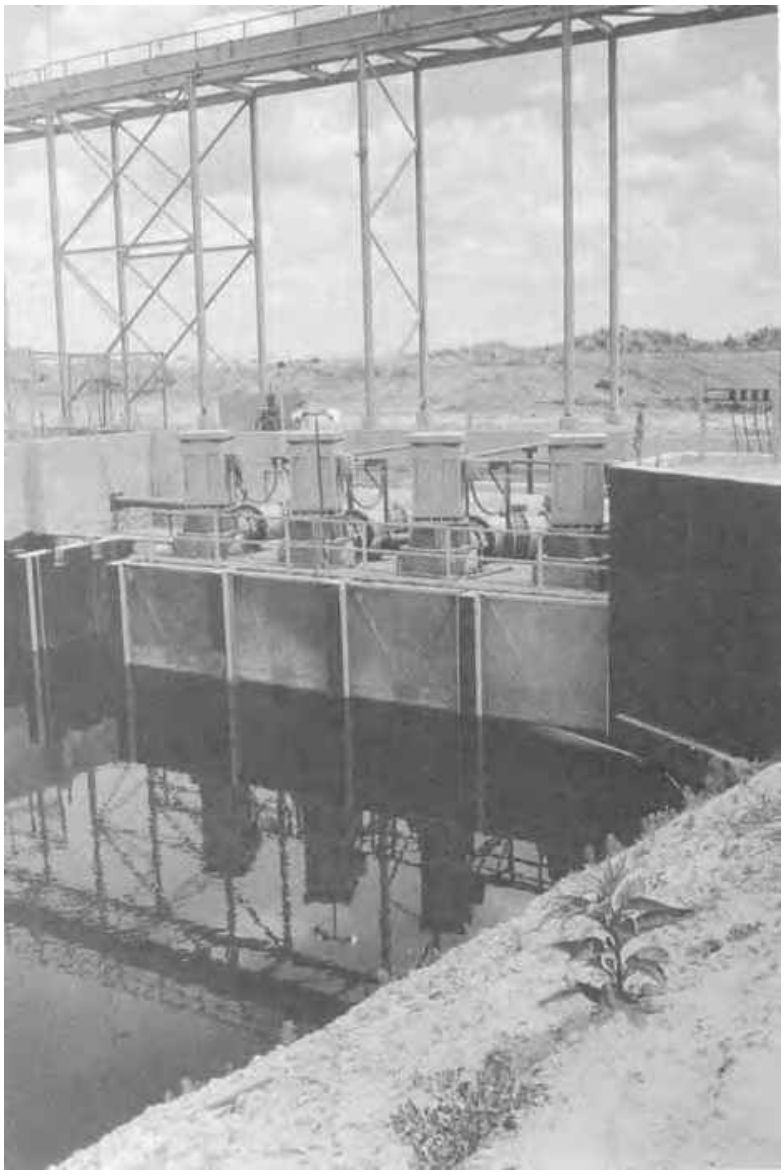


FIGURE 9 24 in (610 mm) discharge, 1250 HP (750 kW) vertical pumps (Hazleton Pumps, Inc.)

(2270 m^3/h). This becomes a study in economics for each installation. In the above example, this installation is obviously poorly designed. As pipeline velocities increase, discharge line transients may present problems on shutdown. The author is aware of one long pipeline that failed in 17 places on the first shutdown. Other factors must also be considered, such as the projected life of the project and the cost if the project is flooded.

Much has been said about pumping from mines. Obviously, the least costly method is to keep the water from entering the mining operation. It is not always applicable, but a ring of "deep-well" turbine pumps, on the periphery of the operation, can lower the water table sufficiently to keep the mining pit relatively dry. A typical large turbine pump with submersible motor is shown in Figure 11.

Pumps for operation in deep mines have different considerations. If a mine is not more than approximately 1200 ft (365 m) deep, Schedule 40 pipe is generally satisfactory for the pressure involved. For example, 12-in (305 mm) seamless Schedule 40 steel pipe is listed as 1200 lb/in^2 (82.75 bar) hydrotest pressure. However, if the water is corrosive, wall thickness allowance must be made, or stainless steel pipe used. 1200 lb/in^2 converts to approximately 2700 ft of water. Allowance must also be made for water hammer if the line velocity is high. Generally, average line velocities of 10 ft/s (3.1 m/s) will produce high

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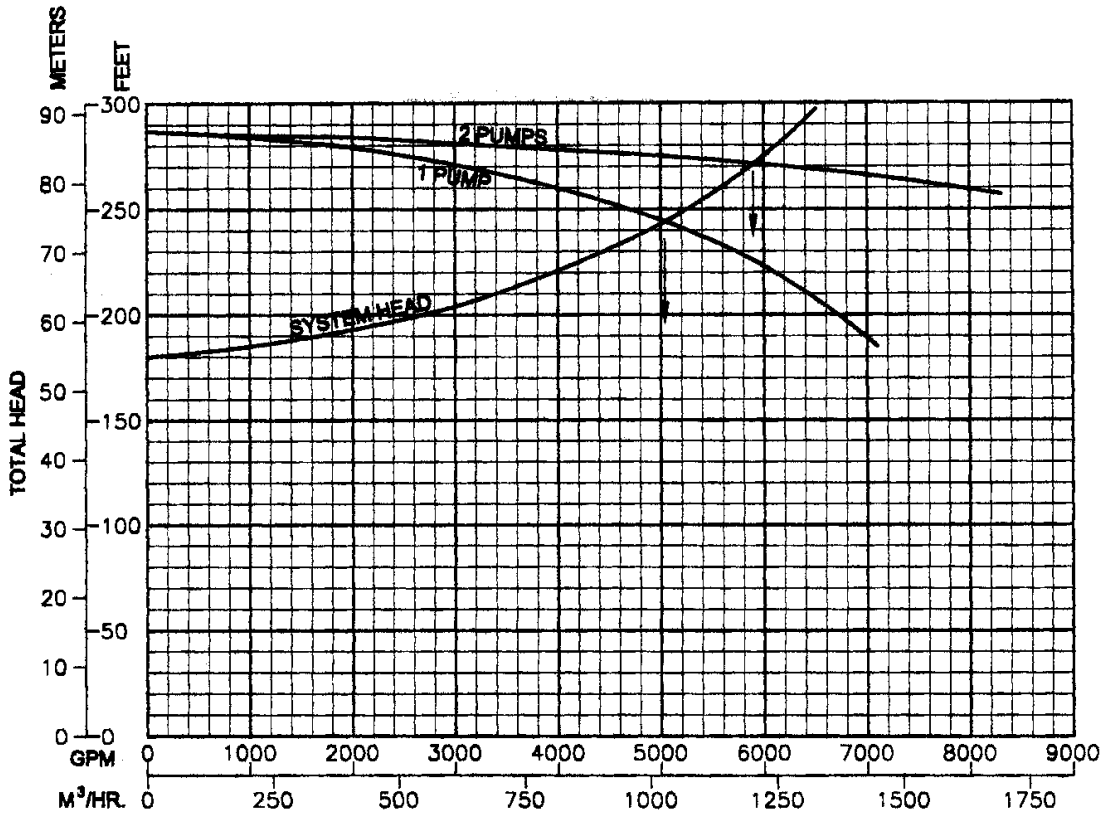


FIGURE 10 One and two pump operation on a system curve

transient pressures upon pump shutdown, so more conservative velocities should be selected. As these installations are generally long term, considerable savings can be obtained by reduced power requirements if the pipeline velocities are near 5 ft/s (1.5 m/s).

In designing a mine pumping system, an analysis must be made where the water is coming from. It is generally not practical to put the main pump station at the bottom of the mine if the bulk of the water is occurring at an upper level. If this is the case, a decision must be made for the pumping condition of the mine bottom pumps. Should they simply pump to the upper level or directly to the surface? Both financial and safety considerations are essential. Failure of upper level pumps may jeopardize the mine safety. These are decisions that involve much more than basic equipment selection. One solution to the problem of a possible power failure is the use of submersible pumps at the shaft bottom. Single-stage submersible pumps, as shown in Figure 12, are available to 600 HP (448 kW) and can readily pump to upper levels in case of flooding at the bottom level.

If the mine is deeper than 1200 ft (365 m), other approaches must be considered. Consideration of Class 250 or 300 valves and pipefittings may show that the cost warrants pumping only to an upper level to stay within the limits of the Class 150 fittings. This does not necessarily mean a duplicate pumping station as series pumps can safely be installed at an upper level. This is practical only if the bulk of the water occurs at or near the bottom level. Because there are mines deeper than 2000 ft (365 m), the problems become more acute. In any event, the main consideration is to conduct a study to determine at what level the greatest flow of water occurs.

A very old but interesting photo (see Figure 13) shows a sealed pump-room housing three 1000 HP (746 kW) bronze constructed pumps in a coal mine. This pump room had access only from an upper level so water could rise in the mine by approximately 200 ft (36.5 m) before flooding the pump room.

Not all mines suffer from ingress of a great deal of water, but many of those that do have closed because the cost of pumping was too great. An example is shown in Figure 14. There were three 2000 HP (1490 kW) pumps in one pump room in a zinc mine. Note in this

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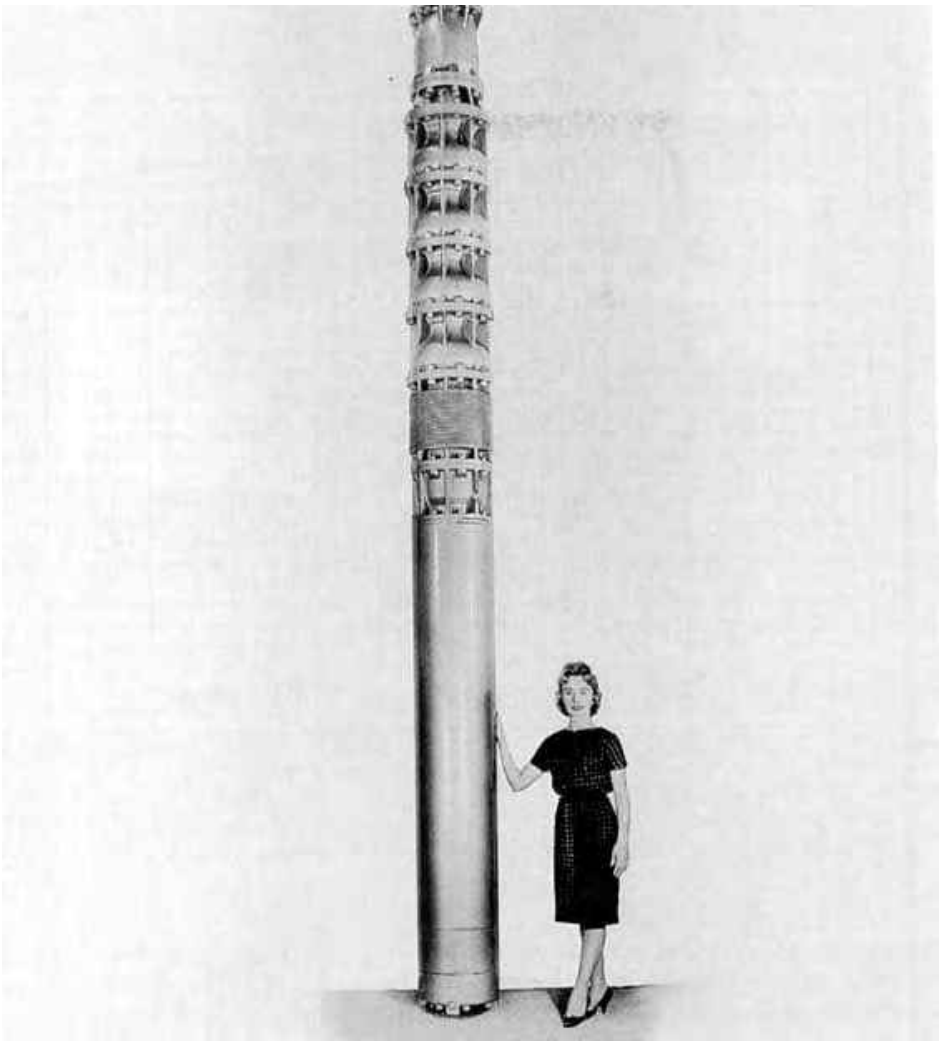


FIGURE 11 A typical large turbine pump with submersible motor (Hazleton Pumps, Inc.)

photograph that these large pumps are in segments so they can be lowered into the mine and reassembled.

Pump discharge line forces must be calculated for all installations. The reactive forces from the piping must be isolated from the pump. Underground pumps generally operate at high pressure and the forces generated are high. Failure to adequately support and restrain the piping can cause pipe failure and result in severe pump misalignment.

MATERIALS OF CONSTRUCTION

Most water pumps perform quite satisfactorily with bronze impellers, wearing rings, and shaft sleeves, but the dirty or corrosive waters in mine service require superior materials. Because mine waters range from neutral (pH of approximately 7) to severely acidic (as low as pH 1.5 in some coal mines) and to very basic (as occurs in limestone mines and so on), there is no universally best material. The choice obviously is the lowest-priced material that gives satisfactory service life.

Many times, the decision must be made not on the basis of the best available materials, but on the basis of which material will best withstand the abrasive conditions during the intended life of the project. If there is a five-year anticipated life of the mine, there is little advantage in selecting materials that will last twice as long. Conversely, a mine with

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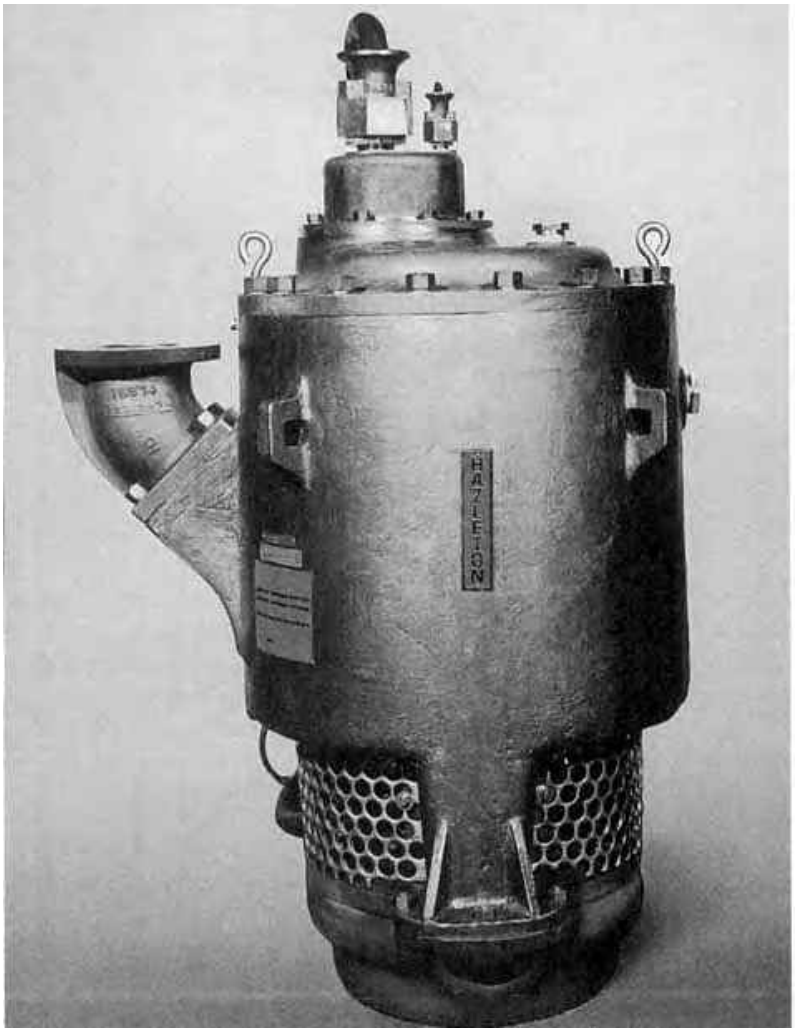


FIGURE 12 Single-stage submersible pump (Hazleton Pumps, Inc.)

a projected life of 25 years would require the best commercially available materials. There are also the exotic alloys, but their use is seldom justified.

Assuming that there will always be a slight amount of solids present, the metallurgy given in Table 2 should be considered.

Care should be exercised when selecting alloys because many stainless steels do not have as great a strength as carbon steel. The same applies to bolting. Thus, a pump rated for high-pressure service with a carbon steel or alloy steel casing and bolting may not be suitable when made of bronze or stainless steel.

Although ceramic-coated shaft sleeves are excellent in many applications, remember that ceramic coatings are porous and that the base metal must be able to withstand the environment. Also, some ceramics will not be suitable in strongly basic water. Others, however, are suitable, and so a general specification for ceramic coating should never be made. A plasma-applied ceramic is generally denser and more serviceable than one applied by a simple flame spray.

HYDRAULIC CONDITIONS

Low-Head Pumps For low-head pumps, up to approximately 150 ft (46 m), the installation must be carefully checked to prevent cavitation during periods of low-head operation. This is particularly important during one-pump operation on a parallel system. For

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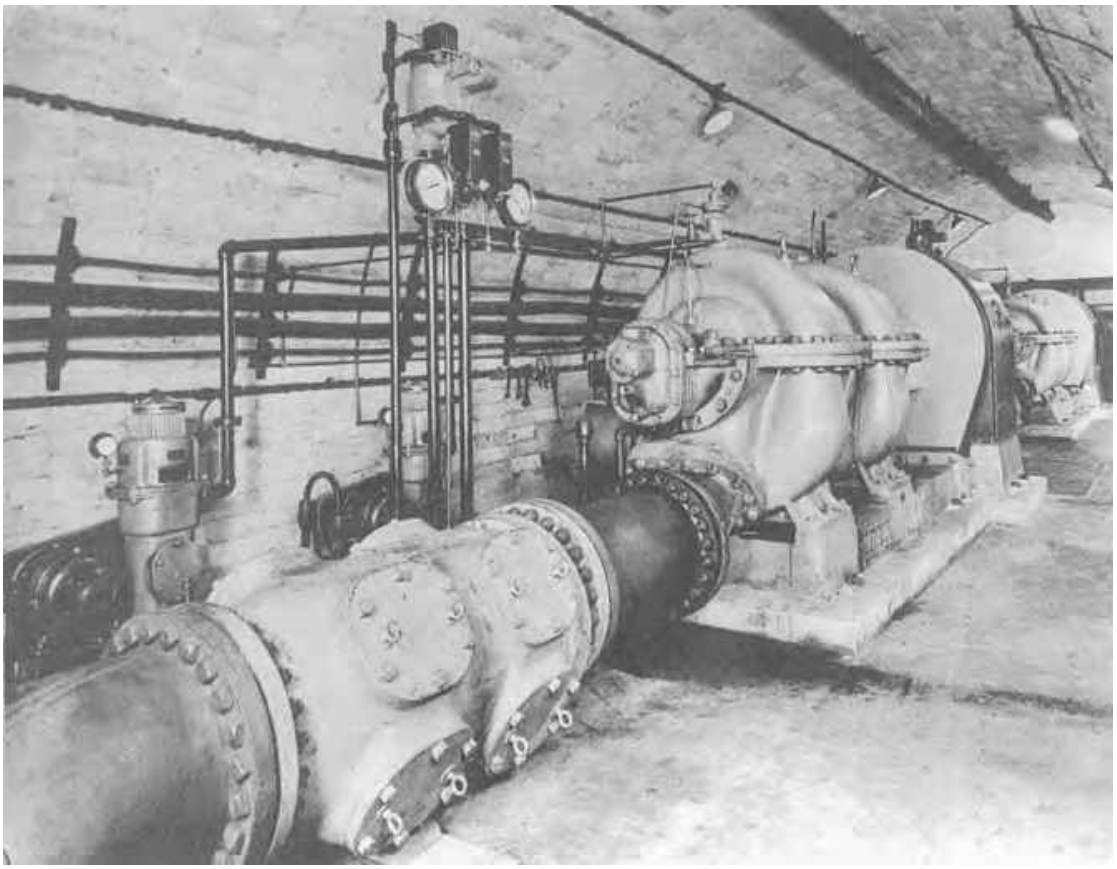


FIGURE 13 A sealed room in a coal mine housing three 1000 HP (746 kW) bronze constructed pumps (Hazleton Pumps, Inc.)

example, if two 5000-gpm (1136-m^3/h) pumps are discharging into a common discharge line against a static head of 80 ft (24 m) and a frictional head of 60 ft (18 m), the frictional head will be only 15 ft (4.6 m) when one pump operates alone at 5000 gpm (1136 m^3/h). The total head will now be only 95 ft (69 m), and the single pump will carry out to a much higher capacity. Unless sufficient NPSH is available for the single-pump runout point, the pump will cavitate. Although the two pumps should be selected for 5000 gpm (1136 m^3/h) at 140 ft (43 m) total head, the required NPSH should be determined not at 5000 gpm (1186 m^3/h) but at the capacity corresponding to the intersection of the pump and system curves.

High-Head Pumps For high-head pumps, 1000-ft (305 m) head or more, the risk of cavitation for single-pump operation on a parallel system is less than for low-head pumps. For example, if the static head is 1000 ft (305 m) and the frictional head is 60 ft (18 m) when two 5000-gpm (1136-m^3/h) pumps are operating, the total head will decrease from 1060 ft (323 m) to only 1015 ft (309 m) when one pump operates at 5000 gpm (1136 m^3/h). This means that, for either one- or two-pump operation, the capacity of each pump will be approximately the same and the risk of run out cavitation is minimal.

Waterhammer and Pressure Pulsations A waterhammer analysis should be made of both high- and low-pressure pumping systems. Although the transient pressure pulsations are related to the rate of change of velocity rather than the magnitude of the steady-state condition, mine experience indicates that waterhammer problems can be anticipated when pipe velocities exceed 10 ft/s (3 m/s). In high-pressure pumping systems, it is not unusual for transient pressure pulsations to be as high as 300 lb/in^2 (2068 kPa) above or below the steady-state pressure.

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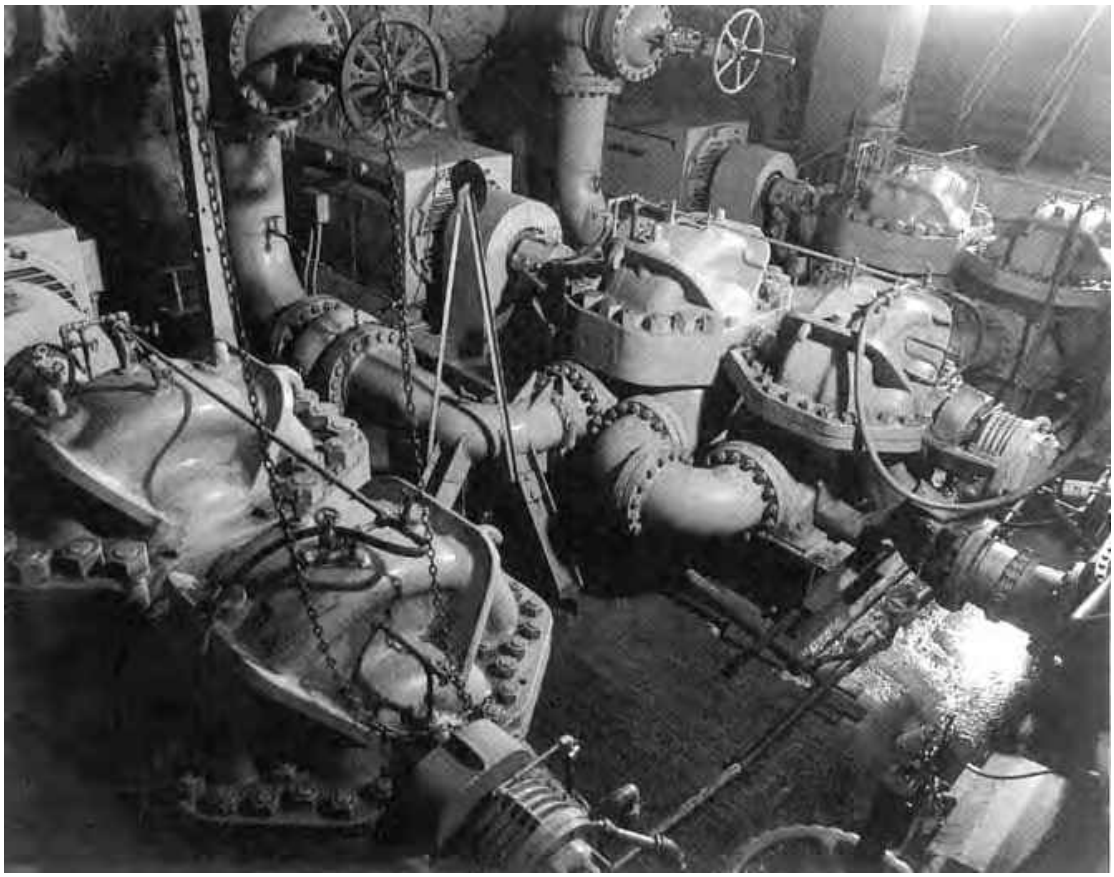


FIGURE 14 Three 2000 HP (1490 kW) pumps in one room in a zinc mine (Hazleton Pumps, Inc.)

Transient pressure pulsations have been experienced in low-pressure pumping systems. The danger here is that the low-pressure portion of the cycle will fall below atmospheric pressure and the pipe will collapse.

Although any pumping system for mine service should be analyzed in detail for transient pressure pulsations, experience has shown that adequate air bottles have proved to be one of the most effective and least expensive means of surge suppression. Slow-closing valves and flywheels have been used, but they must be sized correctly. This is especially true with high-speed pumps because these units possess little rotational inertia and will decelerate very rapidly on shutdown, with accompanying high-pressure surge.

SUMPS

Permanent sumps are seldom used in open-pit mines because the sump area generally moves as the mining operation progresses. This means that the pump station must be portable, and the installation of the pumps on a barge provides the most convenient arrangement. Either horizontal or vertical pumps may be used, but vertical pumps eliminate the need for priming equipment. If the vertical pumps are of the overhung-shaft design, the stuffing box may be eliminated. This is important if the water is dirty. Although small cyclones can be used to clarify the water for pumps that have stuffing boxes, care must be taken to prevent leaves and other trash from blocking the gland water line. In freezing climates, special provision must be made to prevent the suction line, pump, and even the barge itself from freezing in place.

Underground mine sumps present special problems because they function not only as sumps but also as clarifiers. A well-designed sump is a good clarifier, but all too frequently

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TABLE 2 Materials of construction for mine pumps

	Neutral waters		Acidic waters		Basic waters	
	Moderate heads	High heads	Moderate heads	High heads	Moderate heads	High heads
Casing	Cast iron	Ductile iron/ cast steel	316 S.S./ alloy 20	17-4 PH	Cast iron	Ductile iron/ cast steel
Impeller	28% Cr	28% Cr	PH55A/ 17-4PH CD-4MCu	PH55A/ 17-4PH CD4-MCu	28% Cr	28% Cr
Wear rings	28% Cr	28% Cr	Same as impeller	Same as impeller	28% Cr	28% Cr
Shaft sleeve	28% Cr or 303 S.S. ceramic-coated	28% Cr or 303 S.S. ceramic-coated	316 or alloy 20 ceramic-coated PH55A, etc.	316 or alloy 20 ceramic-coated PH55A, etc.	28% Cr or 303 S.S. ceramic-coated	28% Cr or 303 S.S. ceramic-coated
Shaft	Carbon steel	High-tensile alloy steel	316 S.S./ alloy 20	17-4 PH	Carbon steel	High-tensile alloy steel
	Alloy 20	21% Cr	29% Ni	2.5% Mo		
	CD4-MCu	26% Cr	5% Ni	2.0% Mo		
	28% Cr	28% Cr	—	—		
	304 S.S.	19% Cr	10% Ni	—		
	316 S.S.	19% Cr	10% Ni	2.5% Mo		
	17-4 PH	17% Cr	4% Ni	—		Hardenable
	PH55A	20% Cr	10% Ni	3.5% Mo		Hardenable

inadequate provisions are made for cleaning the sump. If it is not cleaned at regular intervals, the loss of storage capacity may be critical in the event of a power failure. Furthermore, a sump partially filled with solids does not give the proper retention time for clarification, and the solids are directed into the pump. Although few sumps can economically be made large enough for complete clarification, it is important that a large portion of the solids be removed. This is particularly true for 3600-rpm pumps because the high-speed generally produces a high head per stage and the high differential pressure between stages causes severe wear if abrasive solids are present. Some mines use conventional thickeners and flocculating agents in an attempt to keep a high concentration of solids from reaching the pump.

Some general rules should be considered in designing sumps for underground pump rooms:

1. Attempt to get a complete analysis of the water (from another portion of the mine or from an adjacent mine if necessary).
2. Analyze the sample for corrosive properties to determine the proper materials of construction for the pump.
3. Analyze the sample for possible scale buildup in the pipeline and pumps. Check the velocity effect, if any, on the buildup rate.
4. Determine the percentage of suspended solids in the sample, its screen analysis, and the settling rate for various fractions. Determine the sump dimensions necessary for removal of all solids and then for progressively larger solids in order to select the most economical size.

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5. Compare the sump size as determined in rule 4 with the size required for physical storage capacity for *(a)* continuous pumping, *(b)* off-peak power pumping, *(c)* programmed pumping, and *(d)* storage during estimated maximum length of power interruption.
6. Calculate practical sump dimensions, considering the geologic conditions.
7. Install grit traps ahead of the sump to remove large, heavy solids. Consider methods for cleaning the grit traps.
8. Install trash screens to prevent wooden wedges, and so on from entering the sump.
9. Review sump cleaning methods and program. The best-designed sump is of no value if it is not cleaned. Compare mechanical cleaning methods with cost of parallel sumps.
10. Review the suction requirements of the pumps to be used. Because of altitude, temperature, distance from low water level to pump centerline, and suction line loss, the available NPSH may be inadequate for even an 1800-rpm pump. If a decision as to pump size, type, and speed has been made and an NPSH problem does exist, a decision must be made either to use low-speed booster pumps or to lower the pump room level to below the sump level. From a safety standpoint, the use of a booster pump is preferable, although it does add another piece of equipment.
11. Where the storage capacity is inadequate to meet possible power failures, consider either vertical pumps—possibly up to 100 ft (30 m)—for the shaft bottom pumping up to the main pump station level, or sealed pump rooms that can operate over wide variations in the sump level from a 15-ft (4.6-m) suction lift to a positive head of several hundred feet.
12. Determine the final design based on a compromise between the mine engineer (who wants maximum output), the electrical engineer (who wants small starting load), the geologist (who wants small sump dimensions), and the mechanical engineer (who wants the most reliable and easily maintained equipment).

AUTOMATIC PUMP CONTROL

With proper instrumentation, almost all pump stations can be operated automatically. Remote monitoring is simple, relatively inexpensive, and can provide safe operation and signaling of nearly all operating conditions. Equipment is presently available to measure, record, and transmit the operating conditions to remote locations. The proper equipment can thus relieve worry about the operation of the facility even if it is many miles from the operation's headquarters. Automatic control can be a simple float switch or a pressure switch, or it can be sufficiently complex to provide reliable operation under the most critical or adverse conditions. Automatic control can provide greater reliability, and its cost can depreciate over only a few years. Furthermore, the automatic recording of flow rate, flow totalizing, and periods of operation provides valuable data for analyzing the performance of the pumping installation as well as the possible cost savings in pumping during off-peak power periods.

Where the safety of a mine is dependent on the reliable operation of the dewatering pumps and controls, the following minimum requirements should be considered:

1. There should be a sump level alarm for high water, both local and remote (at the surface).
2. Sump level control should be dependable. For example, electrodes are generally unreliable in waters that leave a conducting film.
3. The control should be programmed where more than one pump is installed. However, the use of an alternator is not always desirable because all pumps are exposed to the same degree of wear. It is preferable to have one standby pump programmed through a sequence selection switch to operate at least once per week.

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4. Pump priming should be positive. Hydraulic devices should be combined with electric controls so complete dependence is not on the electric control. The presence of water in the pump should be detected to prevent the starting of a dry pump.
5. A delay circuit should be provided to ensure complete priming.
6. The control should provide for at least three starting attempts (unless an overload has occurred).
7. The priming time should be limited (if under a suction lift system).
8. The control should provide for a restart in the event of a false loss of prime on start-up (suction lift system).
9. Pump and motor bearings should have thermostats to stop pumps in the event of bearing failure.
10. Vibration monitoring may be important. This is particularly true for vertical pumps.
11. Pressure controls should indicate normal pressure and fail-safe in the event of a loss of pressure (broken column line, and so on).
12. Flow indication (check valve flow switch) is needed to signal a shaft failure.
13. Remote indication (generally at the mine office) should provide at least an indication of operation and signal pump failure or high water. More detailed information may be transmitted.
14. For long distances, investigate the use of carrier-current indication schemes, together with signal multiplexing, and so on.
15. Provide a method to test the control and alarm system.

DRIVERS

Open-pit mines use electrically driven mining equipment, such as drag lines and shovels, and the availability of power has permitted the use of electrically driven pumps. There are still many gasoline-driven or diesel-driven units, but the convenience of electric power, particularly for automatically controlled units, has increased the trend to electric drive. The availability of reliable high-voltage cable has made portable high-voltage equipment safe and economical. Pumps in open-pit service are seldom provided with sophisticated control or drive mechanisms. The primary requirements are reliability, portability, and wear resistance.

In locations where rainfall may be heavy and there is danger of power failure, a combination of electric drive and engine drive is used. The engine can be direct-coupled to the pump through a motor with a double-extended shaft or with a clutch between the engine and the motor. Automatic control is simple and reliable.

Although some steam-driven pumps still exist in underground service, their number is rapidly decreasing. Electric motor drive is the simplest for automatic control. Variable-speed units, however, are seldom used in underground service because the ratio of static head to total dynamic head is quite high. Thus the frictional loss is not a large percentage of the total head loss and not much advantage is gained by variable speed. The solution is usually a multiple-pump installation. This must be designed with care because it is possible to raise the frictional head to a point where an additional pump produces little additional capacity. Multiple discharge lines are the answer and are frequently used for safety reasons. In normal service, all discharge lines are used in parallel, although conservative design allows each line to handle the required capacity.

As with all pumping installations, a complete set of system-head curves must be prepared to analyze the power requirements under all conditions.

Motor enclosures are important in underground service. Because of the high humidity, special insulation (epoxy encapsulated, and so on) should be specified. Dripproof enclosures are the minimum requirement, with weather-protected Type I the preferred construction. Heaters should also be provided. Screens should be installed to prevent the

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entrance of rats. Winding temperature detectors, bearing thermostats, and ground-fault detectors are recommended in mine service and should be incorporated in the pump-control and alarm circuits.

Although the starting torque of a centrifugal pump is low, the available torque may have to be checked in some cases. A normal-torque motor should be suitable for pumps in the range of 500 to 3000 (10 to 60) specific speed. High-specific-speed pumps, however, have the highest power at shutoff, and it will be necessary to examine the starting arrangements for such pumps.

Starting a pump against a long empty pipeline may present overload problems, and repeated starts may be necessary. In such cases, the number of permissible starts per hour should be checked. Winding temperature detectors are important in such applications.

Although reduced-voltage starters may be required in some instances, most modern mines have electrical facilities designed for across-the-line starting. This is preferable because the starting equipment is cheaper. Before deciding on a reduced-voltage starter, the effect of the starting load on the transformer and line impedance should be checked. It may be that the voltage drop will eliminate the requirement for reduced-voltage starters. On the other hand, the effect on the primary side should be checked so the voltage drop is not so large as to drop out other equipment.

Synchronous-motor drives are seldom used unless they are large (generally at least 1000 HP) (750 kW) and then only if they are in relatively continuous service. Under such conditions, they can be operated under "leading current" conditions for power factor correction. Smaller installations frequently provide capacitors at the pump installation to provide the necessary correction for a particular installation.

Surge protection from lightning should not be overlooked. Some locations are particularly susceptible to lightning damage, especially to long surface lines. Lightning arresters should be provided at the surface, and surge arresters should be mounted at the motor location.

REFERENCES

1. American National Standard for Centrifugal Pumps for Design and Application, ANSI/HI 1.3-2000, Hydraulic Institute, Parsippany, NJ, www.pumps.org.

SECTION 12.12
MARINE PUMPS

WILLIAM J. SEMBLER

Pumps used in marine shipboard applications, both commercial marine and Navy, can typically be divided into several groups. These groups include pumps associated with a vessel's propulsion, pumps used with the generators that produce electricity, pumps used in ship's service systems, pumps used to provide hotel services for crew and passengers, and pumps that are used in cargo or other specialized systems.

Because marine pumps must operate on a moving platform, they should be designed to withstand dynamic loads resulting from vessel motion (for example, pitch, roll, and so on). In addition, they must often operate in a hot, humid, and potentially corrosive environment. In addition, marine pumps must frequently be suitable to operate with a range of flow rates to accommodate anything from operation of the vessel at full speed to operation in port with the propulsion equipment secured. Furthermore, the minimization of size (especially the required deck space) and weight is always important when designing marine equipment. For this reason, many shipboard pumps are mounted vertically (Figures 1 and 2), and smaller units are frequently furnished in a close-coupled configuration (Figure 3) with the pump's rotating parts mounted directly on the driver's shaft. To enable them to stand freely under pitch and roll conditions, vertically mounted shipboard pumps often have larger bases than comparable shore-side units. Typical materials used in the construction of marine centrifugal pumps are listed in Table 1.

A description of the features typically incorporated into the designs of pumps used in selected shipboard applications follows. This information is general in nature, however, and may not apply in all cases based on the requirements for specific installations or the preferences of vessel owners and designers.

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1 Casing
2 Impeller
3 Casing wearing ring
4 Impeller wearing ring
5 Shaft
6 Shaft sleeve
7 Packing
8 Gland
9 Thrust bearing
10 Line bearing
11 Bearing housing
12 Pump base
13 Motor bracket
14 Coupling

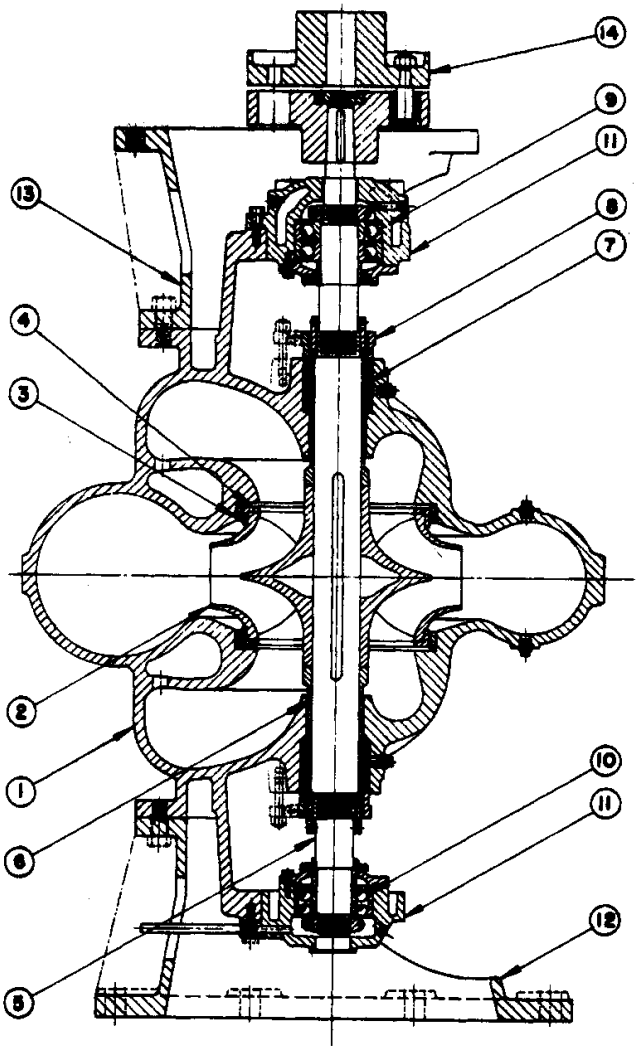


FIGURE 1 Vertical between-bearings centrifugal pump (Flowserve Corporation)

PROPULSION APPLICATIONS

Steam-Turbine-Propelled Vessels

FEED PUMPS A main feed pump is used to return water to a steam-powered vessel's boilers or steam generators. On a typical steam ship with fossil-fueled boilers, a main feed pump takes suction from a deaerating feed tank (DFT) and discharges feedwater to the steam drum in each of the vessel's boilers. In many cases, a main feed pump's discharge is connected to two separate lines that both lead to the boilers: a main feed line and an auxiliary feed line. In addition, the feedwater usually passes through one or more heaters before entering the steam drums. Although a single feed pump is frequently sized to handle a vessel's full-load requirements, some ships have multiple partial-capacity feed pumps that operate in parallel. Additional pumps are ordinarily provided for standby duty.

Typical feed pump configurations include single- and two-stage centrifugal pumps that are close-coupled to steam turbines (Figure 4) and multistage flexibly coupled pumps that are driven by steam turbines or electric motors. Although flexibly coupled feed pumps often have cast axially split volute-type casings (Figure 5), barrel pumps with diffusers and forged cylindrical casings are sometimes used. Turbine-driven feed pumps are usually mounted horizontally. Motor-driven feed pumps, however, have been used in both horizontal and vertical configurations.

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7450	Coupling guard
3261	Bearing cover, driver side
3712	Bearing lock nut
4595	Lock washer
3132	Adapter
4542	Bearing-cover gasket
3262	Bearing cover, pump side
4300/2	Lip seal
6573	Gland stud
4120	Gland
4610/2	Inner O-ring
4113	Cooling cap
1222	Stuffing-box cover
4521	Casing gasket
6710	Impeller key
1111	Casing
4510	Impeller-lock-washer gasket
4552	Impeller-nut gasket
3170	Pump base
2912	Impeller nut
6543	Impeller lock washer
1500	Casing wearing ring
2200	Impeller
1521	Stuffing-box-cover wearing ring
2450	Shaft sleeve
4133	Throat bushing
4134	Lantern ring
4610/1	Outer O-ring
4130	Packing
6731	Shaft-sleeve key
2540	Slinger
3042	Line bearing
3853	Grease nipple
2100	Pump shaft
3133	Bearing housing
3041	Thrust bearing
3853	Grease nipple
6742/2	Pump coupling key
7112	Coupling
3160	Mounting frame
6742/1	Motor coupling key
8010	Motor

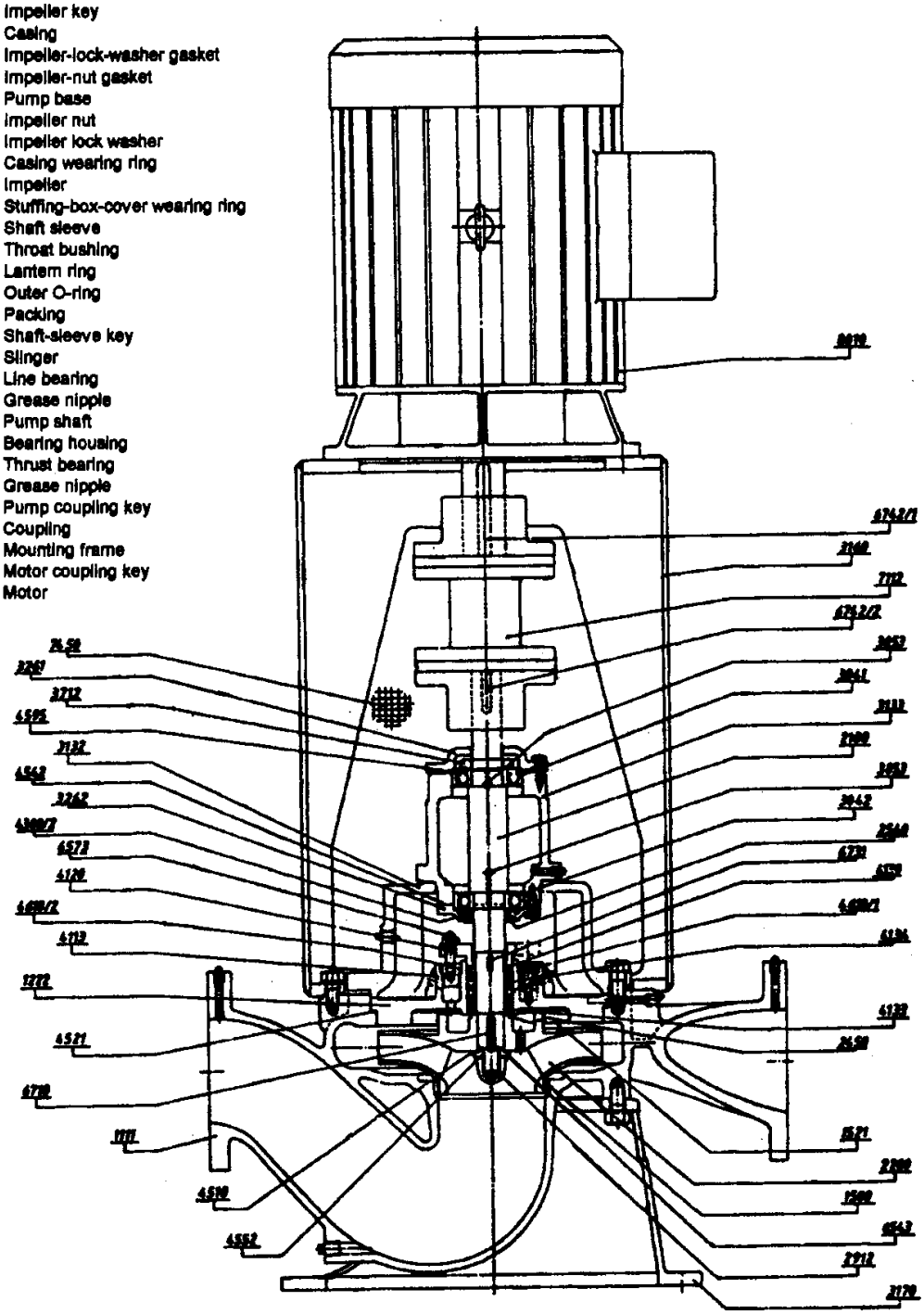


FIGURE 2 Vertical back-pull-out centrifugal pump (Flowserve Corporation)

1	Casing and suction head	7	Seal cage
2	Impeller	8	Shaft
3	Casing wearing ring	9	Slinger
4	Impeller nut	10	Shaft-sleeve seal ring
5	Shaft sleeve	11	Packing
6	Gland		

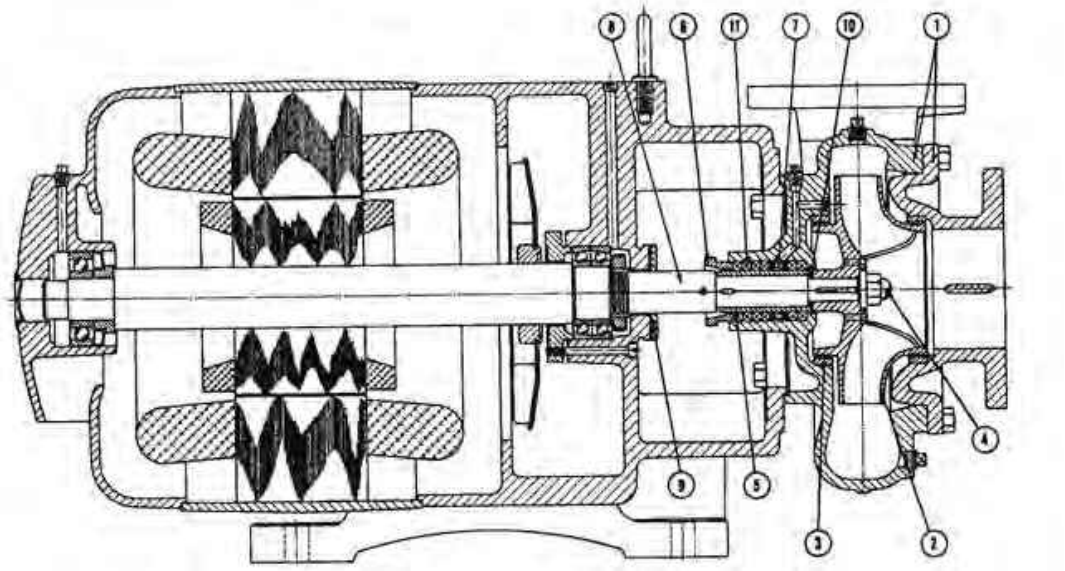


FIGURE 3 Horizontal close-coupled centrifugal pump (Flowserve Corporation)

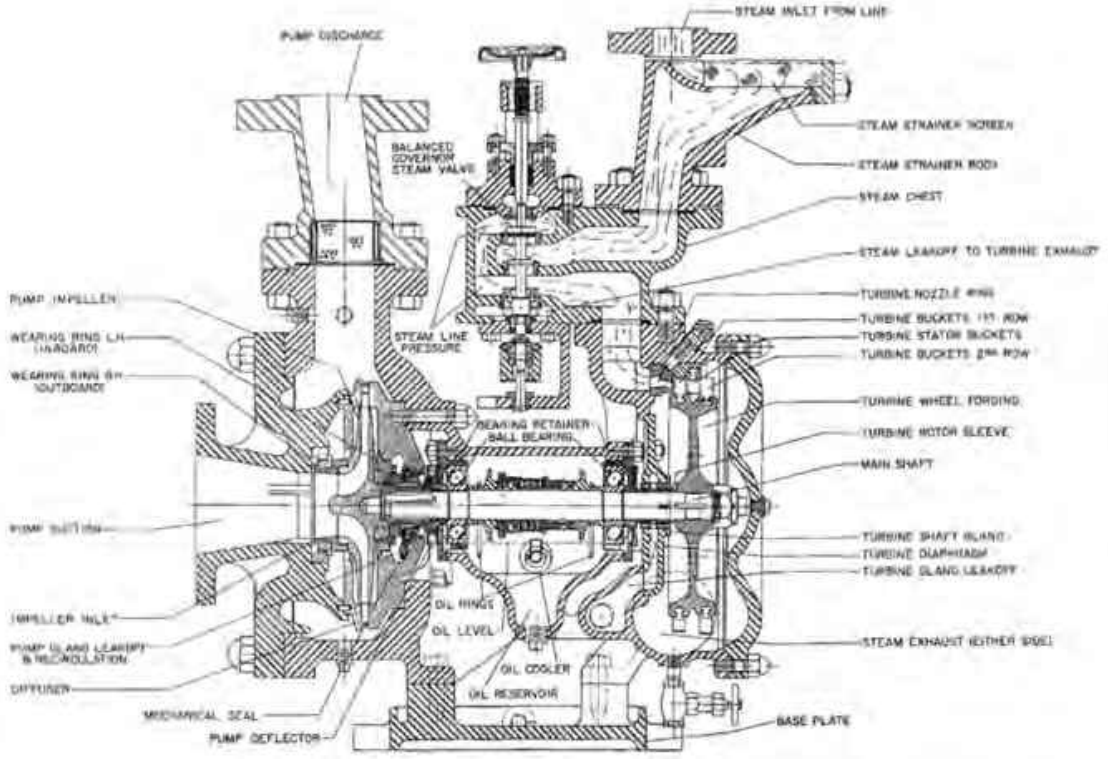


FIGURE 4 Close-coupled single-stage centrifugal main feed pump (Coffin Turbopump)

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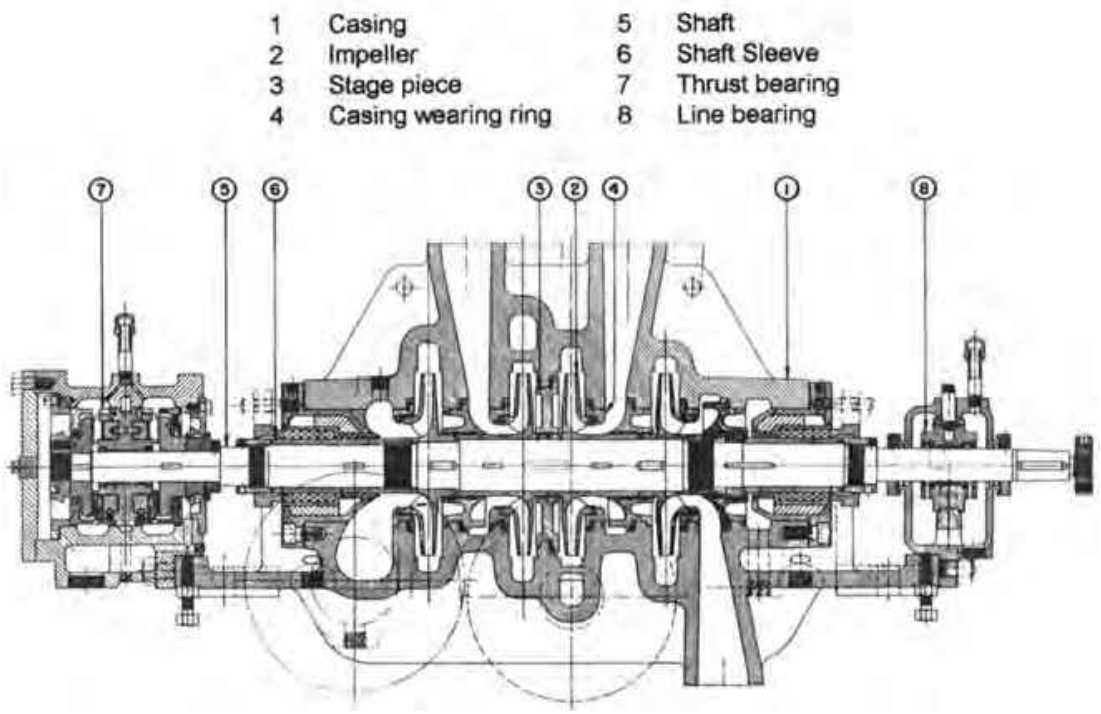


FIGURE 5 Flex-coupled, multistage centrifugal main feed pump (Flowserve Corporation)

Motor-driven feed pumps may have grease-lubricated external bearings. Bearings in a turbine-driven feed pump, however, are usually lubricated with oil that is circulated by a shaft-driven rotary pump. Oil discharged by this pump also ordinarily lubricates the turbine's bearings and may be used in the pump's governor. A strainer, filter, and seawater-cooled heat exchanger are often installed in the lubricating-oil system. The lubricating oil used with the turbine-driven feed pumps on many steam-turbine-propelled vessels is the same grade of oil used in the main propulsion lubricating-oil system.

Although packed stuffing boxes are used to seal shaft penetrations in many marine feed pump casings, to reduce stuffing-box leakage, some feed pumps are fitted with mechanical seals. In addition, a condensate-injection shaft seal consisting either of a nonrotating labyrinth-type fixed breakdown bushing or a series of spring-loaded floating rings that are stacked axially is sometimes used. When a feed pump that has a condensate-injection shaft seal is operating, cool condensate is injected into the seal and fills the close radial clearance between the nonrotating seal parts and a rotating shaft sleeve. A small portion of this condensate may flow into the pump. The remainder, however, flows outward and enters a collection chamber that is usually piped back to the vessel's gland-exhaust condenser.

During constant-speed operation, the capacity of water delivered to a boiler by a main feed pump is typically controlled by the throttling action of an automatic feedwater-regulating valve. However, when steam is used to drive a feed pump, the amount that the regulating valve must be throttled is often reduced by controlling the amount of steam supplied to the pump's driver and, therefore, the pump's operating speed with either a constant-pressure or a constant-differential-pressure governor.

A constant-pressure governor automatically regulates a feed pump's operating speed to maintain a constant pressure at the pump discharge. If there is a reduction in the boiler load and the feedwater-regulating valve begins to close, the feed pump's discharge pressure will rise and the capacity of feedwater that the pump delivers to the boiler will be reduced. The constant-pressure governor, however, will sense the rise in discharge pressure and will reduce the pump's speed until this pressure returns to the desired value. As a result of the speed reduction, the amount that the regulating valve must close to limit the feedwater flow rate is reduced. An increase in boiler load has the opposite effect.

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A constant-differential-pressure governor (sometimes referred to as an excess-pressure governor) regulates a steam-driven feed pump's operating speed to maintain a set difference between the pump discharge pressure and the pressure on the boiler-side of the feedwater-regulating valve. Because changes in the feed flow entering the boiler result, primarily, from variations in pump speed, the throttling action of the feedwater regulating valve is greatly reduced.

A relief valve is often installed on the discharge side of a main feed pump to protect the feed system from overpressurization. In addition, to prevent a feed pump from operating with too low a capacity, which could occur when the boiler load is low, a recirculation line is typically connected from the pump discharge to the DFT. An orifice is typically installed in the recirculation line to limit flow and to reduce the pressure of the water being recirculated to match the pressure in the DFT. A valve that can be closed during high-load operation is also frequently mounted in a feed-pump recirculation line.

Steam turbines that are used to drive feed pumps are generally protected with low-lubricating-oil-pressure, overspeed, and high-turbine-exhaust-pressure trips. In addition, a low-suction-pressure trip is sometimes provided to prevent a feed pump from operating with too low a suction pressure, which can result in cavitation, overheating, and a loss of load on the pump's driver.

In addition to the main feed pumps, a smaller capacity feed pump is also installed on some steam-powered vessels for use in port or during emergencies. Steam-driven, direct-acting piston-type (Figure 6) and motor-driven plunger-type reciprocating pumps, either vertically or horizontally mounted, are often used for in-port feed service.

Because the feedwater removed from a DFT is normally at its vapor pressure, the net positive suction head (NPSH) available to a main feed pump is essentially equal to the elevation of the water level within the DFT above the feed pump, less losses in the feed-pump suction line. On a vessel where the elevation of the DFT is not sufficient to provide an adequate NPSH to the main feed pumps, separate electric-motor-driven centrifugal-type booster pumps are typically installed between the DFT and the main feed pumps. The booster pumps, which operate in series with but at a much lower speed than the main feed pumps, raise the pressure of the feed water entering the main pumps and, therefore, reduce the potential for cavitation.

MAIN CONDENSATE PUMPS A typical main condensate pump takes suction from the hotwell in a main condenser and discharges condensate, through various heat exchangers, to a deaerating feed tank (DFT). Vertically mounted centrifugal pumps with two or more stages are frequently used in this application. Although many of these pumps are driven by electric motors, some main condensate pumps are driven through reduction gears by steam turbines. Two condensate pumps are generally provided for each main condenser, with each pump sized to handle full-load requirements.

A typical two-stage condensate pump (Figure 7) is fitted with grease-lubricated ball bearings at the upper end of its shaft to absorb both axial and radial loads. In addition, an internal water-lubricated radial sleeve bearing is often installed between the two impellers. The first-stage impeller is usually mounted near the lower end of the shaft, which increases its submergence. In addition, its suction eye is directed upward, which enables the impeller to be self-venting. To help facilitate the removal of any air that may enter the pump, a vent line is ordinarily connected from the suction side of a condensate pump's casing to the upper portion of the condenser. The second-stage impeller is ordinarily mounted near the upper end of the condensate-pump shaft with its suction eye directed downward. With this orientation, the hydraulic axial thrust applied the second stage impeller opposes the axial thrust acting on the first-stage impeller, and the net axial load that must be absorbed by the pump's thrust bearing is reduced. In addition, condensate at the base of the shaft seal has already passed through both impellers and is, therefore, at an elevated pressure. This helps to prevent air from being drawn into the pump through the shaft seal. The effectiveness of the shaft seal, which can consist of a packed stuffing box or a mechanical seal, is also frequently increased by injecting pressurized condensate recirculated from the pump's discharge into the seal area.

The condensate removed from a condenser's hotwell is normally at or close to its vapor pressure. Consequently, the NPSH available to a condensate pump is essentially equal to

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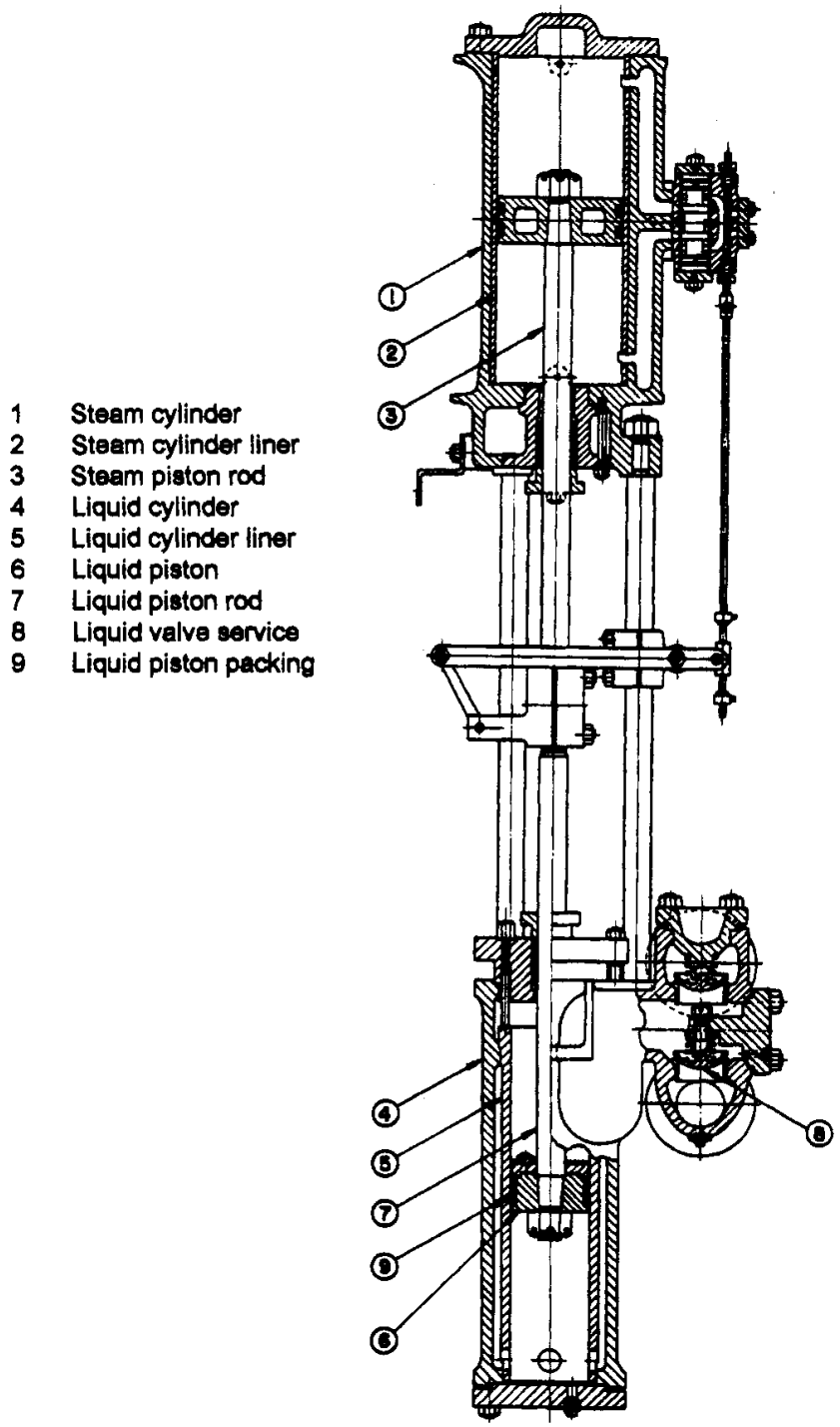


FIGURE 6 Simplex direct-acting reciprocating in-port feed pump (Flowserve Corporation)

the height of the water level in the hotwell above the pump's first-stage impeller, which is seldom more than a few feet, less frictional losses in the suction piping. To help increase the NPSH available to main condensate pumps, they are ordinarily installed as low in a vessel as is practicable. They also frequently have special low-NPSH first-stage impellers.

If a condensate pump is driven by a steam turbine or by a variable-speed electric motor, its operating speed can be adjusted with plant load. This allows the capacity at which condensate is removed from the hotwell to match the rate at which steam is exhausted into

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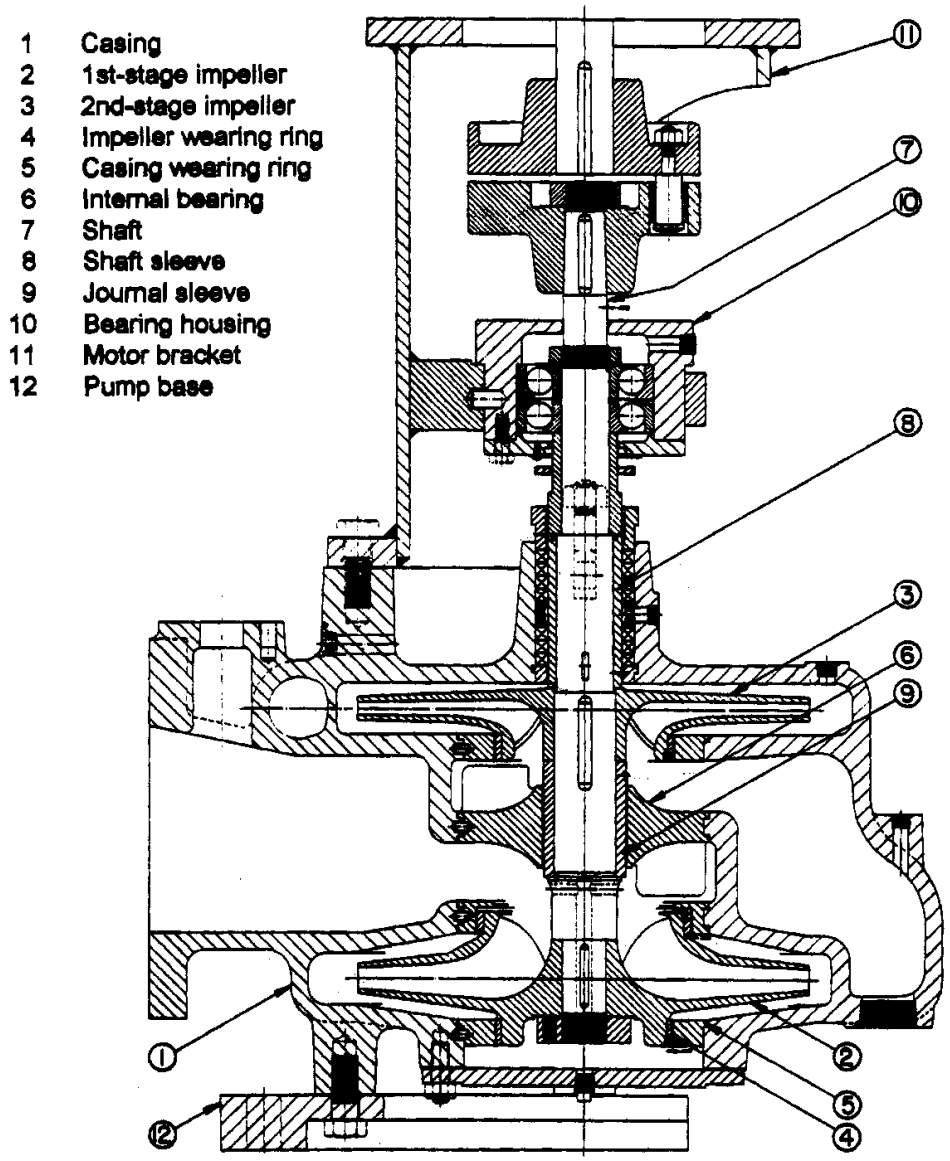


FIGURE 7 Two-stage centrifugal main condensate pump (Flowserve Corporation)

the condenser. This is usually accomplished by means of water level control. Alternatively, when a constant-speed driver is used, the capacity delivered by a main condensate pump is sometimes regulated by cavitation, which is referred to as submergence control. When submergence control is utilized, a reduction in plant load will typically result in a reduction in the main-condenser hotwell level and, therefore, in the NPSH available to the condensate pump. After the NPSH available drops below the condensate pump's NPSH requirement, the capacity being removed from the hotwell will be reduced by cavitation within the pump. As the hotwell level continues to drop, the amount of cavitation occurring will increase, and the pumped capacity will continue to be reduced until it eventually matches the rate at which steam is entering the condenser. When this occurs, the water level in the hotwell will stabilize at an elevation where the NPSH available to the condensate pump is equal to the pump's NPSH requirement at the new reduced capacity being pumped. Pumps designed to operate with submergence control need to be of robust construction and low-energy level (per stage) to prevent damage from cavitation and cavitation-induced vibration.

To avoid operation with excessive cavitation, the capacity delivered by a constant-speed condensate pump can be regulated by throttling the pump's discharge valve as

needed to maintain a constant condenser-hotwell level. However, when steam-jet air ejectors are used to deaerate a condenser, condensate-pump operation at too low a capacity can result in insufficient cooling in the ejector inter- and after-condensers. In addition, low-capacity operation can lead to surging caused by suction and discharge recirculation within a condensate pump. To prevent these problems from occurring and to reduce the need to throttle a condensate pump's discharge valve, a recirculation line back to the condenser is frequently used. This line is connected to a tee in the condensate-pump discharge piping, downstream from any air-ejector and gland-exhaust condensers. With this arrangement, if the hotwell level drops, a valve in the recirculation line can be opened, either manually or automatically, to permit a portion of the water discharged by the condensate pump to be returned to the condenser. The hotwell level can be maintained, therefore, at a value that is sufficient to suppress cavitation. In addition, because the condensate pump can always operate at or near its rated capacity, low-flow operation is avoided. Furthermore, an adequate flow of condensate through air-ejector condensers, when used, and a vessel's gland-exhaust condenser can be maintained during plant start-up and low-load operation. In some cases, the condensate-recirculation line may also be fitted with a thermostatically-controlled valve that opens and permits the condensate flow rate through the air-ejector condensers to increase when the temperature of the condensate leaving these heat exchangers exceeds a pre-set value.

CONDENSER-EXHAUSTING PUMPS Electric-motor-driven liquid-ring vacuum pumps are sometimes used instead of steam-jet air ejectors to deaerate a main condenser. The vacuum created by a liquid-ring vacuum pump is limited by the vapor pressure of the liquid compressant (water) within the pump's casing, which increases with temperature. Therefore, water separated from the gases discharged by a liquid-ring condenser-exhausting pump is generally cooled in a heat exchanger before being returned to the pump. Two pumps that are each capable of maintaining the required vacuum during full-load plant operation are frequently provided for each main condenser.

FRESHWATER-DRAIN-COLLECTING-TANK PUMPS A freshwater-drain-collecting-tank (FWDCT) pump (also sometimes referred to as an atmospheric-drain-tank pump) typically transfers condensate (collected from uncontaminated drains that are above atmospheric pressure) from a freshwater-drain-collecting tank to a deaerating feed tank (DFT). Electric-motor-driven single-stage centrifugal pumps are often used in this application. Two full capacity pumps are usually provided. With this arrangement, the on-line pump is often started and stopped automatically by a float control in the drain tank. The condensate removed from a freshwater-drain-collecting tank, which is ordinarily at a temperature of approximately 200 to 210°F (93 to 99°C), is close to its boiling point. To increase the NPSH available to a vessel's FWDCT pumps, they are frequently installed as far below the drain tank as practicable.

MAIN CIRCULATING PUMPS On a steam-powered vessel, a main circulating pump delivers seawater to a main condenser that receives steam exhausted from a propulsion turbine. In addition, a portion of the seawater discharged by a main circulating pump may be diverted to other seawater-cooled heat exchangers, such as the main lubricating-oil coolers. After passing through the main condenser or another cooler, the seawater is directed overboard. In addition to the main suction flange, which is connected to a sea chest, some main circulating pumps also have an auxiliary side-suction connection that can be used for emergency dewatering of the machinery space bilge.

Typically, one or two main circulating pumps are provided for each of a vessel's main condensers. Single-stage axial-flow propeller pumps (Figure 8) that deliver high flow rates at relatively low discharge pressures are often used in this application. In many cases, the pump is not furnished with a thrust bearing. Instead, the pump shaft is rigidly coupled to the driver's shaft, and axial loads are absorbed by the driver's thrust bearing. Radial loads applied to the pump shaft, however, are generally absorbed by a journal bearing that is located above the overhung propeller. During operation in harbors and inland waterways, the water delivered by a main circulating pump will often contain silt, sand, and other abrasives. Consequently, internal journal bearings that are lubricated by water discharged

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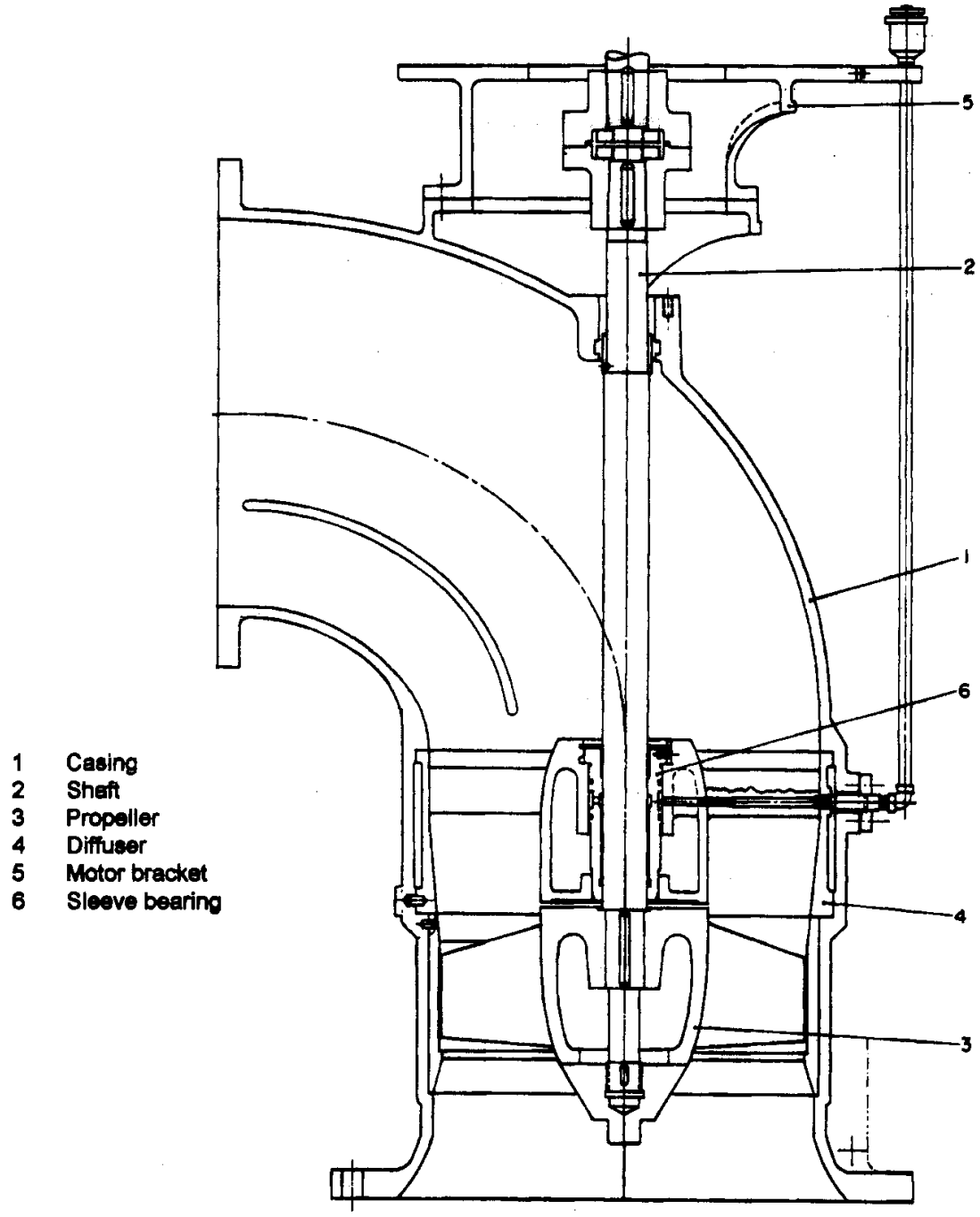


FIGURE 8 Axial-flow main circulating pump (Flowserve Corporation)

from the propeller are sometimes furnished in abrasion-resistant grades of rubber or composite materials. Alternatively, some main-circulating pump internal bearings are lubricated by grease or clean water supplied through an external connection in the pump's casing.

When higher discharge pressures are required, single-stage mixed-flow pumps with overhung end-suction impellers or single-stage radial-flow pumps with double-suction impellers mounted between bearings are sometimes used as main circulating pumps.

Main circulating pumps are generally driven either by steam turbines with reduction gears or by two-speed electric motors. This permits the capacity of seawater discharged through the condenser to be reduced during system startup and shutdown and other

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periods of low-plant-load operation by reducing the speed of the circulating pump's driver. Many steam ships are fitted with a scoop-injection system that permits seawater to be forced through the main condensers by the forward momentum of the vessel. With this arrangement, the main circulating pumps are usually stopped whenever the vessel is operating above a certain speed.

FUEL-OIL SERVICE PUMPS A fuel-oil (FO) service pump typically takes suction through either high- or low-suction ports in the FO service or settling tanks and supplies fuel oil to the burners in a steam ship's oil-fired boilers. At least two service pumps each capable of delivering the vessel's full-power fuel-oil requirement are generally provided. FO service pump suction and discharge lines are generally fitted with duplex strainers. In addition, one or more flow meters and heaters are frequently installed in a FO service system.

Horizontally and vertically mounted multiple-screw rotary pumps that are driven by either steam turbines or two-speed electric motors are often used in this application. In addition, some older ships have steam-driven reciprocating FO service pumps. When a vessel normally burns heavy (residual) fuel oil, which must be heated before being delivered to a boiler's burners, a separate electric-motor-driven rotary pump is also usually provided to supply unheated distillate fuel oil to the boilers during plant start-up.

The steam flow to a turbine-driven rotary or a direct-acting reciprocating FO service pump is often regulated with a constant-pressure governor that acts to maintain a constant FO-service-pump discharge pressure. With this arrangement, changes in the flow rate passing through the FO service pump result from variations in the pump's operating speed. When a two-speed electric motor is used to drive a FO service pump, the motor is ordinarily operated at low speed during low-plant-load periods (for example, in port) and at high speed during high-plant-load periods. However, during operation at either speed, a motor-driven FO service pump typically discharges more oil than the amount required by the burners. The excess oil is usually returned to the suction side of the FO service system through a recirculation line. The flow rate through this recirculation line is often regulated by an automatic valve that maintains a constant FO-service-pump discharge pressure. An additional recirculation line with a hand-operated valve is frequently provided to permit fuel oil to be circulated through a FO-service-system heater and a boiler's burner manifold prior to lighting-off the boiler. To prevent FO-service-system overpressurization, a relief valve should always be installed on the discharge side of each FO service pump.

Remote controls are typically provided that enable a vessel's FO service pumps to be stopped in an emergency from outside of the machinery space. In addition, flanged FO-service-pump discharge connections are ordinarily fitted with wrap-around shields to deflect spray in case of a leak. In addition, a drip pan or similar device is usually installed under a FO service pump to prevent oil that may leak out of the pump from draining into a vessel's bilge.

LUBRICATING-OIL SERVICE PUMPS A lubricating-oil service (LOS) pump removes lubricating oil from the propulsion-reduction-gear sump and discharges it to the propulsion-turbine bearings, the reduction gears and their bearings, and the main thrust bearing for each propulsion shaft. This oil is also usually directed to the inlet side of a speed-limiting governor pump mounted on the forward end of each propulsion turbine's shaft and to the propulsion-turbine throttle- or nozzle-valve controls. In addition, a portion of the pumped oil may be sent to an overhead gravity tank that holds enough oil to lubricate the propulsion machinery for several minutes following a loss of LOS-pump discharge pressure.

Most steam-powered vessels have two or three LOS pumps. Horizontally and vertically mounted multiple-screw pumps are frequently used. When mounted vertically, the pumps may be submerged directly within the lubricating-oil (LO) sump. In most cases, at least one rotary LOS pump is electric-motor-driven. Remaining units may also be motor driven, or they may be driven by steam turbines or off the propulsion shafting. In addition, steam-driven reciprocating-piston pumps are sometimes used as standby LOS pumps. In addition, some vessels have a battery-operated emergency LOS pump.

LOS pumps are typically installed low in a vessel and close to the LO sump. Suction and discharge lines are often fitted with duplex strainers. In addition, a relief valve and

coolers are generally installed in pump discharge lines. Also, one or more orifices are typically installed in a LOS system to reduce the pump's full discharge pressure, which is usually based on the requirements of the propulsion-turbine governors and controls, down to the pressure required by the bearings and gears. A pressure-controlled switch or valve that automatically starts a standby pump if the oil pressure at the discharge of the operating LOS pump drops below a preset value is often provided. Most steam-turbine-propelled vessels also have a device that stops the supply of steam to the ahead propulsion turbines in case of a LOS-system failure.

Diesel-Propelled Vessels

A propulsion diesel engine that operates below approximately 300 rpm and is directly connected to a propulsion shaft is usually classified as a slow-speed engine. Diesel engines that are connected to propulsion shafts through reduction gears and have maximum operating speeds below 900 to 1200 rpm are typically classified as medium-speed engines, and engines that operate above 900 to 1200 rpm and are used with reduction gears are generally classified as high-speed engines.

ENGINE FRESHWATER COOLING PUMPS A pump is ordinarily used to circulate freshwater through jackets in a propulsion diesel engine's cylinders and cylinder heads, the engine's turbocharger, and, on some vessels, an evaporator where the jacket water provides heat to produce fresh water. The jacket water may also pass through the engine's lubricating-oil and charge-air coolers. In addition, a heat exchanger in which the jacket water is cooled by either freshwater or seawater is frequently included in the system. Alternatively, this heat exchanger is sometimes eliminated when a vessel has a central freshwater cooling system. Instead, some freshwater from a separate low-temperature loop is admitted into the high-temperature jacket-water loop where it mixes with and cools the hot jacket water. An additional heat exchanger that can be used to preheat the jacket water prior to engine start-up is often provided as part of a main engine's jacket-water system.

The pumps that circulate cooling water through a propulsion diesel engine are frequently called jacket-water circulating pumps. Alternatively, however, they may be referred to as high-temperature freshwater cooling pumps on a vessel that has a central freshwater cooling system. Two vertically or horizontally mounted single-stage centrifugal pumps that are each sized to meet normal full-load requirements are often provided for each propulsion engine. Although each pump may be driven by an electric motor, when used with a high- or medium-speed propulsion engine, one of the pumps is often mounted on and driven off the engine. An elevated tank is ordinarily included in the system to allow for any thermal expansion of the water and to maintain a positive pressure at the pump suction.

Separate electric-motor-driven pumps may be used to circulate fresh water through a propulsion engine's pistons when they are water-cooled. Although single-stage centrifugal pumps are often used in this application, vertical turbine pumps that are submerged in the piston-cooling-water drain tank are used on some vessels. When required by the engine design, an additional pair of electric-motor-driven centrifugal pumps is provided to circulate fresh water through an independent loop that cools the main-engine fuel valves or injectors.

ENGINE SEAWATER COOLING PUMPS On some vessels, single-stage centrifugal pumps are used to supply seawater to diesel-engine charge-air, lubricating-oil, and jacket-water coolers. Two pumps that are each sized to meet normal full-load requirements are often provided for each propulsion engine. Although both seawater-cooling pumps may be driven by electric motors, when a medium- and high-speed diesel engine is used, one pump is sometimes attached to and driven by the engine.

FUEL-OIL SUPPLY AND BOOSTER PUMPS Fuel-oil (FO) pumps used for diesel engines should be suitable to handle any of the various grades of fuel that may be burned in a vessel's

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engines, which can often include both light distillates that are used during maneuvering and heavy residual oils that are used while underway at sea. When heavy fuel oil is supplied to an engine, it must generally be heated to reduce its viscosity.

A FO booster pump typically takes suction either directly from a diesel-propelled vessel's daily service tanks or from a separate mixing tank and supplies fuel oil to the main-engine injector pumps. Two multiple-screw- or gear-type fuel-oil booster pumps that are each capable of meeting full-power requirements are often provided for each propulsion engine. Both FO booster pumps may be electric motor driven. Alternatively, however, when a medium- and high-speed diesel engine is used, one FO booster pump is sometimes attached to and driven by the engine. Also, in some high-temperature heavy-fuel-oil systems, a pair of electric-motor-driven, screw- or gear-type FO supply pumps that operate upstream of and in series with the FO booster pumps are located between the service tanks and the mixing tank. More oil is typically delivered to an engine than the amount required for combustion, and the excess oil is ordinarily returned to the service or mixing tank through a recirculation line. Strainers, filters, flow meters, and heaters are also frequently installed in the FO service system.

LUBRICATING-OIL PUMPS The main-engine lubricating-oil (LO) pump typically removes lubricating oil from a sump located below a propulsion engine and discharges the oil to the engine's bearings, governor controls, turbochargers, and, when they are cooled by oil, the engine's pistons. Strainers, filters, and a cooler are also usually included in a LO system. Two main LO pumps that are each capable of delivering the oil required during normal full-load engine operation are usually provided for a propulsion engine. Vertically or horizontally mounted multiple-screw and gear pumps are frequently used in this application. Although both pumps may be driven by electric motors, one pump is sometimes attached to and driven off a medium- or high-speed engine. Alternatively, some vessels have electric-motor-driven, vertical turbine pumps that are submerged within the LO sump installed. A device that sounds an alarm following a failure of the LO system is normally provided.

When a vessel is propelled by a crosshead-type diesel engine, some of the oil discharged by the main LO pump is often directed to the suction-side of a lower-capacity rotary-type booster pump that supplies oil to lubricate the engine's crosshead bearings. In addition, a separate low-capacity rotary pump is frequently required to fill a head tank that supplies a different grade of oil to lubricators that inject the oil into each of a crosshead engine's cylinders. Also, with some designs, a separate pair of rotary pumps is required to supply lubricating oil to the engine's camshaft bearings.

In addition to the pumps that supply lubricating oil to propulsion engines, separate rotary pumps are generally used to supply lubricating oil to reduction gears and their bearings on vessels propelled by medium- or high-speed engines.

Gas-Turbine-Propelled Vessels

FUEL-OIL SERVICE PUMPS Typically, fuel oil is delivered to a propulsion gas turbine's combustion-chamber nozzles by a gear pump that is mounted on and driven by the gas turbine. In addition, a pair of two-speed electric-motor-driven rotary pumps that operate upstream of and in series with the attached pump are usually provided. Duplex strainers, filters, and heaters are also ordinarily included in a gas turbine's fuel-oil service system.

LUBRICATING-OIL PUMPS A typical gas turbine is furnished with an attached gear- or centrifugal-type lubricating-oil (LO) pump that is driven off the gas turbine. This pump takes suction from a LO reservoir and discharges synthetic oil to the bearings for the gas turbine and to control devices. Filters are also included in the LO system. In addition, electric-motor-driven gear, vane, or centrifugal pumps are ordinarily used to circulate lubricating oil through the system during start-up and cool-down periods and to serve as a backup to the attached pump. Excess oil delivered by a gas turbine's LO pump is frequently returned through a pressure regulating valve to the LO reservoir. A device is normally provided

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that sounds an alarm and, in some cases, automatically stops the flow of fuel to a gas turbine following a failure of the LO system.

Separate rotary scavenge pumps may be used to return oil that drains from a gas turbine's bearings to the LO reservoir. In addition, multiple-screw pumps are frequently used to circulate mineral oil through an independent lubrication system for a gas-turbine-propelled vessel's reduction gears and their bearings. Although one of these screw pumps may be driven off the reduction gears, the remaining reduction-gear LO pumps are usually electric motor driven. Some of the mineral oil in the reduction-gear LO system frequently passes through a heat exchanger in which it absorbs heat from the synthetic oil that lubricates the gas turbine's bearings. The reduction-gear lubricating oil also usually passes through a second heat exchanger in which it is cooled either by freshwater or seawater.

GENERATOR APPLICATIONS

Steam-Turbine-Driven Turbogenerators

AUXILIARY CONDENSATE PUMPS When the steam leaving a steam ship's turbogenerator exhausts into an auxiliary condenser, an auxiliary condensate pump is ordinarily used to remove condensate from the auxiliary condenser's hotwell and return it to the DFT. A typical auxiliary condensate pump is a vertical two-stage centrifugal pump that is similar in configuration to a main condensate pump but is smaller in size. One auxiliary condensate pump is typically furnished for each auxiliary condenser. Crossover lines are, however, often provided so each auxiliary condensate pump on a vessel can remove condensate from any of the vessel's auxiliary condensers.

AUXILIARY CIRCULATING PUMPS A single-stage centrifugal pump is often used to circulate seawater through the tubes of an auxiliary condenser that receives steam exhausted from a vessel's turbogenerator. Seawater discharged by this pump may also be directed to the generator's lubricating-oil and air coolers. One auxiliary circulating pump is typically furnished for each auxiliary condenser. Crossover lines, however, are often provided so one generator's auxiliary circulating pump can supply seawater to any of the vessel's auxiliary condensers. In addition, a line is often installed that permits seawater discharged from a vessel's auxiliary circulating pumps to be directed, in case of a main circulating pump failure, to a main condenser.

LUBRICATING-OIL PUMPS A typical turbogenerator is fitted with an internal or external gear-type lubricating-oil (LO) pump that is geared to and driven by the turbine. The pump ordinarily takes its suction from a sump located below the turbine's reduction gears and discharges lubricating oil to bearings, the turbine's reduction gears, and the governor that controls the flow of steam to the turbine. A separate hand-operated or electric-motor-driven gear pump may also be provided to enable lubricating oil to be circulated through the LO system during generator start-up and shutdown periods.

A duplex strainer, a cooler, and one or more relief valves are often included in a turbogenerator's LO system. In addition to protecting the system from overpressurization, the relief valves act as backpressure valves that maintain the desired LO pressure in various parts of the system. This arrangement permits lubricating oil sent to the governor to be at a higher pressure than the oil directed to bearings and reduction gears. A turbogenerator is ordinarily fitted with a trip that stops the flow of steam to the turbine if the LO pressure drops below a preset value.

Diesel-Driven Generators

JACKET-WATER CIRCULATING PUMPS A diesel engine that drives a generator is often furnished with an attached centrifugal-type jacket-water circulating pump that circulates freshwater through passages in the engine's cylinders and cylinder heads, the engine's

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turbocharger, and the engine's lubricating-oil cooler. In addition, the jacket-water system may include a preheater that is used to warm-up the water prior to engine start-up. When the jacket water is air cooled, it will also pass through a radiator. Alternatively, the jacket water may be cooled by seawater or freshwater in a heat exchanger. When seawater is used as the cooling medium in a jacket-water cooler, the seawater may be circulated by a centrifugal pump that is attached to and driven by the diesel engine or by a separate electric-motor-driven centrifugal pump.

FUEL-OIL PUMPS A diesel engine that drives a generator frequently has an attached rotary gear-type fuel-oil (FO) pump that is mounted on and driven by the engine. The pump typically takes suction from a daily service tank and delivers fuel oil to the engine's injector pumps. Excess oil delivered to the engine is ordinarily returned through a recirculation line to the service tank. A filter is typically included in the system. A separate hand-operated FO pump that can be used to prime the engine-driven FO pump (for example, following maintenance performed on the FO system, such as filter replacement) may also be provided. A diesel engine's FO supply line is often fitted with a valve that can be closed from outside the machinery space to stop the engine in case of a fire or other emergency.

LUBRICATING-OIL PUMPS A diesel engine that drives a generator is usually fitted with an attached rotary gear-type lubricating-oil (LO) pump that is mounted on and driven by the engine. The pump typically takes suction from a sump located below the engine and discharges oil to the engine's bearings. A filter and cooler are also ordinarily included in an engine's LO system. In addition, a hand-operated or air-motor-driven rotary-type LO pump is often provided to permit the engine's bearings to be prelubricated prior to startup. A diesel engine that drives a generator is often protected by a trip that stops the engine if the pressure in the LO system drops below a preset value.

AUXILIARY APPLICATIONS

Ship's Service Pumps

FUEL-OIL TRANSFER PUMPS A fuel-oil (FO) transfer pump is used to remove fuel oil from one or more of a vessel's FO storage tanks and to transfer the oil to settling tanks, to other storage tanks (i.e., to adjust the list, trim, or stability of the vessel), or to an above-deck connection through which the oil can be directed ashore or to another vessel. Most vessels have at least two pumps that can be used to transfer fuel oil. Although horizontally and vertically mounted multiple-screw rotary pumps are frequently used in this application, steam-driven reciprocating-piston pumps are sometimes used to transfer fuel oil. FO transfer pumps are often installed low in a vessel to improve suction conditions.

Some rotary FO transfer pumps are driven by two-speed electric motors or steam turbines. When this is the case, the capacity of fuel transferred can be changed by altering the speed of the transfer pump's driver. The capacity delivered by a FO transfer pump during constant-speed operation can often be reduced by opening a valve in a bypass line through which a portion of the oil discharged by the pump is recirculated back to the suction line. A FO transfer pump is usually protected with a suction strainer and a discharge relief valve and is often fitted with remote controls that enable the pump to be stopped from outside of a vessel's machinery spaces.

On a steam-propelled vessel with oil-fired boilers, a FO service pump typically takes suction directly from the fuel-oil settlers and discharges oil to burners in the vessel's boilers. On a diesel-propelled vessel, however, fuel oil in the settlers is ordinarily passed through one or more purifiers as it is transferred to separate daily service tanks from which the engine FO supply or booster pumps take suction. The rotary gear-type pump that frequently delivers fuel oil to a purifier is often mounted on and driven by the purifier. In some cases, a second attached rotary pump is used to boost the pressure of the clean

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oil leaving a purifier. Alternatively, independent electric-motor-driven pumps are sometimes used with FO purifiers.

FRESHWATER COOLING PUMPS To reduce corrosion from seawater, some vessels are fitted with a central freshwater cooling system. When this is the case, single-stage centrifugal pumps are generally provided to circulate freshwater through two separate cooling loops: a low-temperature loop and a high-temperature loop. Two centrifugal pumps that are each capable of delivering the full-load capacity are frequently installed in each freshwater loop. Smaller pumps may also be provided for startup and in-port use.

A low-temperature freshwater cooling pump ordinarily circulates freshwater through a vessel's various condensers, oil coolers, and air coolers. Jacket-water coolers for a vessel's auxiliary diesel engines may also be included in the low-temperature loop. In addition, freshwater in the low-temperature loop typically passes through a seawater-cooled heat exchanger. On a diesel-propelled vessel, the high-temperature freshwater cooling pump serves as the jacket-water-circulating pump and typically circulates fresh water through the propulsion-engine jackets, turbochargers, and a freshwater generator (evaporator). On some vessels, freshwater in the high-temperature loop also passes through a heat exchanger in which it is cooled by freshwater in the low-temperature loop. Alternatively, however, the need for this heat exchanger is eliminated in some central freshwater cooling systems with a control valve that allows some of the freshwater in the low-temperature loop to enter, mix with, and cool the hotter water in the high-temperature loop.

SEAWATER SERVICE PUMPS A seawater service pump takes suction from a sea chest and supplies seawater to heat exchangers in which this water serves as the cooling medium. This can sometimes include refrigeration and air-conditioning condensers, various lubricating-oil coolers, and air-compressor coolers. On newer vessels that have a central freshwater cooling system, however, the seawater cooling pumps supply seawater only to large freshwater coolers. With either arrangement, after leaving the heat exchangers that it passes through, the seawater is ordinarily directed overboard.

Two or more horizontally or vertically mounted electric-motor-driven single-stage centrifugal seawater service pumps (these pumps are also sometimes referred to as seawater cooling pumps or auxiliary seawater pumps) are usually installed on a vessel. A seawater pump is typically located low in a vessel so it can operate with a flooded suction. In some cases, the pump's suction line may be connected to both a lower sea chest and an upper sea chest. The use of the lower sea chest, which often receives seawater through an opening in the bottom of a vessel's hull, reduces the potential for air to enter a seawater pump's suction line (especially when the vessel is rolling in rough seas). However, when the vessel is in shallow water, using the upper sea chest, which is typically connected to an opening in the side of a vessel's hull, can reduce the amount of silt, mud, and other contaminants entering the seawater-pump suction. In addition to the sea-chest connections, on a diesel-propelled vessel, the largest seawater-cooling pump may also have an emergency bilge suction line through which the machinery space can be dewatered in case of flooding.

To help protect a seawater pump from foreign material that may enter a sea chest, a strainer is frequently installed in a seawater pump's suction line. In addition, when a mechanical-seal flushing line is used with a seawater pump, it is sometimes fitted with a cyclone abrasive separator. In a typical installation, a recirculation line from the discharge side of the pump's casing is connected to the inlet on the side of the separator. Clean water leaving the top of the separator is directed to the mechanical seal's flushing connection, which is usually in the seal's gland. Dirty water discharged from the bottom of the separator is returned to the suction side of the pump's casing.

LUBRICATING-OIL TRANSFER PUMPS Rotary gear pumps are frequently used to transfer lubricating oil from a vessel's lubricating-oil (LO) storage tanks to tanks in various locations throughout the vessel where the oil is stored for use in auxiliary machinery. In addition, a rotary pump may be used to add lubricating oil directly to oil-lubricated machinery.

Most vessels have one or more LO purifiers that can be used to remove water and other contaminants from the lubricating oil contained in machinery LO sumps (for example, LO sumps for diesel engines, turbogenerators, and main-propulsion-turbines). A LO purifier

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may also be used to transfer lubricating oil from a vessel's LO storage tanks to separate LO settling tanks. A rotary gear pump is often used to deliver lubricating oil to a LO purifier. In many cases, the pump is mounted on and driven by the purifier. A second attached rotary pump may also be used to boost the pressure of the clean oil leaving a purifier. Alternatively, independent electric-motor-driven pumps are sometimes used with LO purifiers.

STERN-TUBE LUBRICATING-OIL PUMPS An electric motor-driven rotary screw, gear, or vane pump is required to circulate lubricating oil through a vessel's stern-lube bearings and seals when these components are oil lubricated. One pump is typically provided for each propulsion shaft. The pressure on the discharge side of a stern-tube lubricating-oil (LO) pump is ordinarily established by the elevation of a head tank that is connected to the system. (Stern-tube LO pumps are not required on vessels that have seawater-lubricated stern-tube bearings and seals.)

AUXILIARY AND EXHAUST-GAS (WASTE-HEAT) BOILER PUMPS Diesel-propelled vessels are frequently fitted with exhaust-gas boilers in which heat contained in the engine exhaust-gas is utilized to generate steam for various purposes, such as heating fuel, liquid cargo, and water. On some vessels, a portion of the steam produced may even be superheated and supplied to turbine-driven equipment, such as turbogenerators. To permit steam to be generated during periods of low engine load or when an engine is secured (for example, in port), most of these vessels are also fitted with auxiliary oil-fired boilers that can operate alone or parallel to the exhaust-gas boilers.

In a typical installation, a feed pump is used to return condensate from a drank tank to the boilers. In addition, a separate pump is generally required to circulate water through the boiler generating tubes. Furthermore, when condensing turbines are installed in an auxiliary steam system, a condensate pump is used to transfer condensate from the condenser's hotwell to the drain tank. Electric-motor-driven, single and multistage centrifugal and regenerative turbine pumps are typically used in these applications.

A fuel oil (FO) service pump is usually required to supply fuel oil to an oil-fired auxiliary boiler. Electric-motor-driven gear pumps are often used in this application.

HYDRAULIC-SYSTEM PUMPS Positive-displacement pumps are ordinarily used to pressurize and circulate the hydraulic fluid in a vessel's hydraulic systems. Typical examples of shipboard hydraulically powered machinery include steering gear, anchor windlasses, cranes, hatch covers, and valves. The hydraulic pumps, which can be installed in a self-contained system that is an integral part of the hydraulically powered component or in a larger central system that provides fluid to all of the hydraulic equipment on a vessel, are usually driven by electric motors or diesel engines.

In a constant-pressure hydraulic system, one or more variable-displacement rotating-piston pumps are often used to supply hydraulic liquid at essentially a constant pressure to the components that are powered by the system. If the capacity delivered by a pump exceeds the system's demand, the pump discharge pressure will include and an automatic controller will reduce the pump's stroke and the capacity pumped until the discharge pressure returns to the set point. If the system demand increases and the pump discharge pressure drops, the pump's stroke and the capacity delivered will be increased. The pump stroke and the pumped capacity are, therefore, automatically adjusted to match the demand on the hydraulic system. In some systems, hydraulic pumps may also be automatically started and stopped as needed.

The hydraulic fluid in a constant-flow variable-pressure hydraulic system is typically circulated by gear, vane, multiple-screw, or fixed-displacement rotating-piston pumps. These pumps, which often operate continuously, take suction from a sump or tank and deliver pressurized oil to the components powered by the system. If the capacity delivered by a system's pumps exceeds the system requirement, excess oil is recirculated through an unloading valve back to the sump.

FIRE PUMPS A fire pump takes suction from a vessel's sea chest and discharges seawater through a fixed fire main to hydrants located throughout the vessel. Most vessels have at least two fire pumps that are each capable of delivering a minimum capacity of seawater

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simultaneously to a specified number of the most remote hydrants at a specified pressure. Although fire pumps are sometimes used for other purposes, at least one fire pump on a vessel should always be available to provide water to the fire main. In addition, some of a vessel's fire pumps, together with their sea connections and sources of power, are ordinarily installed in separate spaces and arranged so a fire in any one location cannot incapacitate all of the vessel's fire pumps. Also, controls are sometimes provided that enable a fire pump's suction and discharge valves to be opened and the pump to be started from outside of the machinery space. A pressure gage and a relief valve are usually installed in a fire pump's discharge line. (See Section 12.4.)

Single-stage and, in some cases, two-stage centrifugal pumps may be used in fire service. Vertically mounted centrifugal fire pumps are generally driven by electric motors. Horizontal fire pumps, however, can be driven by electric motors, steam turbines, or diesel engines. In addition, smaller centrifugal fire pumps and the electric motors that typically drive them are sometimes furnished in a close-coupled configuration. A centrifugal fire pump is ordinarily installed low in a vessel so it operates with a flooded suction. Alternatively, a vertical turbine pump is sometimes used in this application, which enables the fire-pump impellers to be submerged within a tank.

Some vessels also carry portable fire pumps that can be moved throughout the vessel by the crew. A typical portable unit consists of a single-stage end-suction centrifugal pump that is close-coupled to a gasoline engine. An integral vacuum priming pump is often included with a portable unit to enable the fire pump to be used in areas of the vessel that are above the waterline.

BILGE PUMPS Bilge pumps remove liquid from machinery-space bilges, tank tops, the shaft alley, and watertight compartments located throughout a vessel. Fluid discharged from a bilge pump is typically directed, depending on its content and applicable regulations, to an oily-waste holding tank or overboard. Most vessels have more than one bilge pump. In addition to a common suction main that is connected to branches that lead to suction wells (sometimes referred to as rose boxes) located throughout a vessel, one or more independent bilge-pump suction lines are generally provided in machinery spaces. In addition, a vessel's bilge pumps, together with their sources of power, may be installed in various watertight compartments throughout a vessel so the flooding of one compartment will not incapacitate all of the vessel's bilge pumps. Alternatively, some vessels have an emergency bilge pump that is suitable for use even when the pump is submerged. The submersible bilge pump, which is normally powered through the vessel's emergency switchboard, can normally be operated remotely from outside of the vessel's machinery spaces.

Horizontally and vertically mounted centrifugal pumps are frequently used for bilge service. Centrifugal bilge pumps are frequently driven by electric motors. Some bilge pumps, however, are driven off a vessel's propulsion machinery. Because bilge pumps frequently operate with a suction lift, a non-self-priming centrifugal bilge pump is usually connected to a vacuum-priming pump. The vacuum pump can be part of a central system that is used to prime multiple pumps on a vessel (Figure 9), or it may be a dedicated unit that is used only with the bilge pump. When a dedicated vacuum pump is provided, it is sometimes electric-motor-driven and mounted on the same baseplate as the bilge pump that it primes. Alternatively, a dedicated vacuum pump may be mounted on and driven directly off the bilge pump.

To eliminate the need for a separate priming pump, self-priming centrifugal pumps are sometimes used in bilge service. A typical self-priming centrifugal pump (Figure 10) has a casing with an enlarged suction chamber that retains liquid whenever the unit is stopped. When the pump is started with air in its suction line, the liquid in the suction chamber is pumped through the impeller and enters a discharge chamber installed at the top of the casing. The evacuation of the suction chamber creates a vacuum that draws air from the suction line into the pump. The liquid in the discharge chamber is returned to the pump's impeller through either an internal port or an external recirculation line and mixes with this air. The liquid-and-air mixture is then pumped through the impeller and into the discharge chamber where the air is vented from the pump. The liquid is again returned to the impeller, and the cycle is repeated. Each time the stored liquid is recirculated through the pump, an additional amount of air is removed from the suction line.

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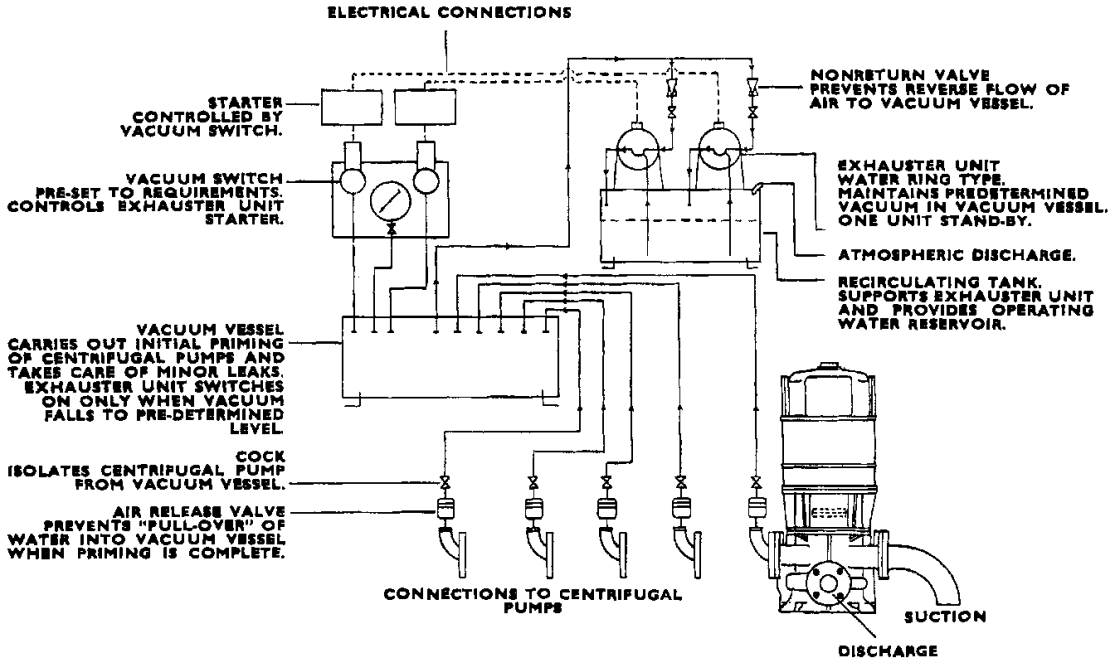


FIGURE 9 Central priming system (Flowserve Corporation)

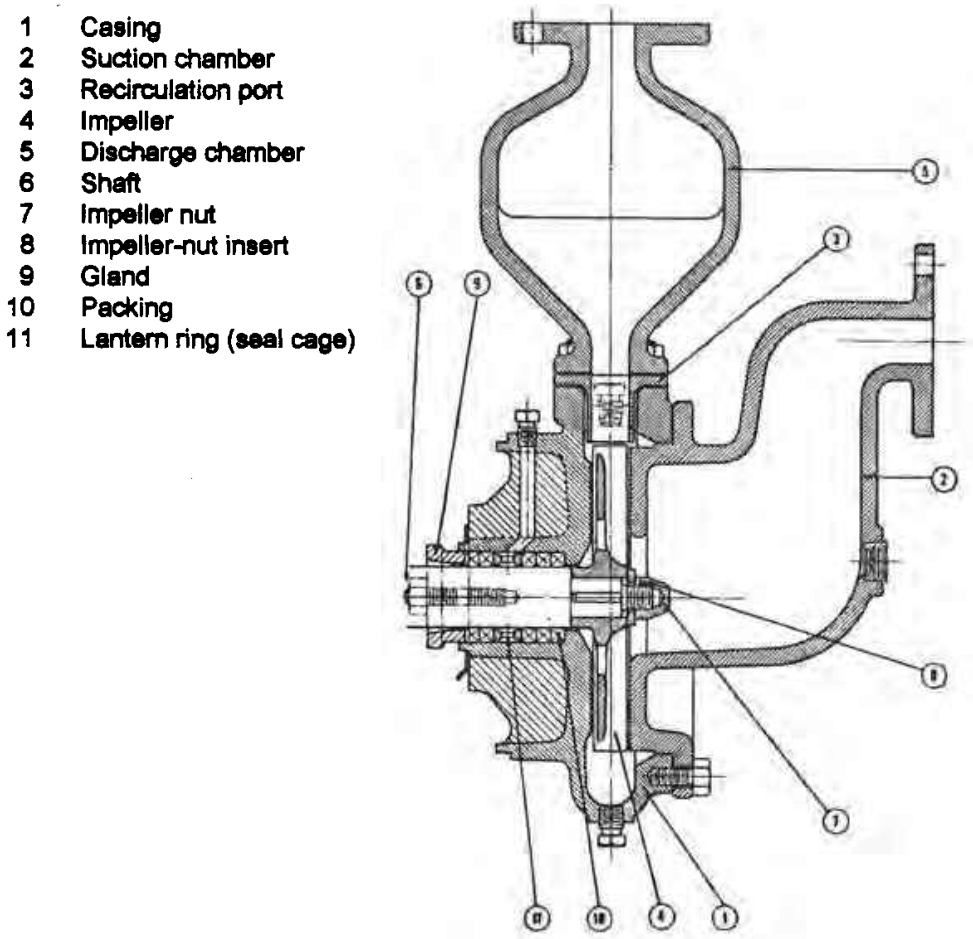


FIGURE 10 Self-priming centrifugal pump (Flowserve Corporation)

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After this occurs, the pump will discharge liquid in a normal fashion. However, priming times can be excessive when large amounts of air must be removed from a suction line; consequently, self-priming centrifugal pumps are usually only used for bilge service when the length of the bilge suction piping is relatively short.

Additional types of bilge pumps that do not require priming pumps include motor-driven sump pumps that are submerged within the bilge drain wells; rotary-type vane pumps; air-operated diaphragm pumps; and motor-, steam-, or air-driven piston-type reciprocating pumps.

An oily-water separator (OWS) is used to remove oil from bilge water so the water can be discharged overboard. A typical OWS is fitted with one or more dedicated pumps that take suction either from an oily-water holding tank or directly from bilge suction wells and circulate the oily water through the separator, discharge clean water overboard, and transfer separated oil to a waste-oil tank. Electric-motor-driven progressing-cavity pumps are often used with an OWS.

BALLAST PUMPS A ballast pump is used to transfer seawater into and out of a vessel's ballast tanks to adjust list, trim, draft, and stability. Ballast pumps may also be used during a voyage to exchange water contained within ballast tanks to prevent the introduction of nonindigenous aquatic species into coastal and inland waterways. A ballast pump must, therefore, have the capability to take suction either from a sea chest or from ballast tanks that are being emptied. In addition, it must be possible to direct seawater discharged by a ballast pump either to ballast tanks that are being filled or overboard. Furthermore, the seawater discharged by a ballast pump may be used as the motive fluid in eductors that are provided to strip ballast tanks. On some vessels, pumps used for ballasting and deballasting may also be used for other purposes, such as bilge dewatering (that is, a bilge and ballast pump) or fire fighting. Vessels with large segregated ballast tanks, such as many tankers, however, frequently have dedicated ballast pumps.

Horizontally and vertically mounted single-stage centrifugal pumps that are installed in a vessel's machinery space are often used for ballast service. Although many of these pumps are driven by electric motors, larger ballast pumps are sometimes driven by steam turbines. A centrifugal ballast pump is normally located low in a vessel so it will operate with a flooded suction when it is lined up to the sea chest or begins to empty a ballast tank that is full. As the water level in a ballast tank being emptied drops, however, the submergence of the pump's impeller is continuously reduced. To enable it to be reprimed if suction is lost prematurely during operation with a suction lift, a centrifugal ballast pump may be connected to a priming system.

As an alternative to centrifugal ballast pumps installed in a machinery space, hydraulic-motor-driven centrifugal ballast pumps that are submerged directly in the ballast tanks are used on some vessels. In addition, some vessels have vertical line-shaft deep-well ballast pumps. So the pump can take suction from multiple locations, a deep-well ballast pump is frequently installed in a suction can that is connected to the ballast-system suction piping. To enable air and vapor to be removed from the can and suction piping, a deep-well ballast pump is often fitted with self-priming valves.

Hotel Service Pumps

FLASH-DISTILLING-PLANT PUMPS Many steam ships have multistage flash distilling plants that are used to generate freshwater from seawater. Several electric-motor-driven single-stage centrifugal pumps are used with a typical flash unit. Smaller pumps are often provided in a close-coupled configuration.

A distiller-feed pump takes suction from a sea chest and supplies seawater to the distilling plant's first-stage flash chamber. This water often passes through various heat exchangers, such as a distillate cooler, distillate condensers, air-ejector condenser, and seawater heater before it enters the flash chamber. The distiller-feed pump is usually located sufficiently below the vessel's waterline so it operates with a flooded suction.

The hot high-salinity brine remaining in a flash distilling plant's last stage is typically removed by a brine pump that discharges it overboard. A line is often provided to permit

a brine pump's shaft seals to be flushed with seawater discharged from the distiller-feed pump. In addition, a vent line is typically connected from a brine pump's suction to the last-stage flash chamber.

The freshwater or distillate produced by a flash distilling plant is removed from the last-stage distillate condenser by a distillate pump. During normal operation, water discharged from this pump ordinarily passes through a cooler and is then directed to a vessel's distilled-water (reserve-feedwater) or potable-water tanks. A salinity cell in the pump discharge line, however, will typically trip a three-way valve that dumps the distillate to the bilge if salinity of this water is excessive. A vent line may be connected from a distillate pump's suction to the last-stage distillate condenser.

An additional pump is sometimes provided to remove condensate from the hotwell of a flash distilling plant's seawater heater and transfer it to a freshwater drain tank. Alternatively, however, this condensate may be returned directly to a main or auxiliary condenser through a vacuum-drag line. With this latter arrangement, the feed-heater condensate pump is not required.

Because the distillate, brine, and feed-heater condensate pumps each take their suction from a chamber in which the pumped liquid is at or near its vapor pressure, pumps used in these applications should have low NPSH requirements. In addition, to increase the submergence of their impellers, these pumps are often located as far below the distilling plant assembly as practicable.

HEAT-RECOVERY DISTILLING-PLANT PUMPS It is common for a diesel-propelled vessel to be fitted with one or more heat-recovery distilling plants in which jacket water from the vessel's diesel engines serves as the heating medium. In a typical heat-recovery distilling plant, this jacket water is pumped through either a plate or a submerged-tube evaporator.

Seawater is often delivered to a heat-recovery distilling-plant by a dedicated distiller-feed pump (sometimes referred to as an eductor or ejector feed pump) that takes its suction from a sea chest. In some units, this seawater initially passes through the distilling plant's condenser where it absorbs heat from the distilled vapor generated in the evaporator. A portion of this seawater then enters the evaporator section of the shell as feed. The remaining portion of the seawater frequently serves as the motive fluid for an eductor that removes brine from the evaporator and air from the condenser portion of the distilling plant's shell. After leaving the eductors, this seawater is directed overboard with the brine and air. (In some units, the mixture of seawater, brine, and air passes through the distilling plant's condenser before being discharged overboard.) The feed pump is typically located low in a vessel to enable it to operate with a flooded suction.

A distillate pump is used to remove the freshwater collected in a heat-recovery distilling plant's condenser and discharge it either to the vessel's freshwater tanks or, when the salinity is high, to the bilge. Both the eductor-feed pump and the distillate pump are generally electric-motor-driven single-stage centrifugal-type pumps. In addition, they are often furnished in a close-coupled configuration.

POTABLE-WATER PUMPS A potable-water pump typically takes suction from a vessel's potable-water or domestic tanks and discharges freshwater either directly to sinks, showers, and other potable-water fixtures located throughout the vessel or to an air-charged pressure tank, sometimes referred to as a hydrophore or a hydropneumatic tank, that is frequently included in a potable water system. Most vessels have at least two full-capacity potable-water pumps. Electric-motor-driven single-stage centrifugal pumps are often used in this application. In addition, regenerative turbine pumps are sometimes used. Smaller pumps are often furnished in a close-coupled configuration. To improve suction conditions, a potable-water pump may be installed as far as practicable below a vessel's potable water tanks.

In a typical potable-water system in which a hydrophore is connected to the pump discharge line, a potable-water pump may be cycled on and off automatically by a pressure switch installed on the hydrophore. During periods when the pump is not running, any potable-water usage will result in a drop of the water level within the hydrophore, the expansion of the compressed air located in the upper portion of this tank, and a reduction in the hydrophore pressure. However, the force exerted by the compressed air ordinarily

prevents the potable-water system pressure from dropping too suddenly. If the potable-water usage continues, the pressure exerted by the compressed air within the hydrophore will continue to drop until it reaches the cut-in pressure for the potable-water pump's motor. At this point, the pump will be started and the water level and pressure within the hydrophore will normally increase. This will continue until the pressure switch's cut-out pressure is reached and the potable-water pump is stopped.

Although it is less common, on a vessel with a high potable-water demand, a potable-water pump may be operated continuously. When a potable-water pump operates continuously, however, a recirculation line is typically provided to prevent the pump from overheating during periods of low water usage.

HOT-WATER CIRCULATION PUMPS Some of the freshwater discharged from a vessel's portable-water pumps is circulated through a heater and is then directed to sinks, showers, and other hot-water fixtures located throughout a vessel. One or more hot-water circulation pumps are ordinarily used to recirculate unused water contained in the hot-water distribution piping through the heater so the water will remain hot. Electric-motor-driven single-stage centrifugal pumps are typically used in this application. In addition, most hot-water circulating pumps, which generally operate continuously, are furnished in a close-coupled configuration.

SANITARY PUMPS Seawater is sometimes used as the flushing medium in bathrooms on a vessel. When this is the case, a sanitary or flushing-water pump may be provided to take suction from a sea chest and discharge seawater to the vessel's flushometer valves. Two electric-motor-driven single-stage centrifugal pumps are frequently used in this application. When each pump is sized to meet peak-demand requirements, one pump can be cycled on and off automatically by a discharge-pressure switch while the second pump serves as a standby unit. To reduce the number of sanitary-pump starts and stops, an air-charged pressure tank similar to the potable-water-system hydrophore described previously is often included in a sanitary system. A sanitary pump is frequently located low in a vessel so it can operate with a flooded suction.

SEWAGE PUMPS When a vessel has a sewage holding tank, a sewage pump typically takes suction from the holding tank and discharges the sewage and waste water, based on applicable regulations, overboard or to an above-deck connection for transfer ashore. Electric-motor-driven single-stage centrifugal pumps are frequently used in this application. Some of these pumps have casings with large waterways and are fitted with special non clog impellers. Alternatively, some vessels have sump-type centrifugal pumps that are submerged within the sewage tank and are driven by submersible motors or through vertical line shafting by above-tank motors. With either arrangement, two pumps are often provided for each holding tank.

Instead of a sewage holding tank, many vessels now have an onboard sewage treatment plant, referred to as a marine sanitation device (MSD). At least two pumps are ordinarily provided with each MSD to discharge the treated effluent overboard. Standard electric-motor-driven single-stage centrifugal pumps are frequently suitable for this purpose.

Some vessels also have one or more lift stations in which sewage and wastewater is collected before being transferred to an MSD or a larger sewage holding tank. Macerator pumps are frequently used to transfer a lift station's contents to the MSD or holding tank. A macerator grinds the sewage and breaks up solids into small particles that are more easily treated. Lift-station transfer pumps are usually started and stopped automatically by a float switch in the lift station. Two pumps are often provided for each lift station.

AIR-CONDITIONING CHILLED-WATER PUMPS Some vessels utilize freshwater as a secondary refrigerant for air conditioning. With this arrangement, a chilled-water pump circulates the fresh water through a chiller where it is cooled by the system's primary refrigerant. The water then passes through duct-mounted cooling coils, where it absorbs heat from air being supplied to temperature-controlled spaces located throughout the vessel. The chilled water may also be used to cool electronic components. Electric-motor-driven single-stage

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centrifugal pumps are frequently used in this application. A pressurized or elevated expansion tank that is typically installed on the suction side of the system maintains a minimum pressure at the inlet to the pumps.

Cargo Pumps and Associated Pumps

Cargo pumps are used to discharge the liquid cargo transported in a vessel's tanks. The types of vessels that carry liquid bulk cargo include ultra-large and very-large crude carriers (ULCC's and VLCC's), multi-petroleum product and chemical carriers, and liquefied natural gas (LNG) and liquefied petroleum gas (LPG) carriers. In addition, some freighters and supply vessels have tanks in which they can carry a limited amount of liquid cargo.

As liquid cargo is discharged from a vessel, the level in a cargo tank being emptied and, therefore, the suction head to the cargo pump are continuously reduced. This, as well as the fact that many cargoes have relatively high vapor pressures, means cargo pumps must often operate with relatively low values of available net positive suction head (NPSH). In addition, as a cargo tank is being emptied, vortices can form on the surface of the liquid in the tank, and air or inert gas (the space in a cargo tank above a flammable liquid is typically filled with inert gas) from the tank's atmosphere can be drawn through a vortex and into the cargo pump's suction. To reduce the potential for cavitation and vortex formation, the flow rate discharged by a cargo pump is frequently reduced by the use of either manually controlled or motorized remote-operated discharge valves during the latter stages of a pump-out cycle.

The materials used in the construction of a cargo pump's wetted components must be compatible with all of the fluids that the pump will discharge. In addition to the cargoes that the vessel will carry, this can also include seawater if the cargo pumps will be used to remove slops during tank washing. In addition, if fluids that are flammable or explosive will be pumped, components with contacting surfaces should be constructed from nonsparking materials.

A description of some of the different types of pumps used in marine cargo service follows.

CENTRIFUGAL CARGO PUMPS Single-stage centrifugal pumps are used to discharge liquid cargo on many crude-oil carriers. In addition, they are used on some large product carriers that transport a limited number of different liquid cargoes. Typically, three or four centrifugal cargo pumps are installed in one or more pump rooms located low in a vessel. Each pump can frequently remove cargo from multiple tanks through interconnected suction piping.

A centrifugal cargo pump may be furnished with a double-suction impeller mounted in a between-bearings configuration (Figure 11). Pumps of this type, which must be fitted with two shaft seals, are installed in both horizontal and vertical configurations. Alternatively, a vertically mounted centrifugal cargo pump design with an impeller that is overhung on the end of a shaft (Figure 12) is used on some vessels. This latter configuration enables both of the pump's external bearings and the single shaft seal to be located above the impeller, which facilitates maintenance. A centrifugal pump's bearing housings are often fitted with explosion-proof intrinsically safe resistance temperature detectors (RTDs) that permit the operator to detect a hot bearing before it leads to a serious casualty. In addition to the suction and discharge connections, one or more vents are often provided in the suction area of a centrifugal cargo pump's casing so vapor and gas mixed with the liquid entering the pump during a pump-out can be removed from the casing.

Although centrifugal cargo pumps are frequently driven by steam turbines, they are also sometimes driven by diesel engines or electric motors. To isolate it from the explosive vapor that is often present in a pump room, a centrifugal cargo pump's driver is usually installed in a separate machinery space located adjacent to the pump room. With this arrangement, each driver is typically flexibly coupled to the cargo pump that it drives through an intermediate jackshaft. The opening in the pump-room bulkhead (for a horizontal pump, Figure 13a) or overhead (for a vertical pump, Figure 13b) through which the jackshaft passes is ordinarily sealed with a gas-tight stuffing box to prevent explosive

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1 Casing
2 Shaft
3 Casing wearing ring
4 Impeller
5 Vent / stripping connection
6 Thrust bearing
7 Bearing housings
8 Line bearing
9 Auxiliary stuffing box
10 Shaft nut
11 Mechanical seal

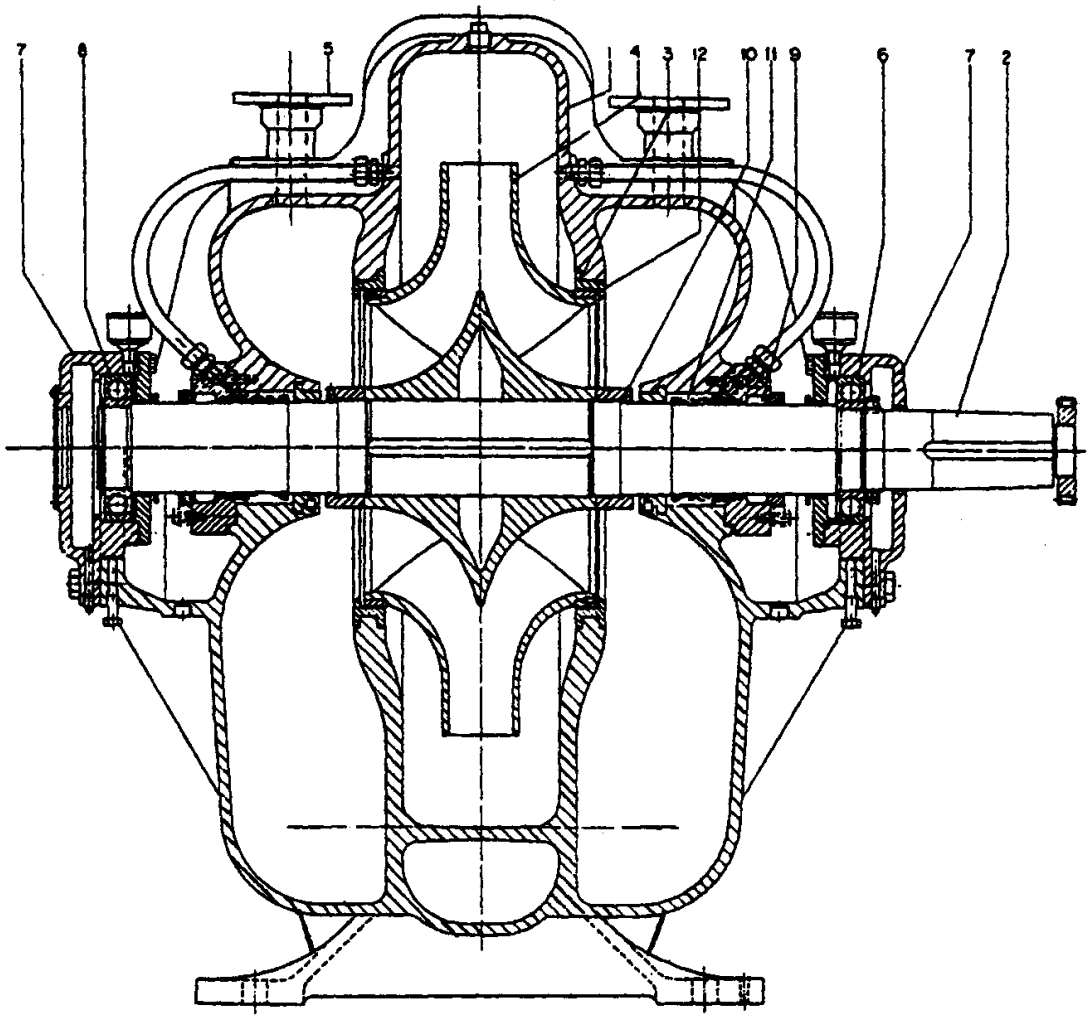


FIGURE 11 Between bearings centrifugal cargo pump (Flowserve Corporation)

vapor in the pump room from entering the machinery space. A thrust bearing is also frequently provided in the driver to support the weight of a jackshaft used with a vertically mounted cargo pump.

As the liquid level in a cargo tank being emptied approaches the inlet to the suction tailpipe (referred to as a suction strum), air or inert gas from the tank's atmosphere can be drawn into the cargo pump through vortices that form on the surface of the liquid. This entrained gas can cause a centrifugal cargo pump to lose suction before the cargo tank has been emptied. Although a separate stripping pump may then be used to remove the liquid remaining in the cargo tank, stripping pumps typically deliver much lower capacities than large centrifugal cargo pumps. Therefore, self-priming/stripping systems are sometimes employed to increase the amount of cargo that can be discharged by a centrifugal cargo pump. A recirculation-priming system and an automated-vacuum-stripping system are two types of systems that are used.

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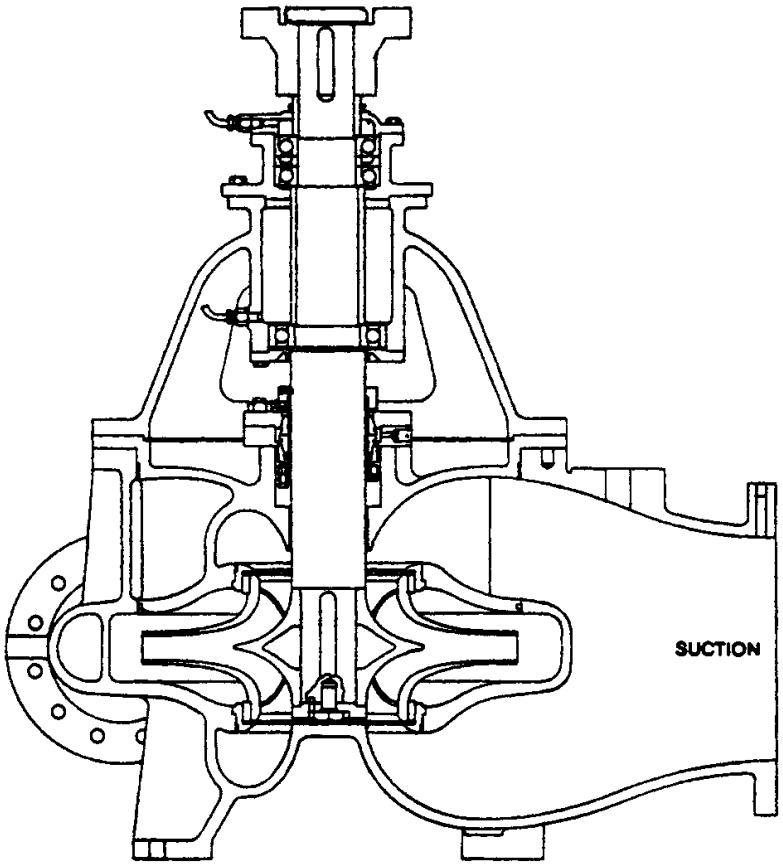


FIGURE 12 Vertical overhung centrifugal cargo pump (Flowserve Corporation)

In a typical recirculation-priming system, a recirculation tank is mounted between the cargo-system suction piping and the inlet to the cargo pump. In addition, one or more priming valves are mounted in the cargo discharge line near the outlet from the pump. Also, a recirculation line is connected from these valves back to the recirculation tank at the pump's inlet, a check valve is installed in the cargo-pump discharge piping above and

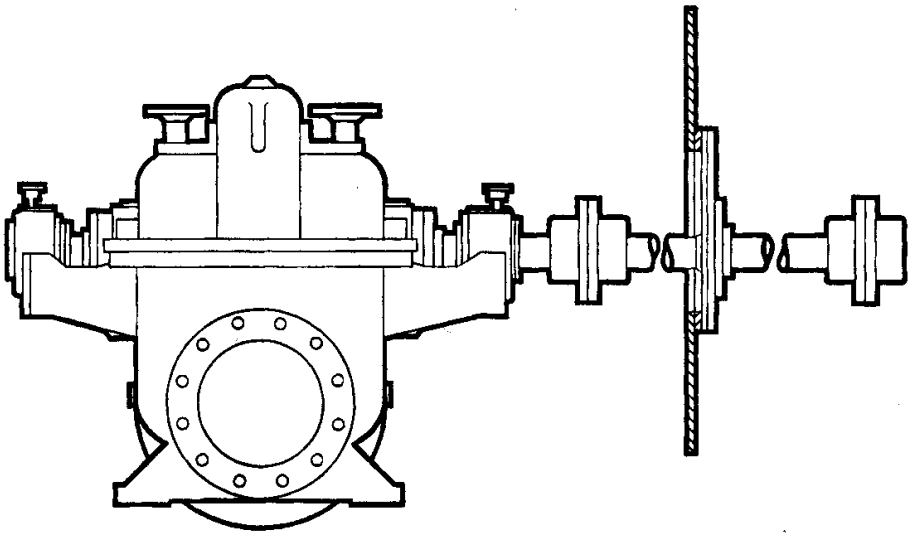


FIGURE 13A Horizontal centrifugal cargo pump with jackshaft (Flowserve Corporation)

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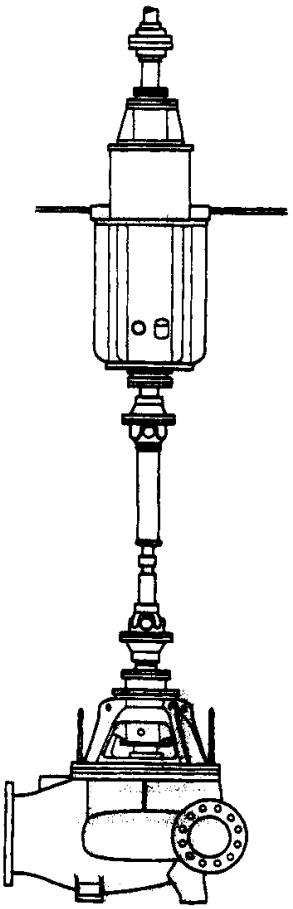


FIGURE 13B Vertical centrifugal cargo pump with jackshaft (Flowserve Corporation)

downstream from the priming valves, and a vent line with a second check valve is connected from the top of the recirculation tank to the cargo-pump discharge line just upstream of the discharge check valve. A branch from the recirculation tank's vent line is ordinarily piped to the vent/stripping connections in the cargo pump's casing.

At the beginning of a pump-out, the recirculation tank is filled with liquid cargo and the priming valves are closed. However, if, as the level in the cargo tank being emptied is reduced, the percentage of gas in the liquid being transferred increases sufficiently for the cargo pump to lose suction, the priming valves in the discharge line will open. This permits the cargo contained in the discharge line between these priming valves and the discharge check valve, which closes when the flow through it stops, to be returned through the recirculation line to the recirculation tank. As this liquid drains through the open priming valves, a vacuum is created in the discharge line. The gas contained in the recirculation tank is displaced by the returning liquid and is drawn by the discharge-line vacuum through the vent line and into the portion of the discharge piping that has been evacuated.

As the recirculated liquid cargo enters the recirculation tank, the liquid level within the tank rises until the submergence of the cargo pump's impeller is sufficient for the pump to regain suction. After this occurs, the flow of liquid through the pump will resume, the priming valves and the check valve in the vent line will close, and the gases that have accumulated in the discharge line will be forced through the discharge check valve. The priming cycles will continue until enough gas has been removed from the cargo suction line for the cargo pump to operate normally. As the level in the cargo tank being emptied continues to be reduced, the amount of gas entering the suction strum will increase, and the cargo pump will lose suction more frequently. When the amount of liquid remaining in the cargo tank is not sufficient for the cargo pump to be reprimed, the pump should be stopped.

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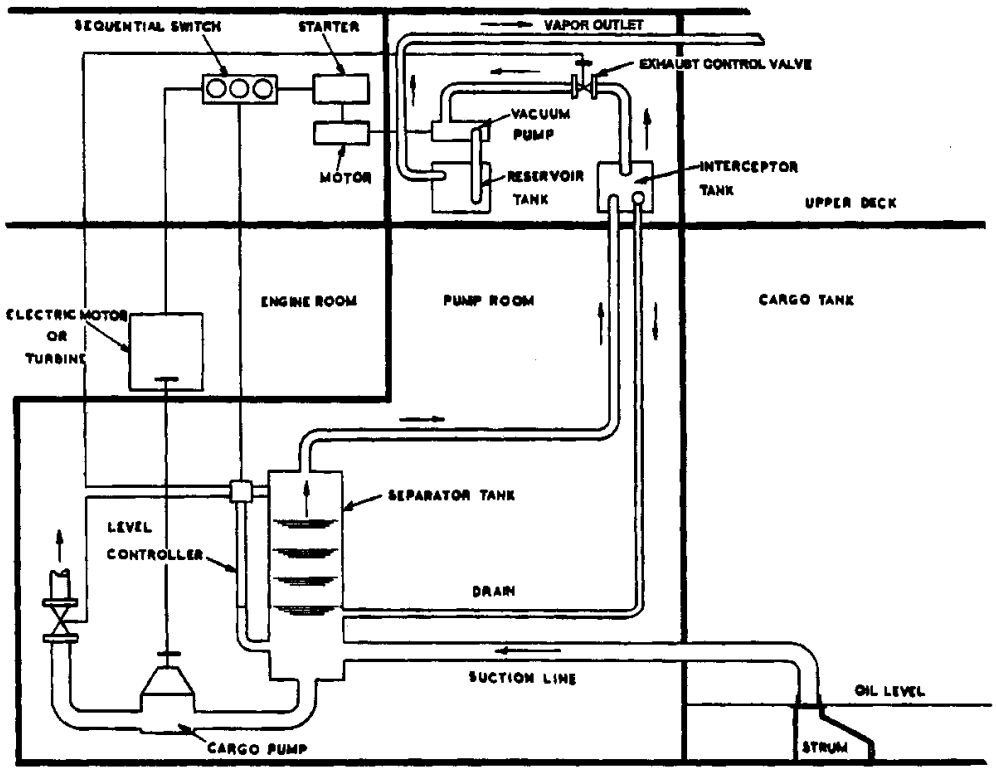


FIGURE 14 Automated-vacuum-stripping system (Flowserve Corporation)

The key components used in a typical automated-vacuum-stripping system (Figure 14) include a separator tank located at the cargo pump's inlet, a vent line connected to the top of the separator tank, an automatic exhaust control valve mounted in the vent line, one or more electric-motor-driven vacuum pumps, a cargo-pump discharge valve with actuator, and associated controls. As with the recirculation system, a branch from the separator tank's vent line is ordinarily piped to the vent connections in the cargo-pump casing.

At the beginning of a pump-out, the separator tank is ordinarily filled with liquid cargo. However, as the pump-out progresses, the liquid level in the cargo tank being emptied and the cargo-pump's suction pressure will both be reduced. Eventually, cargo vapor will typically begin to accumulate in the top of the separator tank and the liquid level in the separator tank will drop. At a specified point, the exhaust control valve in the vent line opens to permit vapor in the separator tank to pass through the vent line before it can enter the cargo pump. If the venting process is not sufficient, the vacuum pump will start and draw vapor out of the separator tank. After the vapor has been removed from the separator tank, the separator-tank liquid level may rise sufficiently for the vacuum pump to stop and the exhaust control valve to close. The exhaust control valve will, however, reopen and the vacuum pump will restart automatically if, because of the reaccumulation of vapor in the top of the separator tank, the separator tank level again drops. The exhaust control valve will continue to open and close and the vacuum pump will continue to cycle on and off automatically, as needed, based on the level in the separator tank.

With further reductions in the cargo-tank liquid level, the amount of cargo that vaporizes in the cargo-pump suction line and separator tank will increase. In addition, as the cargo-tank liquid level approaches the suction strum, vortices will typically form on the surface of the liquid in the cargo tank and draw gas from the tank's atmosphere into the suction line. As the percentage of vapor and entrained gas in the liquid cargo being unloaded increases, the frequency of the exhaust-control-valve/vacuum-pump cycles described above will increase. Eventually, near the end of the pump-out, it will usually become necessary for the exhaust control valve to remain open and for the vacuum pump to operate continuously.

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In addition to operating the exhaust control valve and the vacuum pump, the automated-vacuum-stripping system will normally automatically throttle the cargo pump's discharge valve whenever the vacuum pump is started. In addition, when permitted by the type of driver used, the cargo pump's operating speed may be reduced. Controls may also be included in the system to automatically shut down the cargo pump driver when the pump-out has been completed.

DEEP-WELL CARGO PUMPS Vertical line-shaft deep-well pumps are used to discharge liquid cargo on some multi product and chemical carriers. Therefore, a deep-well cargo pump must frequently be suitable to transfer a wide range of liquids having different specific gravities, vapor pressures, viscosities, and temperatures. Some cargoes, such as lubricating oils, waxes, and other viscous cargoes, may be heated to improve pumpability. With certain cargoes, such as molten sulfur, steam, or a heated liquid may even be circulated through jackets that surround the deep-well pump to prevent the cargo from solidifying within the pump. In addition, deep-well pumps are sometimes used to discharge cryogenic cargoes, such as liquefied petroleum gas (LPG).

A deep-well cargo pump can be driven by a vertical electric or hydraulic motor mounted on top of the pump's discharge head. Alternatively, a deep-well cargo pump may be driven through a right-angle gear mounted on the discharge head by a horizontal motor, steam turbine, or diesel engine. Although a deep-well cargo pump's discharge head is often mounted on a vessel's main deck, on some vessels, the deep-well-cargo-pump discharge heads are located below deck in a pump room. In addition, in some cases, a deep-well cargo pump's horizontal driver is located in an adjacent space and is coupled to the pump's right-angle gear with a jackshaft that passes through a bulkhead separating the pump's discharge head from the driver. The opening for the jackshaft in the bulkhead is ordinarily sealed with a gas-tight stuffing box so the driver can be isolated from any explosive vapor that may be emitted from the pup. Some designs are also available with submersible motors that allow the entire pump-driver assembly to be located at the bottom of the cargo tank.

Many deep-well cargo pumps are furnished with a multistage bowl assembly that has single-suction impellers. When high-vapor-pressure or high-viscosity cargo will be pumped, an inducer is sometimes mounted on the lower end of the bowl assembly's impeller shaft to reduce the pump's net positive suction head requirements. Alternatively, a deepwell pump may be fitted with a special low-NPSH first-stage impeller, or, in some cases, a double-suction first-stage impeller. A spool piece is sometimes installed between the top of a deep-well pump's bowl assembly and the lower end of its column assembly so the bowls can be removed for maintenance while the column and discharge head are still in place.

Hydraulic axial unbalance resulting from the use of single-suction impellers can result in the generation of a high axial thrust. Some deep-well pumps are fitted with a balancing device, such as a balancing drum or front and back impeller wear rings, to reduce this thrust. Alternatively, a thrust bearing is frequently used to absorb the axial thrust acting on a deep-well pump's shaft. This bearing is sometimes installed in a housing that is an integral part of the pump's discharge head. In many cases, however, the thrust bearing is in the vertical driver or right-angle gear (when a horizontal driver is used) that is mounted on top of the pump's discharge head. With this latter arrangement, the deep-well pump's top shaft is typically secured to the vertical driver or gear either with a rigid coupling when the driver or gear has a solid shaft or with an adjusting nut when a hollow-shaft driver or gear is used. It is always important that the pump shaft be raised the proper amount during assembly to prevent contact between the pump's rotating and stationary parts during operation.

Mating sections of a deep-well pump's line shaft are typically connected with threaded or keyed couplings. Radial bearings that support the line shaft are frequently mounted in brackets, sometimes referred to as spiders, that are sandwiched between mating sections of column pipe. Impeller- and line-shaft bearings are often lubricated by the pumped cargo. Consequently, bearing materials must typically be compatible with all of the fluids that will be pumped. The bearings should also be able to tolerate operation with loss of suction if the deep-well pump will be used for stripping or during tank cleaning. Although bronze bearings are common, bearings constructed from carbon, polytetrafluoroethylene (PTFE) compounds, and various composites and plastics have also been used.

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The use of a single shaft seal at the location where a deep-well pump's top shaft penetrates the discharge head is sometimes suitable. A multiple sealing arrangement consisting of two packed stuffing boxes or a double mechanical seal may be utilized, however, when a deep-well pump will discharge a volatile cargo. In addition, if a deep-well pump is fitted with a double mechanical seal, a static seal may also be provided to prevent vapor from escaping around the pump's shaft during periods when the mechanical seals are being replaced. A reservoir tank that stores liquid for seal lubrication is often mounted on the side of the discharge head of a deep-well pump that has one or more mechanical seals.

On some multi-product and chemical carriers, a separate deep-well pump is installed in each cargo tank. This arrangement eliminates the need for suction piping, and it can result in a high degree of cargo segregation. On a double-bottom vessel, the deep-well pump's suction opening is frequently submerged in a suction well located in the bottom of the cargo tank. In-tank supports are often provided to stabilize the pump during periods of pitch and roll. However, to prevent these supports from restricting thermal expansion and distorting a pump's column or line shaft as a vessel flexes, they are generally not rigidly attached to the pump.

To prevent fluid in a deep-well pump's discharge head, column pipe, and bowl assembly from draining back into a cargo tank after the pump is stopped, a deep-well cargo pump's inlet opening is sometimes fitted with a nonreturn suction valve (Figure 15). During normal operation, the valve is open. However, after the discharge cycle has been completed, the suction valve closes automatically. The deep-well pump's driver is then stopped, the pump's above-deck discharge valve is closed, and compressed air or inert gas is injected into the pump through a connection in the discharge head. The gas forces the cargo contained within the deep-well pump out through a bypass line connected to the lower portion of the bowl assembly and into the vessel's piping.

On some vessels, such as those that carry either a limited number of different cargoes or cargoes that are not sensitive to contamination, each deep-well pump may be used to discharge cargo from several of the vessel's tanks. When this arrangement is used, each deep-well pump is generally mounted in a suction tank or can that is connected to multiple cargo tanks via suction piping.

A deep-well cargo pump that is installed in a suction can is frequently fitted with automatic priming valves that enable the pump to remove gas and vapor from the can and from the attached suction piping. The operation of these priming valves and their associated components is often similar to the operation of the recirculation system used to reprime centrifugal cargo pumps. A typical self-priming deep-well pump is fitted with two or three priming valves that are mounted in the pump's column pipe just above the bowl assembly (Figure 16). When the pump is discharging fluid, these valves are closed. However, when the deep-well pump loses suction, the priming valves open and permit the liquid contained within the pump's column and discharge head to drain into the suction can. As this liquid drains through the open priming valves, a vacuum is created in the discharge head and column pipe. The gas in the bottom of the can, which is displaced by the returning liquid, is drawn by the vacuum through a vent line that is connected from the top of the suction can to the pump's discharge head and into the evacuated portion of the discharge head and column. When the liquid level in the suction tank rises sufficiently for the deep-well pump to regain suction, the flow of liquid through the pump resumes, the priming valves and a check valve in the vent line close, and the gases that have been drawn into the column and discharge head are forced out through a check valve installed at the pump's discharge connection. The priming cycles will continue until enough gas has been removed from the cargo suction line for the deep-well pump to operate normally. As the level in the cargo tank being emptied continues to be reduced, the amount of gas entering the suction can will increase, and the deep-well pump will lose suction more frequently. When the amount of liquid remaining in the cargo tank is not sufficient for the deep-well pump to be reprimed, the pump should be stopped.

HYDRAULIC-MOTOR-DRIVEN SUBMERSIBLE CARGO PUMPS Hydraulic-motor-driven submersible pumps are used to discharge cargo on many chemical and multi-petroleum-product carriers. In addition, they are used for cargo discharge on some crude carriers. A

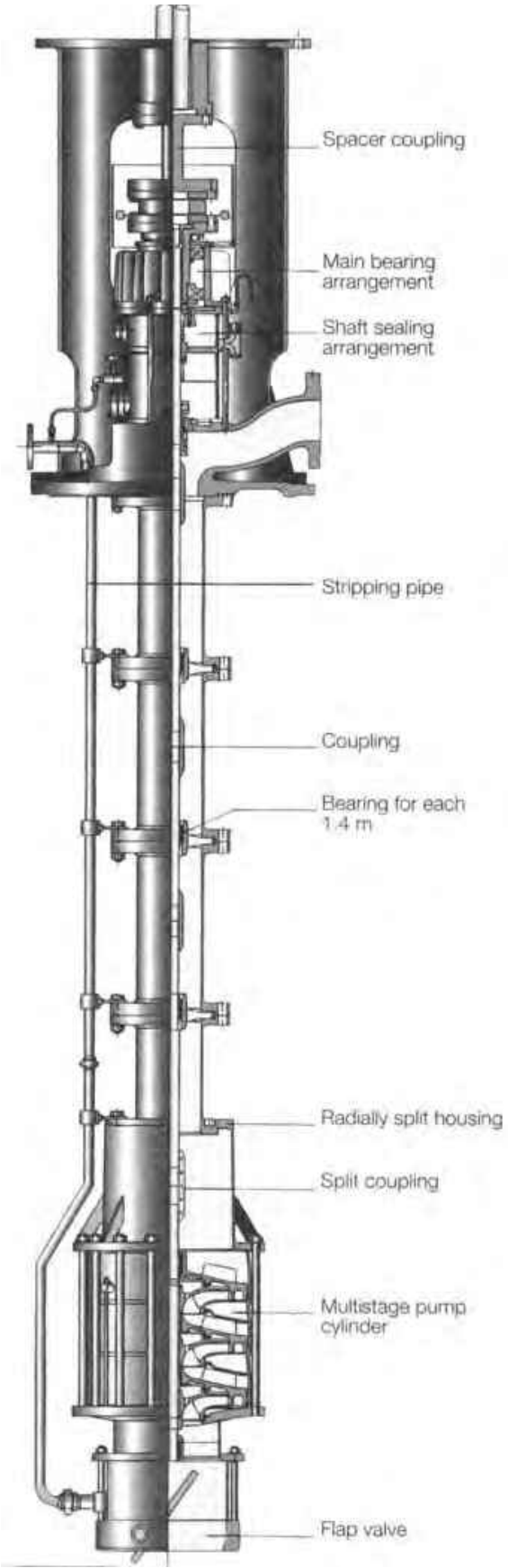


FIGURE 15 Vertical deep-well cargo pump (Svanejoh International A/S)

600	Discharge Head
604	Adjusting nut
608	Head or top shaft
616	Stuffing box
617	Stuffing-box bushing
618	Gland
620	Packing
624	Bypass pipe
641	Top column pipe
644	Bottom column pipe
646	Line shaft
649	Line-shaft coupling
652	Bearing retainer or bracket
653	Line-shaft bearing
655	Impeller-shaft coupling
660	Pump or impeller shaft
666	Top bowl
670	Intermediate bowl
672	Bowl bearing
673	Impeller
674	First-stage (priming-stage) impeller
677	Impeller taper lock
685	Adapter case
689	Suction bell
690	Suction-bell bearing
702	Autoprime (self-priming) valve housing
704	Autoprime (self-priming) valve spring
705	Autoprime (self-priming) valve
706	Air-release (vent) check valve
707	Air-release (vent) line
708	First-stage (priming-stage) impeller
762	Column-flange bolt
764	Bowl cap screw
779	Stuffing-box gasket

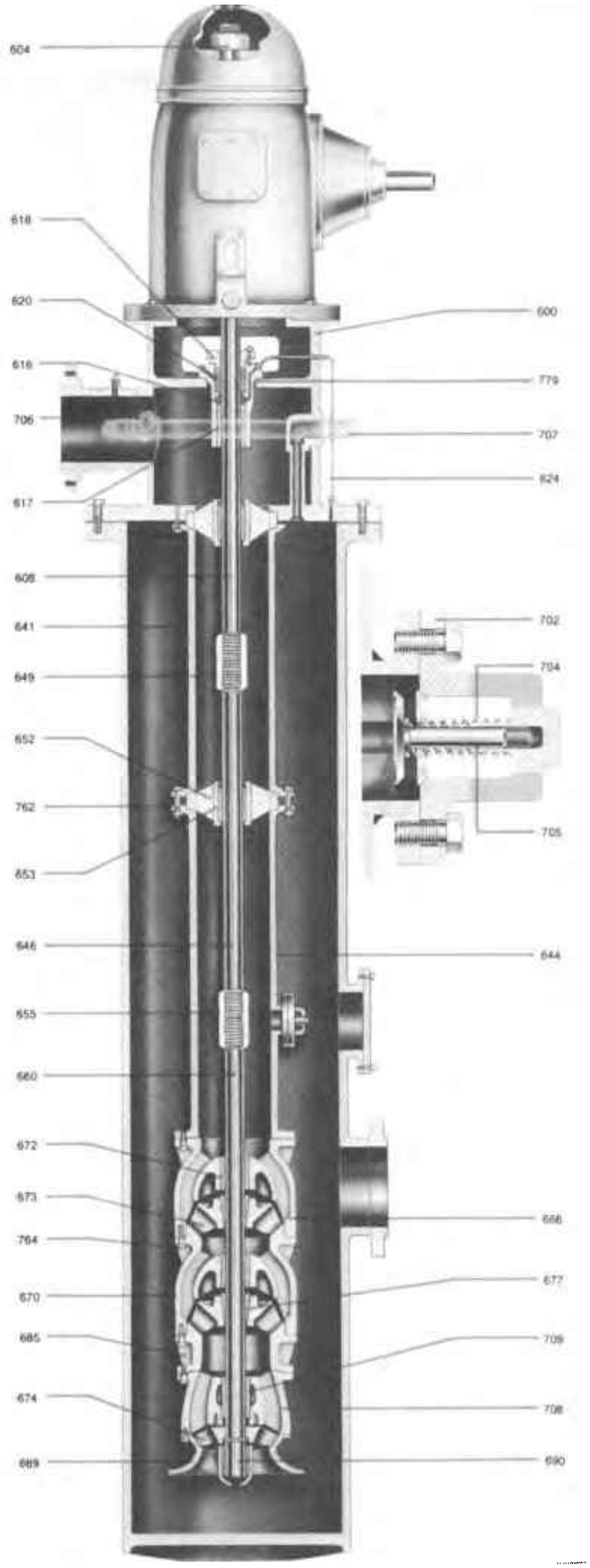


FIGURE 16 Self-priming vertical deep-well cargo pump (ITT/Gould Pumps, Inc.)

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separate pump is usually installed in each cargo tank, which eliminates the need for suction piping. In addition, to prevent different cargoes carried by a vessel from mixing, each pump may be connected to an independent above-deck discharge line. Although the pressurized oil required to drive the hydraulic motors for all of the cargo pumps on a vessel is frequently supplied by a central hydraulic system, a self-contained hydraulic power pack is furnished for each cargo pump on some vessels.

Each submersible unit usually consists of a single-stage end-suction centrifugal pump with a short shaft that is driven, often through either a splined connection or a coupling, by a hydraulic motor (Figure 17). The pump's shaft is typically supported by antifriction ball or roller bearings that are submerged in and lubricated by the hydraulic oil that drains from the motor. The pump is mounted on the lower end of a vertical support pipe that is suspended from a top plate installed on the vessel's main deck. The hydraulic-oil supply and return lines are ordinarily enclosed within this support pipe. Fluid discharged from the pump's volute-type casing typically passes through a second vertical pipe that terminates at an above-deck discharge connection. A control valve that can be used to vary the flow of hydraulic oil to the motor and, therefore, the pump's operating speed valve is usually mounted on the above-deck top plate.

Mechanical or lip-type seals are generally used at the shaft penetrations in a submersible pump-and-hydraulic-motor assembly to prevent hydraulic oil from leaking into a cargo tank and to prevent cargo from mixing with hydraulic oil. Air or inert gas can usually be circulated through a void space or cofferdam that runs through the vertical support pipe that surrounds the hydraulic supply and return lines, the housing that surrounds the hydraulic motor, and a chamber between the pump's seals. After leaving the cofferdam, this gas frequently passes through an above-deck trap in which liquid is separated so a hydraulic-oil or cargo-seal leak can be detected. Compressed air or inert gas is also frequently injected into a submersible pump's above-deck discharge connection after a pumpout has been completed to force cargo contained within the vertical discharge pipe out through a small bypass line connected to the base of the pump. However, because a submersible pump's inlet is generally not fitted with a nonreturn valve, this operation must be performed before the driver is stopped.

It is often possible to disassemble a submersible pump and remove it from a cargo tank for maintenance without disturbing the vertical support pipe or top plate. In most cases, vessels fitted with hydraulic-motor-driven submersible cargo pumps carry one or more portable hydraulically driven submersible pumps that can be lowered into a cargo tank with a winch and used to discharge cargo if the main pump in the tank is inoperable.

In addition to being used for cargo discharge, a submersible cargo pump is sometimes operated while a vessel is underway to circulate cargo through a diffuser so sediment in the liquid does not settle in the cargo tank or through an above-deck heater to prevent the cargo from cooling. Also, to eliminate the need for separate drop lines, on some vessels, liquid is loaded into small cargo tanks by allowing it to flow backwards through a vessel's submersible cargo pumps. When this is done, a nonreverse brake is generally mounted on the submersible pump's shaft to prevent reverse rotation during loading. However, because of the resistance created by the pump's impeller, loading through a submersible cargo pump can increase loading times.

ELECTRIC-MOTOR-DRIVEN SUBMERSIBLE CARGO PUMPS Electric-motor-driven submersible pumps are often used to discharge cargo from liquefied natural gas (LNG) and liquefied petroleum gas (LPG) carriers. Each submersible unit typically consists of a vertical centrifugal pump with either one or two impellers that are mounted on the lower end of an electric motor's shaft. In addition, an inducer is often installed at the inlet to the first-stage impeller (the only impeller in a single-stage pump) to reduce the pump's net positive suction head requirements.

Pumps used for cargo unloading are frequently installed directly within a vessel's cargo tanks. Cargo typically enters an electric-motor-driven submersible pump through an opening in the bottom of the pump. After being discharged by the pump's impellers, the cargo often enters an annular passage formed between the motor frame and an outer casing that surrounds the motor. A portion of this liquid usually passes through openings in the motor frame and flows through the motor. In addition to cooling the motor, this bypass

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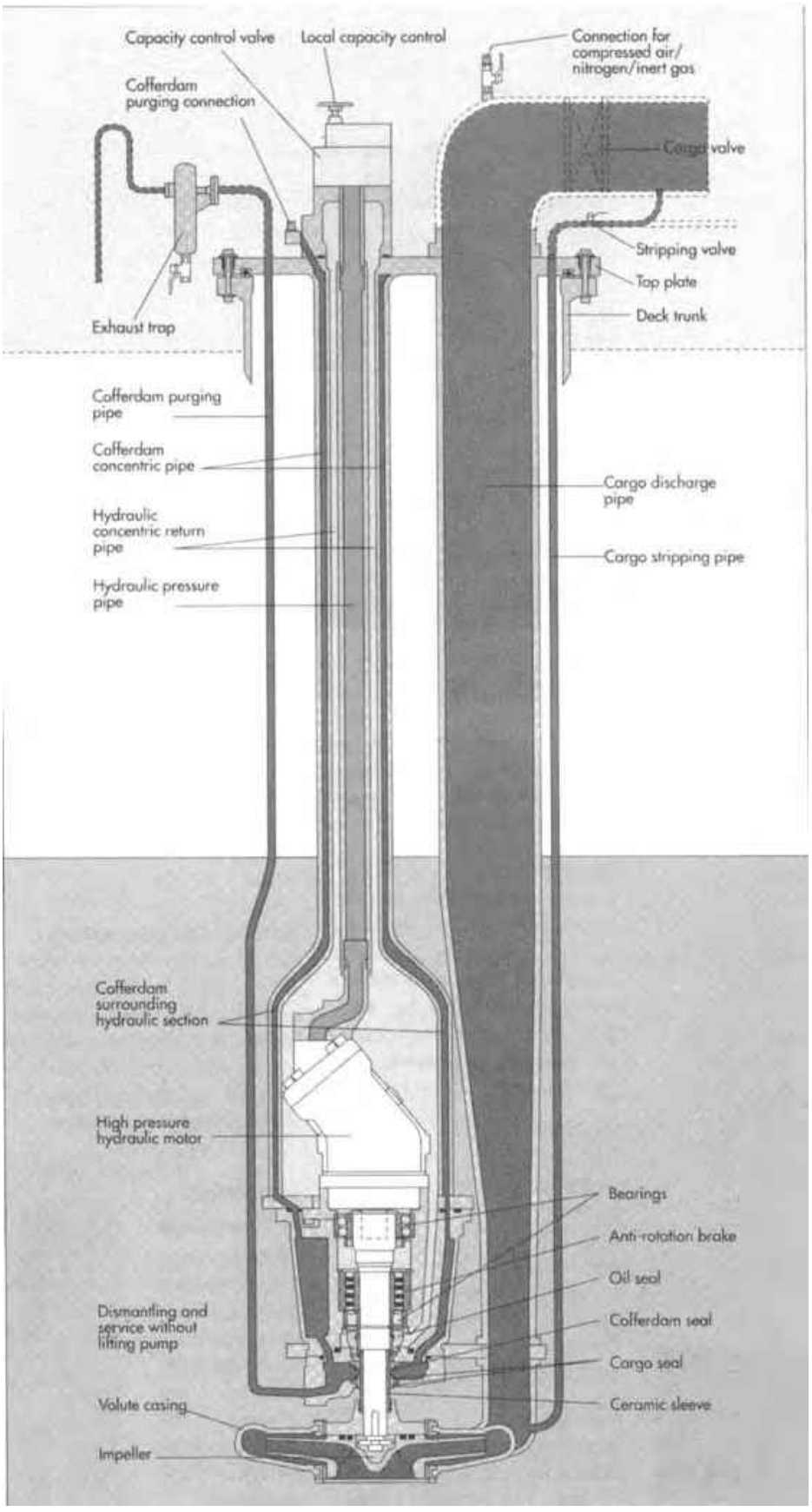


FIGURE 17 Hydraulic-motor-driven submersible cargo pump (Frank Mohn AS)

flow lubricates the ball bearings that support the common pump and motor shaft. After leaving the top of the annular passage, the cargo that has been discharged by the pump passes through a connection located above the motor and enters a vertical pipe that leads to the main deck.

In addition to the main cargo unloading pumps, smaller spray or cool-down pumps are often installed on an LNG carrier. When an LNG carrier is unloaded, some cargo is typically left in each tank. During the voyage back to the loading terminal, the spray pumps circulate this LNG through a cool-down header and spray nozzles in each tank. The LNG vaporizes as it passes through the spray nozzles and absorbs heat from the cargo tanks. This enables the tanks to be kept cold until the vessel is reloaded with additional cargo.

ROTARY CARGO PUMPS Some vessels that carry high-viscosity cargoes have multiple-screw- or lobe-type rotary main cargo unloading pumps. In addition, vessels that have centrifugal-type main cargo pumps sometimes also have lower-capacity screw, lobe, or sliding-vane pumps that are used to strip cargo tanks.

A rotary main cargo or stripping pump may be installed in a pump room located in the lower part of a vessel. With this arrangement, the pump can typically take its suction from multiple cargo tanks through interconnected suction piping. So the driver can be isolated from explosive vapor in the pump room, it is frequently installed in an adjacent space and is coupled to the pump with a jackshaft that passes through a bulkhead stuffing box. A typical rotary cargo pump is driven either by a variable-speed driver or by a constant-speed driver through a fluid coupling, which enables the pump speed and, therefore, the capacity delivered to be changed during a pump-out.

Some multiple-screw rotary cargo pumps are furnished in a deep-well configuration (Figure 18), which can eliminate much of the suction piping in a cargo system. Although a vertical driver can be used to drive a deep-well rotary pump, many deep-well rotary pumps are driven through right-angle gears by horizontal motors or engines to facilitate maintenance and reduce vertical height requirements. Typically, the output shaft of the above-deck vertical driver or right-angle gear is connected to the power rotor in the pump through a line shaft that is enclosed within a vertical column pipe. Cargo discharged by a deep-well rotary pump may pass through the column, or it may pass through a separate vertical pipe mounted adjacent to the column. Bearings that support the line shaft, together with the pump's bearings and timing gears, when used, are sometimes lubricated by the pumped fluid. Alternatively, a deep-well rotary pump may be furnished with a pressurized forced-feed lubrication system.

RECIPROCATING CARGO PUMPS Direct-acting reciprocating pumps are used to strip cargo tanks on some vessels. These units typically have double-acting pistons in both the liquid- and the drive-end cylinders. Some reciprocating stripping pumps are mounted in a pump room and are connected to the vessel's cargo tanks through suction piping. With this arrangement, one pump can be used to strip multiple tanks. Pumps installed in this fashion are frequently duplex units (that is, two liquid cylinders) that are mounted vertically and are driven by steam (Figure 19).

When used on multi-product carriers, a separate reciprocating stripping pump may be used for each cargo tank. These stripping pumps are frequently driven by compressed air or inert gas. On some vessels, a horizontal duplex reciprocating pump is mounted in the bottom of each cargo tank. Alternatively, a vertical simplex (i.e., one liquid cylinder) pump with a liquid cylinder that is submerged within a cargo tank and a drive cylinder that is mounted on deck is sometimes used. With this latter configuration, the liquid-end piston is coupled to the piston in the drive cylinder through a long intermediate shaft.

INERT-GAS SYSTEM PUMPS To reduce the risk of explosion and fire, the space in a cargo tank above a flammable liquid must typically be kept filled with an inert gas. Some vessels use flue gas from either a fossil-fueled steam boiler or a dedicated oil-fired inert-gas generator to inert cargo tanks. A pump is usually required to deliver seawater to a scrubber where the water is used to cool, clean, and desulfurize the inert gas. A separate pump is also frequently required to supply seawater to a wet-type deck seal that is used to prevent

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1 Pumping screws
2 Timing gears
3 Coupling
4 Line shaft or jackshaft
5 Column pipe (standpipe)
6 Discharge head with carrying plate
7 Bearing bracket
8 Column pipe (standpipe)
9 Bearing housing
10 Long pump shaft
11 Short pump shaft
12 Pump body or housing
13 Suction connection / bearing housing

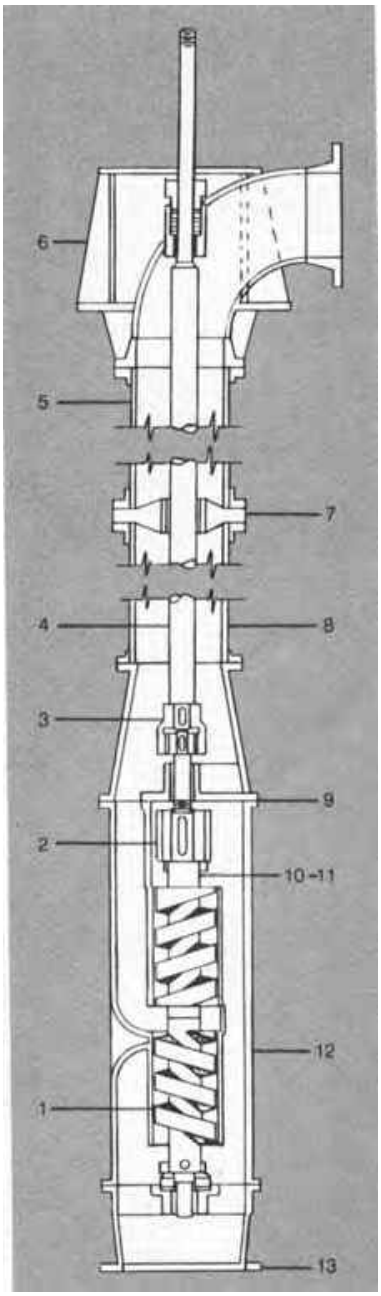


FIGURE 18 Deep-well screw pump (IMO-Warren Pump)

vapor in the vessel's cargo tanks from flowing backwards through the inert-gas supply piping and into the machinery spaces. It is common to use single-stage electric-motor-driven centrifugal pumps for both of these applications.

TANK-CLEANING PUMPS On some vessels, hot seawater is used to clean cargo tanks of residue that remains in the tanks after the liquid cargo has been discharged. A typical tank-cleaning or tank-washing pump takes suction from a sea chest and discharges seawater through a heater in which it is often heated to a temperature of approximately 200°F (93°C). The hot seawater then passes through nozzles in tank washing machines and is sprayed onto the sides of the cargo tanks being cleaned. Single- and two-stage centrifugal pumps that are driven by steam turbines, electric motors, or hydraulic motors are frequently used in this application. A tank-washing pump may also be used for other purposes, such as fire fighting.

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1 Steam cylinder
2 Steam piston rod
3 Liquid cylinder
4 Liquid-cylinder liner
5 Liquid piston rod
6 Liquid piston
7 Liquid piston packing
8 Liquid valve service
9 Air chamber (pulsation dampener)

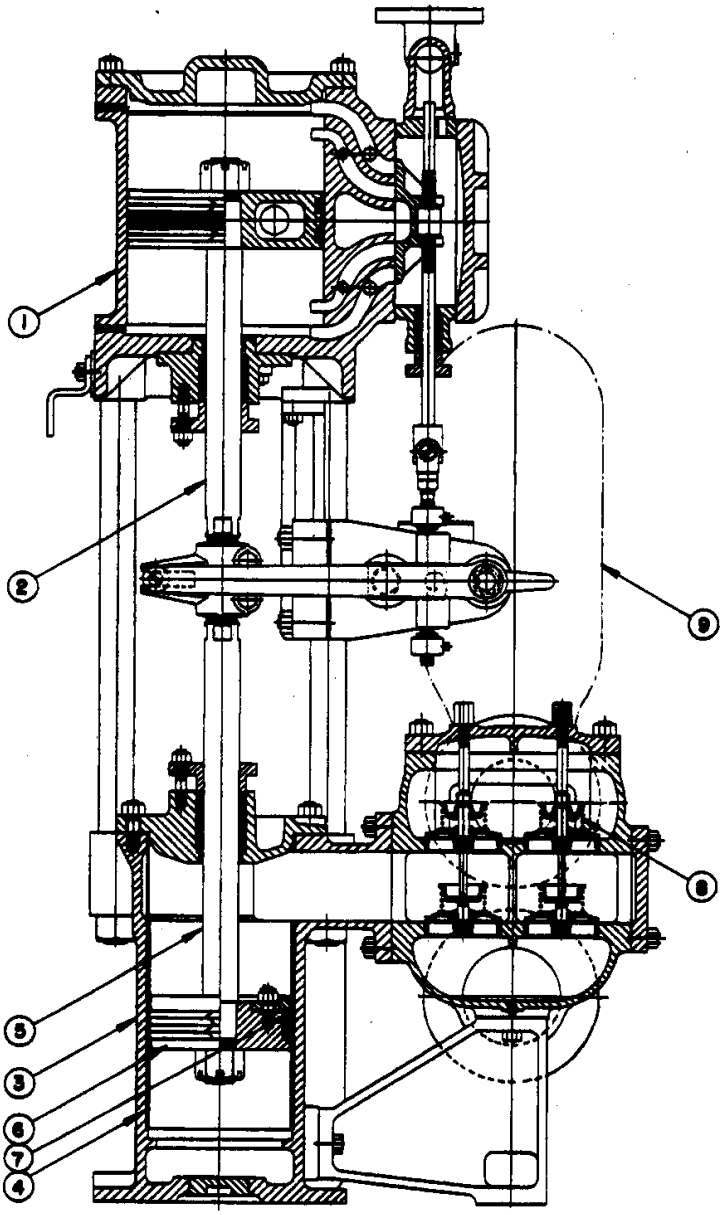


FIGURE 19 Duplex direct-acting reciprocating pump (Flowserve Corporation)

As an alternative to washing cargo tanks with seawater, crude carriers frequently clean tanks with crude oil, referred to as crude-oil washing (COW). A general-service pump is often used on a crude carrier to deliver pressurized crude oil to the vessel's tank-washing machines. The pump may receive this oil from a holding or slop tank on the vessel. The general service pump may also be used to deliver cargo to an eductor that strips cargo tanks during the final stages of unloading or during crude-oil washing. This cargo, which serves as the motive fluid in the eductor, mixes with the fluid being removed from the cargo tank and is then usually discharged by the eductor into one of the vessel's slop tanks. On a typical crude carrier, the general-service pump is similar in configuration to the vessel's main cargo pumps but is smaller.

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SECTION 12.13
HEATING AND AIR CONDITIONING

BURT RISHEL

HEATING AND AIR CONDITIONING

Heating, as defined herein, is the warming while air conditioning is the cooling of people, space, and equipment. These subjects will be discussed with reference to their use of pumps.

Two great developments have had a profound effect upon these fields. They are digital control and variable speed drives. Microprocessor-based controls have replaced pneumatic control systems for improved reliability, response time and system feedback, while new software tools provide the flexibility to address any conceivable control problem. Digital control technology is at work in onboard boiler and chiller control as well as variable speed drives that operate pumps and fans. Standard building automation system protocols and internet capability have further enabled communications between manufacturers.

Digital controls also contribute to energy improvements by providing the necessary platform for adaptive control techniques. Adaptive control algorithms manage operation of individual components like boilers, chillers, pumps, and cooling towers so that the total energy usage is minimized. Adaptive algorithms eliminate the operation of the system based on fixed, preset parameters. Rather, they automatically change set points for addition/subtraction of equipment as conditions change; this minimizes energy consumption.

Figure 1 provides an example of how adaptive algorithms control two variable speed pumps to achieve minimum energy use. Under original system conditions, the second pump would be started at 12 kW and 1140 gpm. As the system conditions change, the algorithm calculates that the second pump should be added at 10 kW and 1080 gpm with adaptive control. Concluding, adaptive control has eliminated the forced operation of an HVAC system to preconceived control parameters.

Variable speed drives have been developed for almost any size pump motor. Integral motors for pumps from 5 to 750 hp are usually for electrical service of 460 V 3 phase 60 Hz while large motors are designed for medium voltages of 2300 or 4160 V. The popularity of these drives has reduced their cost so that they can be applied to any heating or cooling system that has a variable flow requirement. This has eliminated the need for manual adjusting devices such as balance valves and multiple duty valves and special circuiting of piping such as crossover bridges. These hydronic systems require much less energy today with variable speed drives. Conversion of constant speed pumping systems to variable speed has reduced the pumping energy to less than half that required with constant speed pumps. Also, the number of pumps required has been reduced substantially. For

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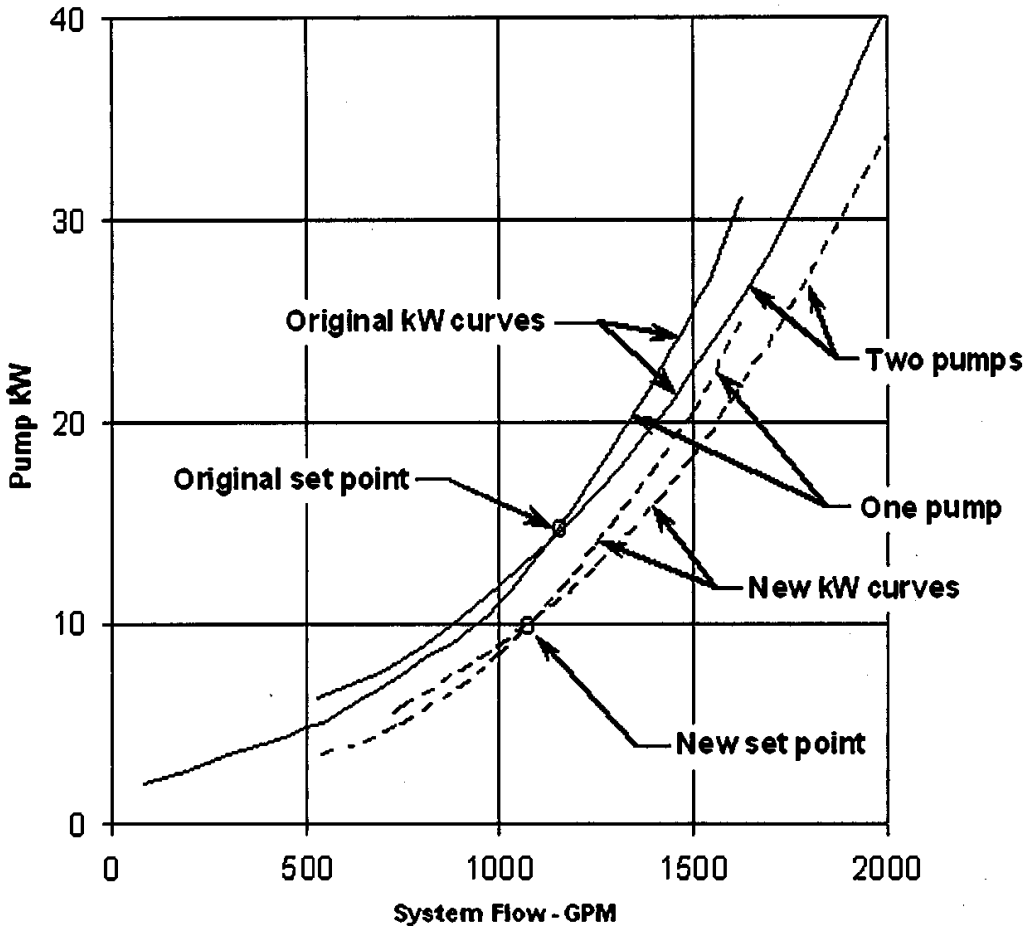


FIGURE 1 Adaptive control of two centrifugal pumps (tekWorx, LLC, Cincinnati, OH)

example, the preferred pumping arrangement was primary/secondary, Figure 2, but it is now replaced by variable primary pumping, Figure 3, on most systems.

Heating The majority of residential and commercial heating is now accomplished with forced air systems that do not use pumps. Larger commercial, public buildings, hospitals, educational facilities, and institutions use hot water heating. Most of these installations are heated with hot water boilers. Smaller hot water systems use in-line circulators as shown in Figure 4 while larger systems may use end suction or axial split case pumps as described in Section 2.2. A recent development is the double suction, in-line pump as shown in Figure 5 with capacities as high as 8000 gpm.

The boilers can be noncondensing or condensing. Condensing boilers are those that operate at lower water temperatures so that the water vapor in the flue gases can be condensed; this results in combustion efficiencies in the range of 85 to 95%. Figure 6 describes a general combustion efficiency curve for condensing boilers based upon return water temperature. Noncondensing boilers are designed of nonferrous materials that do not rust in the presence of water vapor and carbon dioxide (weak carbonic acid). The water temperatures used with these boilers vary from 80 to 150°F (26 to 65°C). Another significant advantage of condensing boilers is the ability to operate at any water flow rate. They do not have a minimum flow rate to eliminate problems such as thermal shock. Therefore, variable speed pumps can operate easily with them. A simple pumping system, known in the industry as variable primary pumping and shown in Figure 3, is used with these boilers.

Noncondensing boilers are those that have been in use for many years; they include cast iron and steel fire tube or water tube boilers. They operate on natural gas with water temperatures above 140°F (60°C) to avoid condensation in the flue collectors. Many of them cannot withstand variable flow, so primary/secondary pumping as shown in Figure 2

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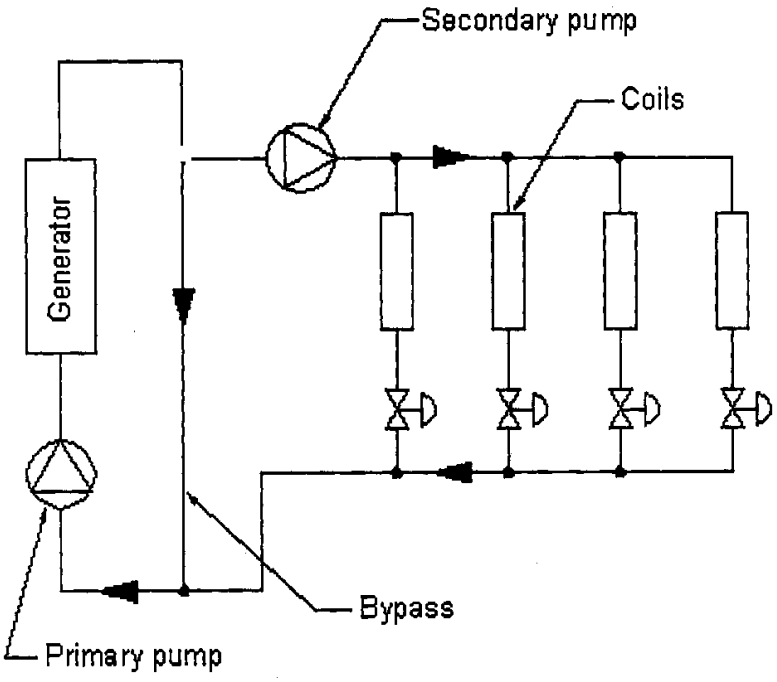


FIGURE 2 Primary/secondary system (Reference 1)

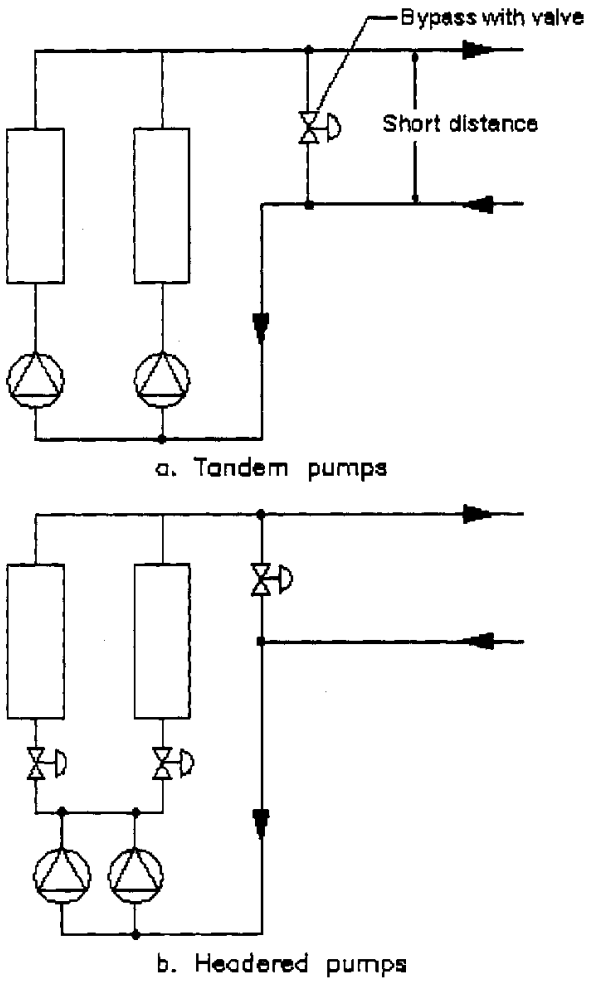


FIGURE 3 Piping arrangements for variable primary pumps (Reference 1)



FIGURE 4 In-line circulator (Flowserve Corporation)

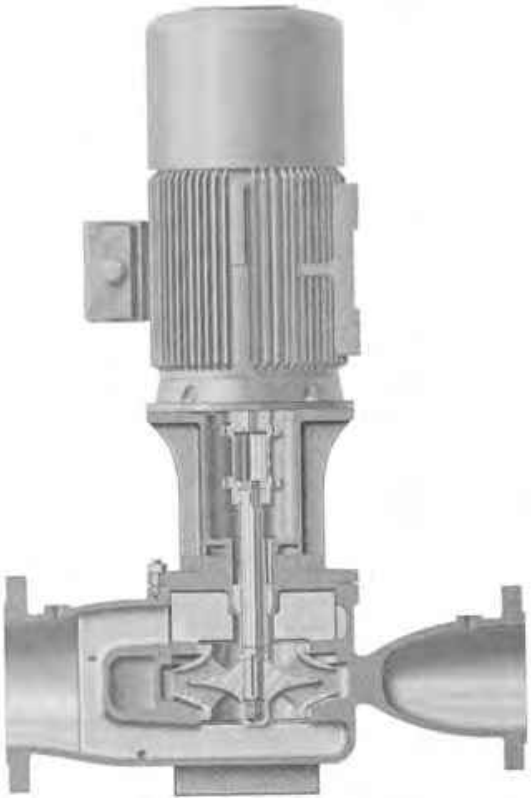


FIGURE 5 Double-suction in-line pump (Flowserve Corporation)

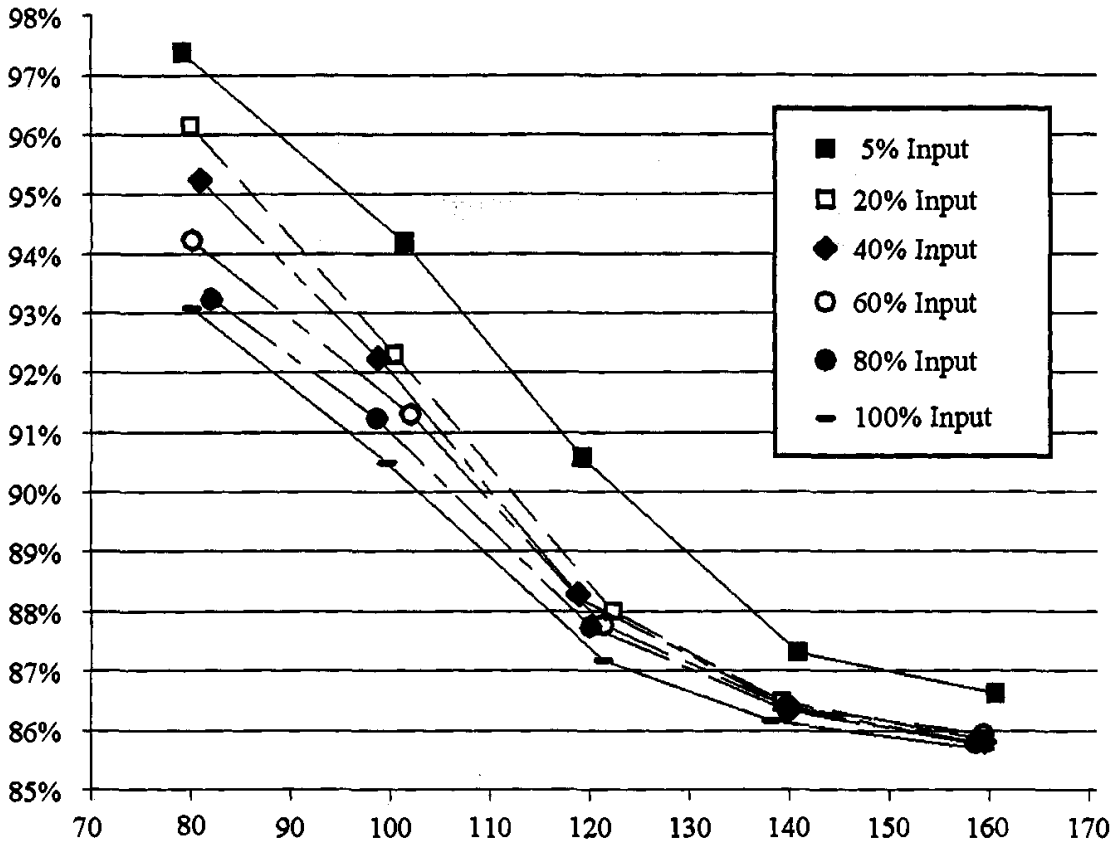


FIGURE 6 Typical thermal efficiency of condensing-type boilers (AERCO International, Inc., Northvale, NJ)

is needed to avoid thermal shock in the boilers. Their combustion efficiency on natural gas is in the range of 78 to 85% at design firing rate. They are being supplanted by condensing boilers due to this lower combustion efficiency.

Air Conditioning Most air conditioning in larger commercial, industrial, and institutional facilities is accomplished by air-cooled or water-cooled chillers. Installations with cooling requirements up to 200 tons utilize air-cooled chillers with closed circuit coolers. Their coefficient of performance (COP) will be in the range 2 to 3. If higher efficiency is required, these installations will have water-cooled chillers with a coefficient of performance of 5 to 6. Continued development of water-cooled chillers has resulted in performance at 0.5 kW/ton or a COP as high as 7. With variable speed drives, these chillers have provided performance at 0.35 kW/ton or COP of 10 at part load.

One of the most important developments in air conditioning is production of water-cooled chillers with the capability of varying the flow in their evaporators (coolers). Digital control of the refrigerant has enabled the chiller designer to allow a flow variation in them from a minimum velocity to design flow. The minimum allowable velocity is subject to the leaving water temperature of the chiller as well as other design criteria. The chiller manufacturer can provide the allowable flow range for a specific machine.

Most water-cooled chillers are electric motor driven even though models are available gas fired or diesel engine driven. The need to acquire higher efficiency has resulted in greater popularity of variable speed units and others with multiple compressors. Figure 7 describes a typical single chiller installation. The chilled water pump is variable speed with a bypass valve to insure minimum allowable evaporator flow. The condenser pump is constant speed excepting on very large installations where the condenser water piping has a sizeable amount of friction.

Multiple chiller installations can be installed with the evaporators in series, parallel, or series/parallel configurations as shown in Figure 8. The size, load range, leaving

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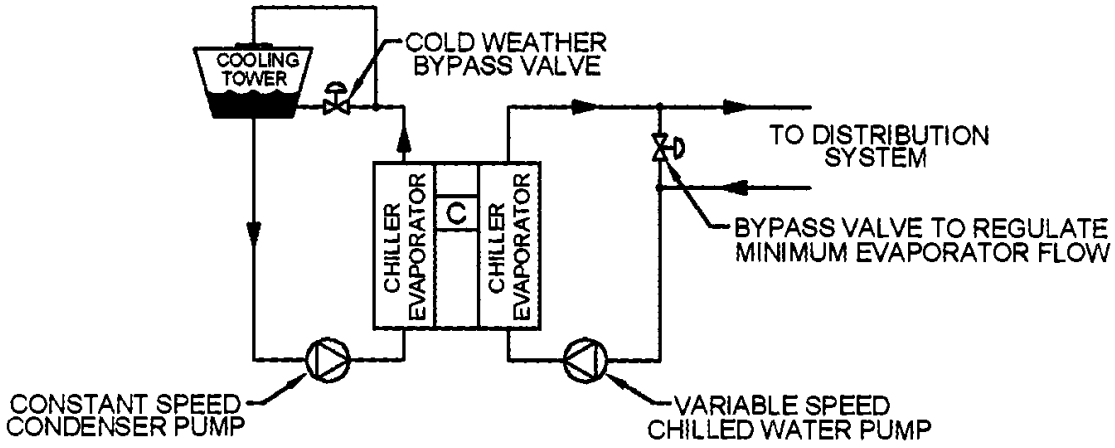


FIGURE 7 Typical installation of a water-cooled chiller

temperature, and temperature differential determine the optimum chiller arrangement. Most installations of two to four chillers are usually installed in parallel. Chilled water pumps were installed in tandem with the chillers in the past. Now, with the capability of great energy savings with variable speed, the pumps are installed in parallel in a header as shown in Figure 8b and 8c.

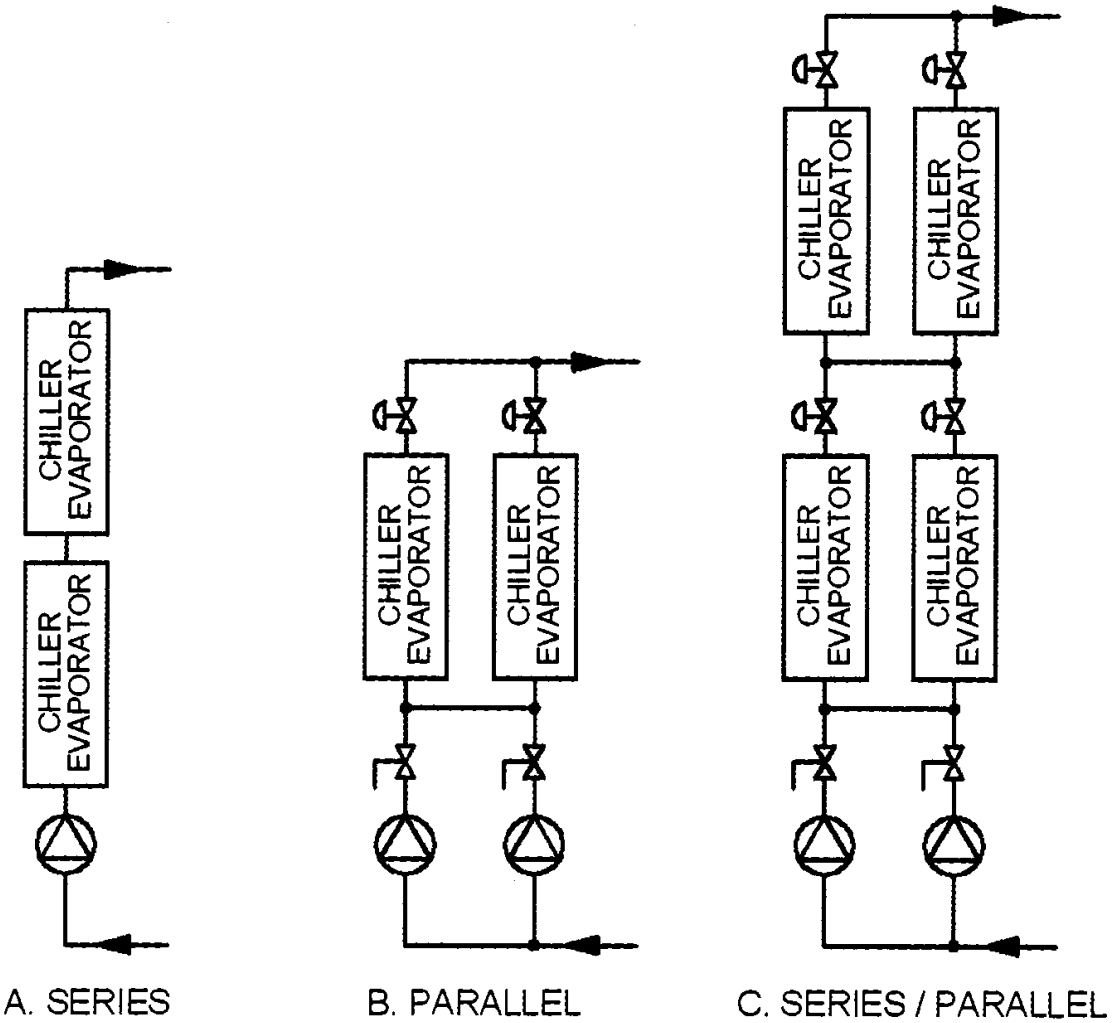


FIGURE 8 Installation of chillers

A significant effort in energy conservation is the recovery of heat from chillers rather than eliminating it by cooling towers. Heat recovery chillers are now available in several different configurations. The design of low temperature heating systems has enabled the use of this recovered heat for space and domestic water heating.

COOLING TOWERS Open cooling towers provide most of the heat rejection means for water-cooled chillers. Closed circuit coolers are available, and first cost determines the type selected unless there are extenuating circumstances such as plume or noise abatement with open cooling towers. Internal sumps may be desirable on extremely cold installations. The internal sump must have a capacity to hold all of the condenser water when the condenser pumps are shut down. It should be located near the top of tall buildings to reduce the required head for the condenser pumps.

Condenser pumps are of the end suction type on small installations; medium size applications use axial split case pumps, while large installations will have vertical turbine or mixed flow pumps in a sump below the towers. All of these pumps are described in Section 2.2 of this book. NPSH available and required must be checked for all condenser pumps operating with open cooling towers or sumps.

HEAT PUMPS Heat pumps have offered an economical first cost installation for multiple housing and school heating and cooling. A typical installation is shown in Figure 9. Their future popularity is contingent upon the cost of electric power. One way of reducing power costs is through the use of a geothermal field as shown in Figure 10.

Distribution of Chilled Water A principal use of pumps in the HVAC field is the distribution of chilled water. Before variable speed and digital control, many complicated circuits using many pumps were used to control the flow and pressure in chilled water systems. Now, variable speed has reduced these systems to simply one set of pumps as shown in Figure 3 on many installations. Campus-type or multiple building such as college or airport applications may require building pumps to reduce the overall energy consumption. Distributed pumping as shown in Figure 11 accomplishes this without excessive pressure on any of the buildings. Small circulators are required at the chillers to maintain the minimum flow needed for the chillers.

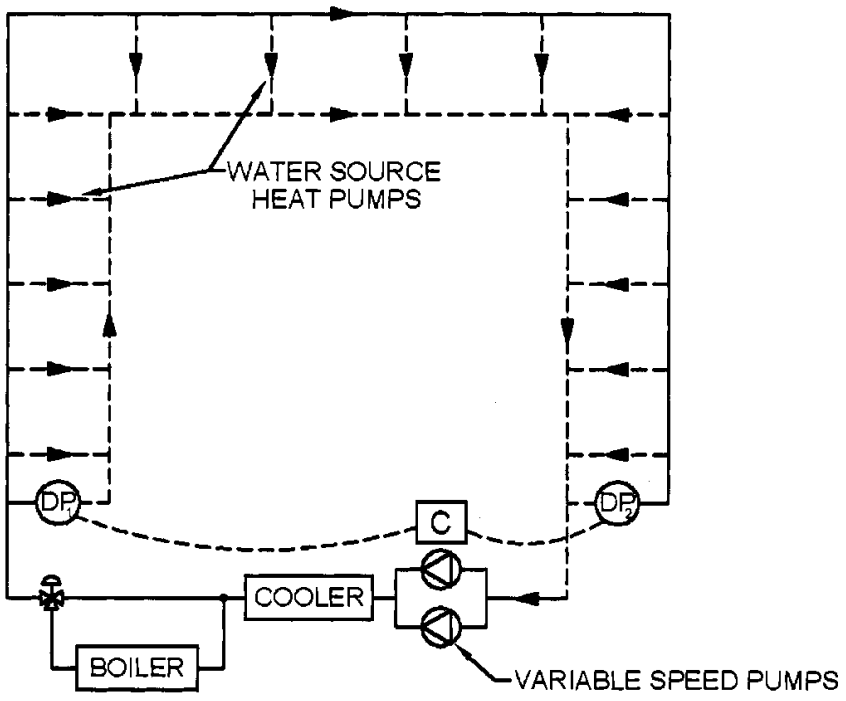


FIGURE 9 Water source heat pumps with closed circuit cooler (Reference 1)

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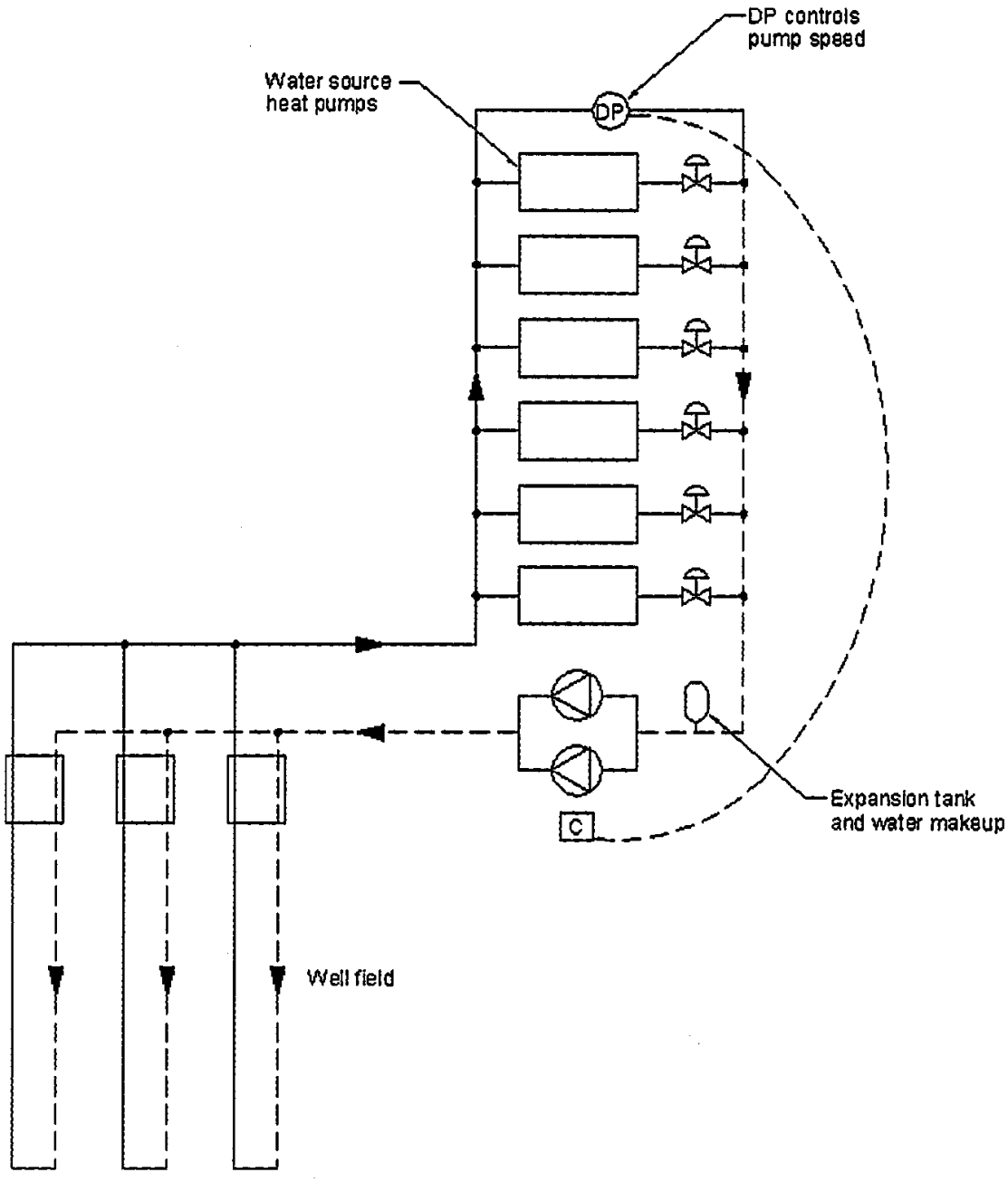


FIGURE 10 Geothermal heat pump installation for low-rise building (Reference 1)

USGBC Green Buildings In an effort to improve the use of energy in buildings, the United States Green Building Council (USGBC) was formed. They have developed the LEED program, Leadership in Energy and Environmental Design. They develop standards and levels of quality to reduce the use of energy in public buildings. They provide Green Building Awards to actual buildings that comply with their standards.

The use of energy in heating and cooling systems is of great concern and, along with USGBC, the American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE) is developing procedures and practices that will conserve energy in all types of buildings.

Summation The HVAC market has become important to the centrifugal pump industry due to the growth of air conditioning. There are many installations of large axial split case pumps as well as vertical turbine and mixed flow pumps. The energy consumed by

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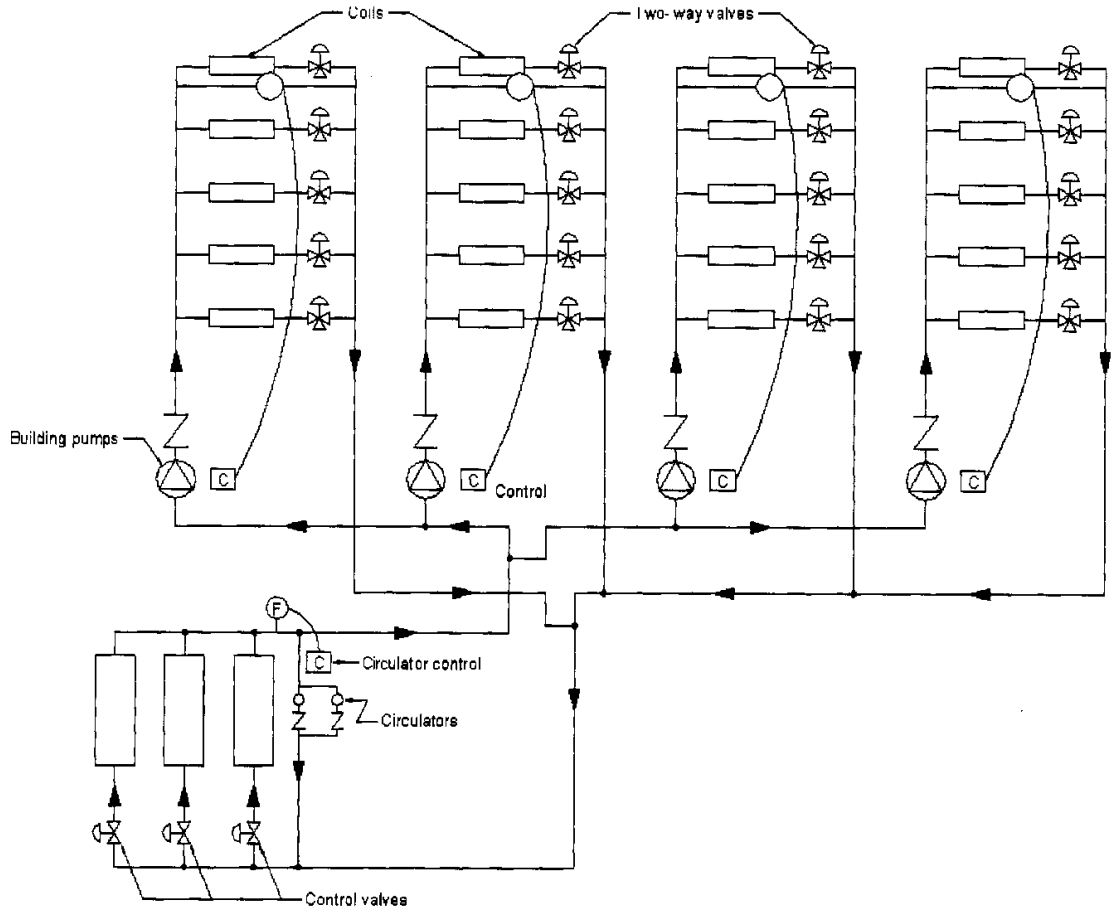


FIGURE 11 Distributed pumping system for four separate buildings (Reference 5)

these pumps along with chillers has resulted in the development of kW/ton for the entire chiller plant, not just the chillers themselves. Figure 12 describes the instrumentation required to acquire a continuous indication of kW/ton for a chiller plant. Due to these efforts, the pumps now used in HVAC applications must be of the same quality and efficiency as those in the general industrial market.

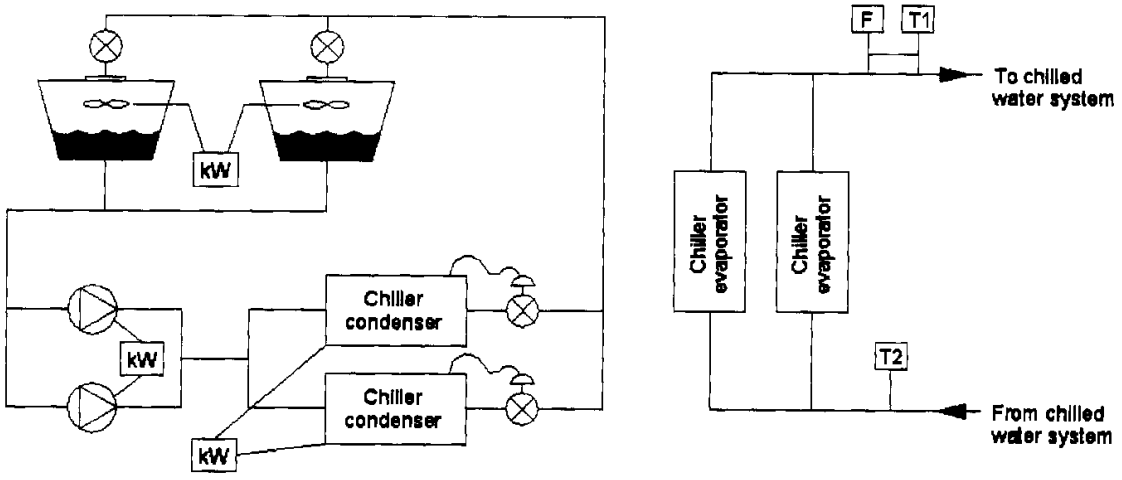


FIGURE 12 Kilowatts per ton instrumentation for central chilled water plant (Reference 1)

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SECTION 12.14
PUMPED STORAGE

C. SAMUEL MARTIN

INTRODUCTION

The variable demand for electrical energy during the day and weekly and the concomitant cost necessitates networks with not only base loads from thermal plants but peaking plants to meet the instantaneous electricity requirement. Ideally, direct storage of electricity would be the most desirable, but currently there are no means of storing large quantities of energy via batteries or other electrical or chemical devices. Presently, peaking loads are satisfied by employing gas turbines or hydroelectric plants, often by means of pumped storage. Pumped storage is a simple concept, requiring two bodies of water at suitable differences of elevation with means of pumping from lower to higher during periods of low electrical demand, followed by generation of power through a hydraulic turbine during periods of high demand.

Compared to steam plants (fossil or nuclear) hydroelectric installations—either conventional or pumped storage—are much faster in meeting electrical demands. An obvious value of pumped storage is its ability to regulate the timing of power generation. Pumped storage accounts for nearly all the effective electrical storage in place around the world, which corresponds to approximately 3% of total electrical generating capacity. For industrialized countries the pumped storage contribution ranges from 2.4% in the United States to approximately 16% for Austria.

The highly reliable and controllable pumped storage technology for fast load change, use as spinning reserve, frequency regulation, and voltage regulation as a synchronous condenser has been recognized and implemented for more than half a century. In Europe, there has been a recent trend of employing reversible pump/turbines rather than separate pumps and turbines—tandem units. In contrast, the two countries with the greatest installed pumped storage capacities—United States and Japan—have utilized only reversible pump/turbines. Figure 1 shows the number of pumped storage plants in selected countries compared with those with pump/turbines.

Pumped storage plants are often designed with utilization of an existing body of water as the lower reservoir in conjunction with a man-made upper pool—termed "pure pumped storage." The term "mixed pumped storage" refers to installations where the power plant may be on a river, or both reservoirs may be natural with water inflow to the upper one. Projects have been planned utilizing abandoned mines or the sea for the lower reservoir. Hence pumped storage need not be restricted to mountainous areas. Many environmental concerns can be addressed by locating all structures underground.

Recently, there has been considerable activity in the planning and construction of pumped storage plants around the world, in particular in Austria, Germany, Japan, and China, as

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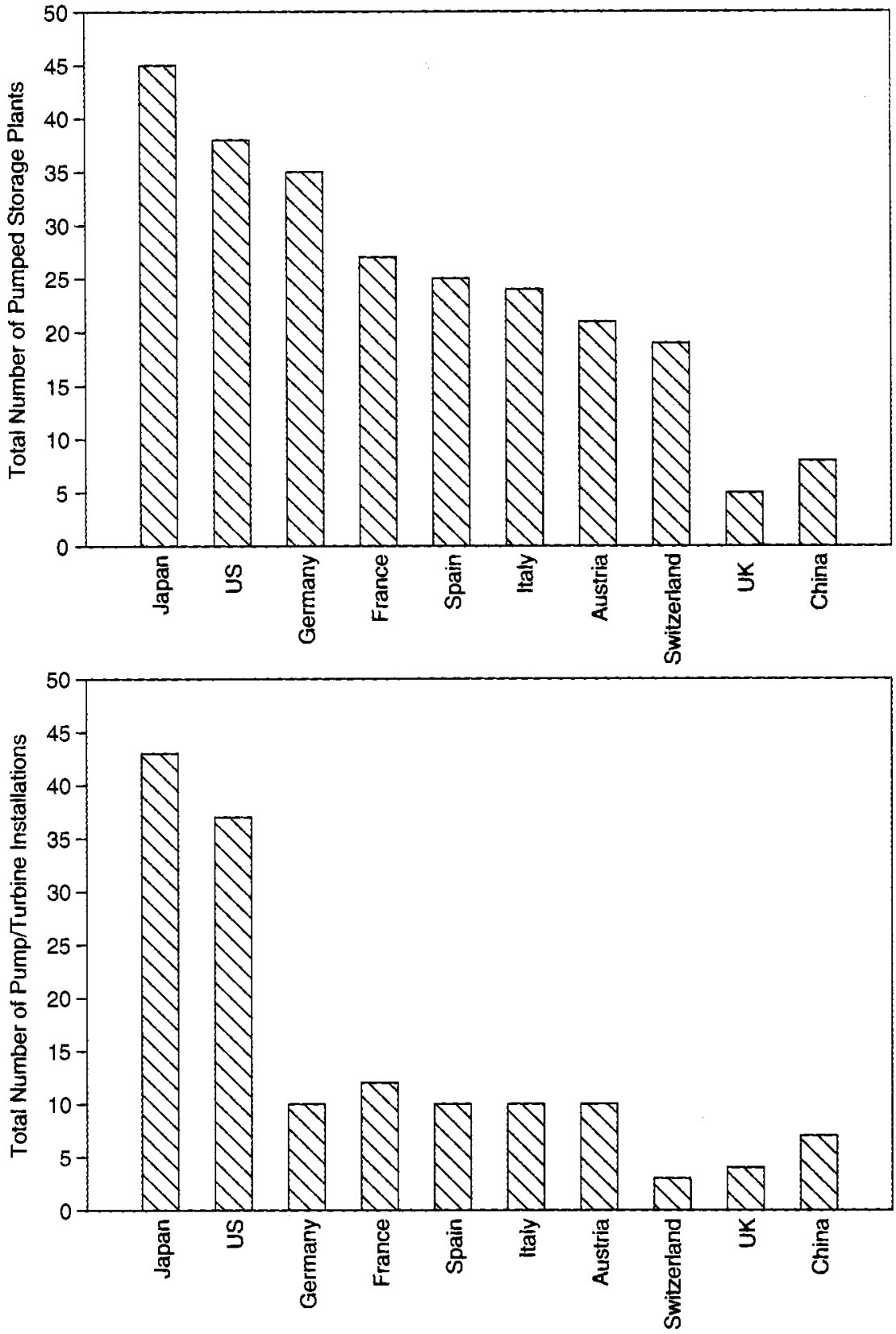


FIGURE 1 Distribution of pumped storage plants and pump/turbines in major countries

well as in smaller countries, but not in the United States. Several large installations in the planning stage in the United States have been postponed or cancelled because of the effect of deregulation of electrical energy generation. However, increasing fuel prices associated with gas turbines employed for peak loads may have an effect in the future.

PHYSICAL LAYOUT OF TUNNELS AND PENSTOCKS

Basically, the pumped storage installation consists of an upper reservoir with intake, inclined or vertical shaft, upper tunnel, possible upper surge tank, penstock(s), powerhouse, possible lower surge tank, tailrace tunnel, and lower reservoir. Three typical conduit configurations are illustrated as schematics in Figure 2. Typography and local geological conditions will play an important role in dictating the configuration. In order to compare the differences from plant to plant Table 1 is prepared to summarize the configuration of eleven pumped storage installations in the United States. For Figure 2a (Bath County and Castiac) the upper tunnel is on a slight slope leading up to the inclined shaft. In both examples the powerhouse is connected directly to the lower reservoir through the draft tube. The existence of the upper surge tank is required to prevent negative pressures in the piping at the knee. Local conditions permitting, a vertical shaft from the upper reservoir to the upper tunnel, as shown in Figure 2b, would eliminate the upper surge tank. A lower surge tank was in fact installed at Raccoon Mountain, but not for Bad Creek. Both plants have an underground powerhouse. Both upper and lower surge tanks were necessary for Plant Helms. The remaining seven plants listed in Table 1 correspond to the schematic layout illustrated in Figure 2c.

CONFIGURATION OF POWERHOUSE MACHINERY

In alpine countries of Europe pumped storage installations before the 1960s were traditionally configured with separate pumps and turbines—called tandem units. Some very old installations utilized the same water conveyances, but perhaps separate powerhouses, necessitating four machines—pump, motor, turbine, and generator. This configuration is referred to as **quaternary** (base 4). The advantage is the speed of changing from generating to pumping and back to generation. Many installations were (and are) configured to be **ternary** (base 3), with pump, motor/generator, and turbine—also referred to as tandem. Figure 3 is a photograph of a double-suction two-stage storage pump in a tandem arrangement. The horizontal axis arrangement for Plant Säckingen with tandem units is shown in Figure 4. Both pump and turbine (Francis or Pelton) rotate in the same direction, and are either directly coupled or linked by torque converter and/or clutch. Fast switching from pump mode to turbine mode is a clear advantage. Disadvantages are (1) requirement of extra hydraulic machine and mechanical equipment, (2) large powerhouse, and (3) maintenance costs. Figure 5 is a vertical axis arrangement of a **ternary** configuration comprised of impulse turbine, motor/generator, torque converter, and storage pump. Table 2 lists the various machine arrangements, coupling, and orientation of axes.

As shown by Figure 1, however, most of the pumped storage installations in the world are comprised of reversible pump/turbines, defined as **binary** (base 2). The two machines are pump/turbine and motor/generator, directly coupled, but normally required to rotate in opposite directions. The exception is a turbomachine that rotates in the same direction in both pump and turbine modes, named Isogyre. Although there are a few installations, this design is restricted to lower capacity applications. The main advantage of the **binary** unit compared to **ternary** is the fact only one hydraulic machine is required. However, there is a sacrifice in efficiency when operating in the turbine mode, but this is becoming less with modern designs influenced by computational fluid dynamics (CFD). Disadvantages to **binary** are (1) switching modes of operation and (2) starting the pump/turbine in the pumping mode. A typical pump/turbine impeller is illustrated by the photograph in Figure 6.

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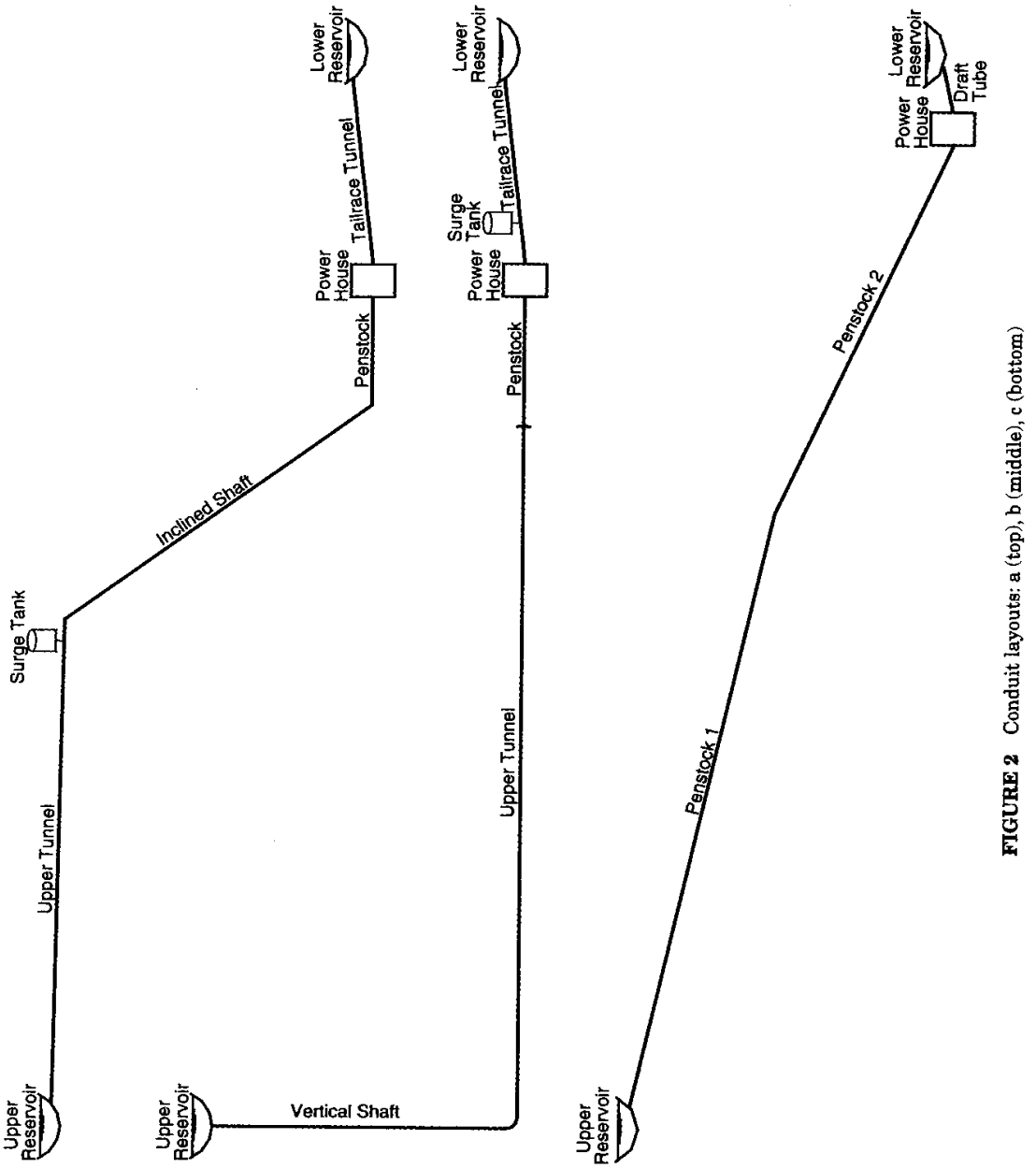


FIGURE 2 Conduit layouts: a (top), b (middle), c (bottom)

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TABLE 1 Lengths of tunnels and penstocks of selected U. S. plants

Plant	Units	Tunnel or penstock length (m) (Angle of tunnel or penstock)								Upper length	Lower length
Bear Swamp	2	228 (–90°)	125 (~0°)	53 (~0°)	▭	152 (~0°)	–	–	–	406	152
Ludington	6	158 (–3°)	246 (–18°)	▭	–	–	–	–	–	396	0
Fairfield	8	115	220	▭	–	–	–	–	–	335	0
Blenheim-Gilboa	4	316 (–90°)	274 (~0°)	597 (~0°)	▭	–	–	–	–	1,186	0
Northfield Mountain	4	259 (–90°)	104 (–0°)	1,563 (–0°)	▭	–	–	–	–	1,926	0
Rocky Mountain	3	173 (–90°)	590 (~0°)	147 (~0°)	▭	–	–	–	–	910	0
Bad Creek	4	278 (–90°)	1,128 (~0°)	121 (~0°)	▭	136 (~0°)	290 (~0°)	–	–	1,527	426
Castiac	6	11,514	⌀	1067	▭	–	–	–	–	12,581	0
Bath County	6	1,103 (–2°)	⌀	302	1,054	320	▭	–	–	2,779	0
Raccoon Mountain	4	274 (–90°)	297	▭	48	⌀	480	–	–	571	528
Helms	3	1,293	2,806	⌀	686	▭	82	⌀	1,180	4,785	1,262

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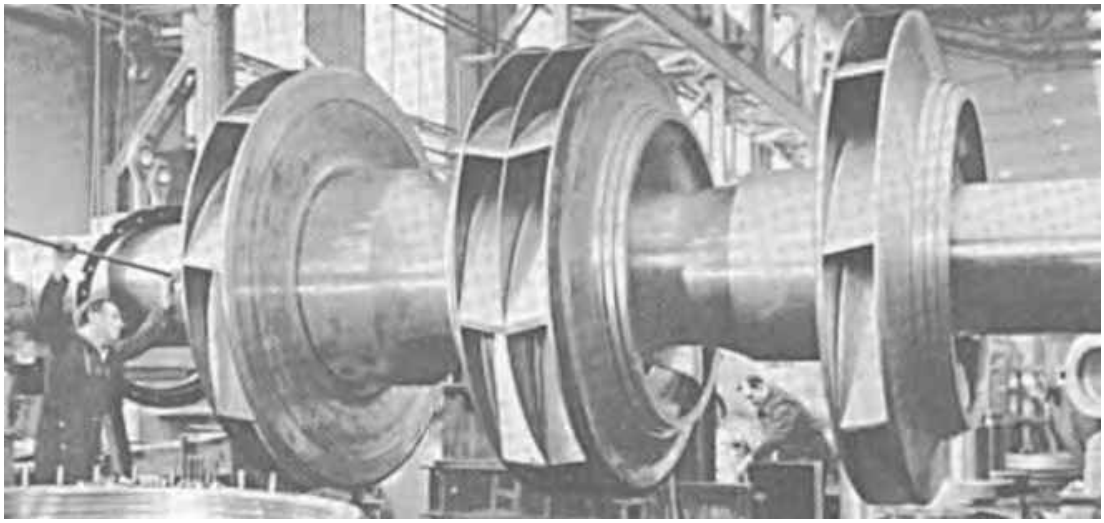


FIGURE 3 View of double-suction two-stage storage pump impeller

PUMP/TURBINE INSTALLATIONS

Selected pump/turbine installations around the world will be listed to emphasize the various aspects of the machines and the various trends. Of the some 38 U.S. plants Table 3 is included to list the salient pump/turbine characteristics of the same installations in Table 1. The last installation finished was Rocky Mountain (1995). Since then, however, numerous plants have been constructed around the world. Figure 7 is the arrangement for a **binary** arrangement of a five-stage pump/turbine for Plant Edolo in Italy. Selected plants with total power capacity of 1200 MW or more are listed in Table 4. Sites with multistage pump/turbines and storage pumps are summarized in Table 5.

PUMP/TURBINE DESIGN

The pump impeller is typically designed for pump mode, resulting in acceptable performance in turbine mode. With considerable enhancements in CFD in recent years, however, improvements are also possible for turbine mode operation. Multistage machines have

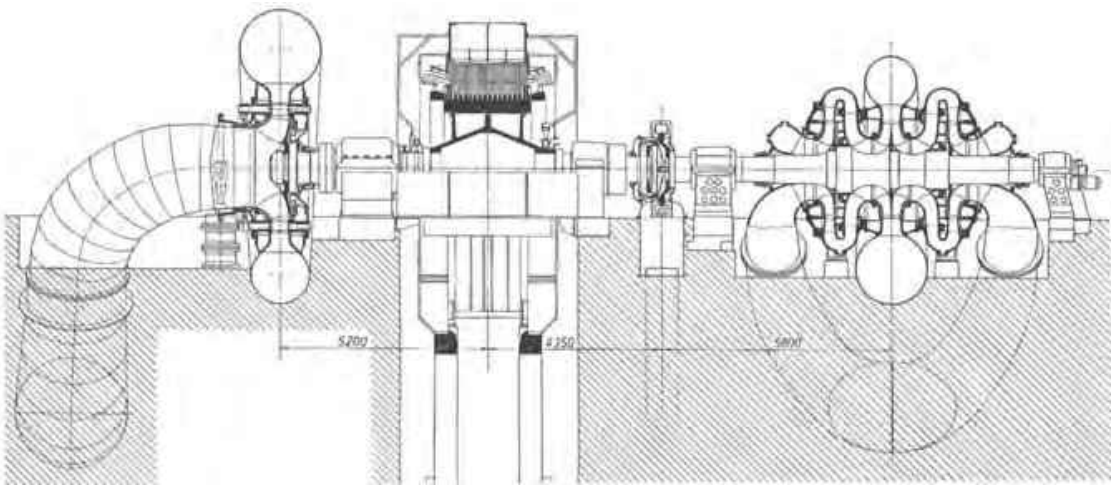


FIGURE 4 Tandem arrangement with Francis turbine and two-stage double-suction storage pump (Plant Säckingen)

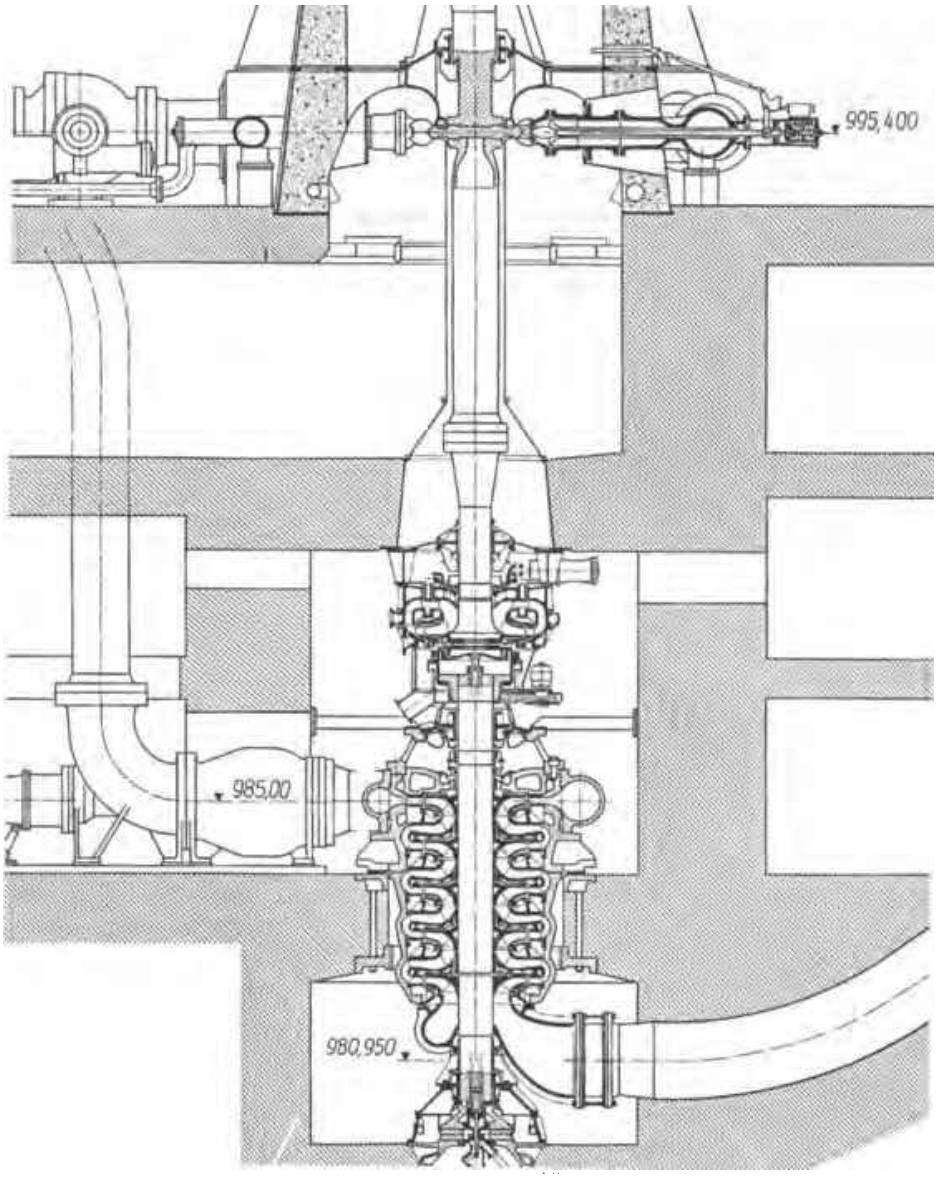


FIGURE 5 Tandem arrangement with Pelton turbine and five-stage storage pump (Plant Lünersee)

TABLE 2 Definition of machine sets

Machine set	Machines	Coupling	Rotation direction	Pump casing	Orientation	Example
ternary	SP-MG-FT	Rigid	Same	Dewatered	VA	Waldeck II
ternary	SP-MG-IT	TC	Same	Filled	VA	Lünersee
ternary	SP-MG-IT	TC	Same	Filled	HA	Säckingen
binary	PT-MG	Rigid	Opposite	Dewatered	VA	Ludington
binary	PT-MG-MS	Rigid	Opposite	Filled	VA	Edolo
binary	Isogyre PT-MG	Rigid	Same	Filled	VA	Malta

SP: Storage Pump; PT: Pump/Turbine; FT: Francis Turbine; IT: Impulse Turbine; MS: Multistage; G: Generator; M: Motor; MG: Motor/Generator; TC: Torque Converter; C: Clutch; VA: Vertical Axis; HA: Horizontal Axis

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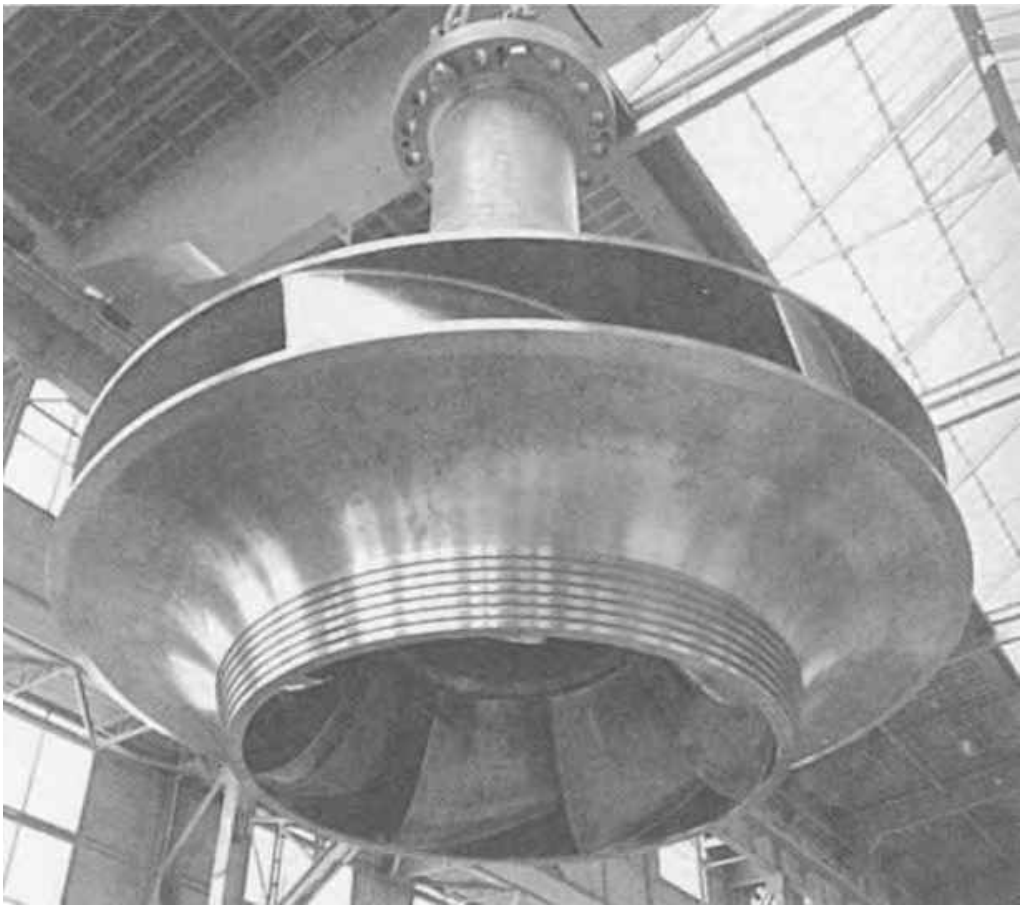


FIGURE 6 Pump/turbine impeller for Plant Palmiet

TABLE 3 Pump/turbine characteristics of selected U. S. plants

Plant	Year	Units	N (rpm)	Unit MW (Pump)	Plant MW	Static head (m) (Pump)	Flow (m^3/s) (Pump)	Specific speed N_{Qp}
Bear Swamp	1974	2	225	300	600	209	125	45.8
Ludington	1973	6	112.5	308	1,980	87	315	70.1
Fairfield	1978	8	150	62	512	49	136	94.4
Blenheim–Gilboa	1973	4	257.1	270	1,000	358	65	25.2
Northfield Mountain	1972	4	257.1	260	1,080	226	102	44.5
Rocky Muntain	1995	3	225	250	760	187	146	53.8
Bad Creek	1991	4	300	270	1,000	380	51	24.8
Castiac	1973	6	257.1	255	1,250	325	64	26.8
Bath County	1985	6	257.1	380	2,100	335	119	35.8
Raccoon Mountain	1978	4	300	400	1,530	305	109	42.9
Helms	1984	3	360	360	1,205	457	71	30.7

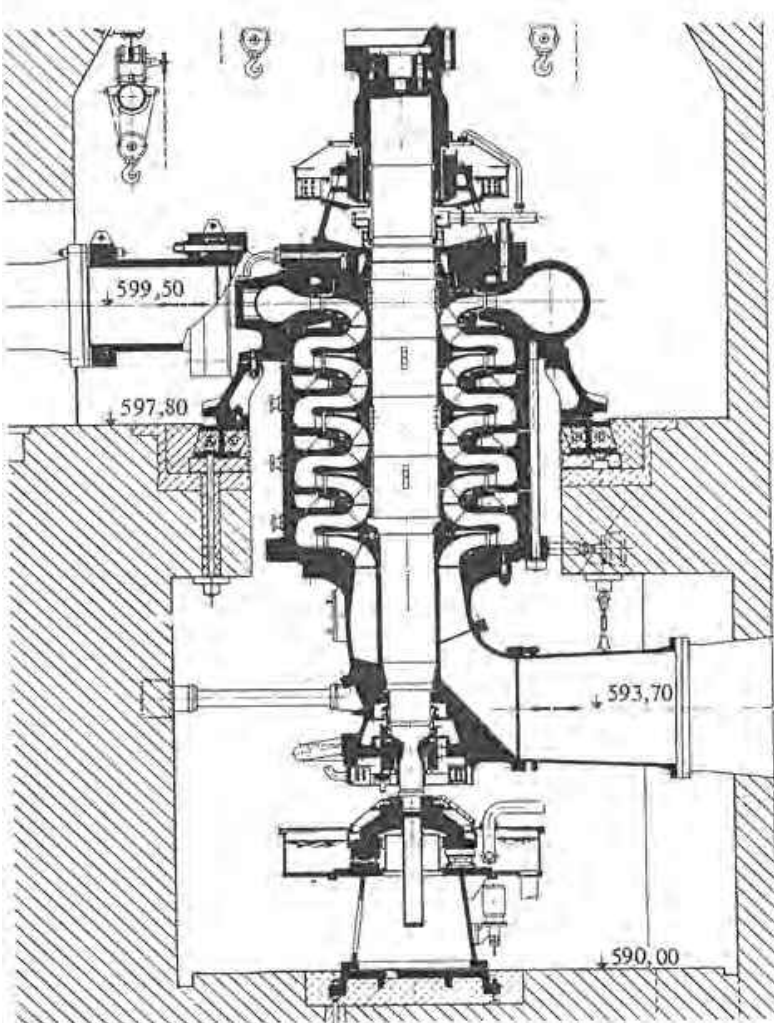


FIGURE 7 Binary arrangement with five-stage pump/turbine (Plant Edolo)

TABLE 4 Selected plants with total pump power exceeding 1,200 MW

Plant	Country	Year	Units	Plant power (MW)	Head (m)	Speed (rpm)
Blenheim-Gilboa	USA	1973	4	1,200	358	257
Ludington	USA	1973	6	1,980	108	112.5
Okutataragi	Japan	1974	4	1,930	380	300
Raccoon Mountain	USA	1978	6	1,530	305	300
Castiac	USA	1978	6	1,250	325	257
Tagasegawa	Japan	1978	4	1,280	229	214
Okuyoshino	Japan	1978	6	1,200	530	514
Dinorwig	UK	1982	6	1,800	545	500
Helms	USA	1984	3	1,200	457	360
Bath County	USA	1985	6	2,100	335	257
Tamahara	Japan	1986	4	1,200	518	428
Ohkawachi	Japan	1993	4	1,280	395	333 to 390
Mingtan	Taiwan	1994	6	1,620	403	400
Tianhuangping	China	1998	6	1,800	608	500
Kazunogawa	Japan	1999	4	1,600	779	500
Guangzhou	China	2000	4	1,200	550	500
Kannagawa	Japan	2005	6	2,820	653	500

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TABLE 5 Selected plants with multistage pumps or pump/turbines

Plant	Country	Machine type	Stages	Unit pump power (MW)	Head (m)	Speed (rpm)
Chiotas	Italy	PT	4	152	1,070	600
Roncovalgrande	Italy	SP-IT	4	94	750	500
Grand Maison	France	PT	4	151	900	600
Edolo	Italy	PT	5	130	1,287	600
La Coche	France	PT	5	80	940	600
Ste Helene	France	PT	5	81	910	600
Lunersee	Austria	SP-IT	5	43	950	750
San Fiorono	Italy	SP-IT	6	110	1,440	500

SP: Storage Pump; PT: Pump/Turbine; FT: Francis Turbine; IT: Impulse Turbine

already exceeded the 1400 m limit. The improvement in hydraulic design through CFD over the years has resulted in an increase in maximum head for single-stage pump/turbines, attaining values approaching 800 m in the pumping mode. This has also led to upgrading of pump/turbine installations around the world. For pump/turbines the specific speed can be defined for either the pump or turbine mode. Since the machine is designed as a pump the definition is based on pump performance to be

$$N_{Qp} = \frac{N\sqrt{Q}}{H^{3/4}} \tag{1}$$

where metric units (rpm · m^3, s, m) are used. The more general dimensionless representation is

$$\Omega_s = \frac{\omega\sqrt{Q}}{[gH^{3/4}]} \tag{2}$$

Correlation of pump/turbine head versus specific speed for a few selected plants is presented in Figure 8. For each plant there are the two limits to head, and corresponding values of specific speed in metric terms. For Francis turbines past correlations have been shown to be approximately represented by

$$H = \frac{\lambda^2}{N_{Qp}^2} \tag{3}$$

where λ is a constant, apparently not formally defined. This correlation is plotted in Figure 8 for $\lambda = 600$ and 800, yielding slopes of –2.0 on log-log plots. Although not precisely calculated the data roughly correspond to a slope of –0.5.

ADJUSTABLE SPEED PUMP/TURBINES

Another enhancement of pump/turbine performance is the implementation of adjustable speed motor/generators, yielding a wider range of higher efficiencies in the pumping mode. Table 6 lists the chronology of variable-speed units, starting with Japan, and recently being employed in Germany (Goldisthal 2002), to be followed in Austria with Limberg II (2012). Variable-speed units can widen the cavitation characteristics in the pumping mode, as well as provide easier start-up in the pumping mode. Moreover, the broader range of performance provides more operating flexibility.

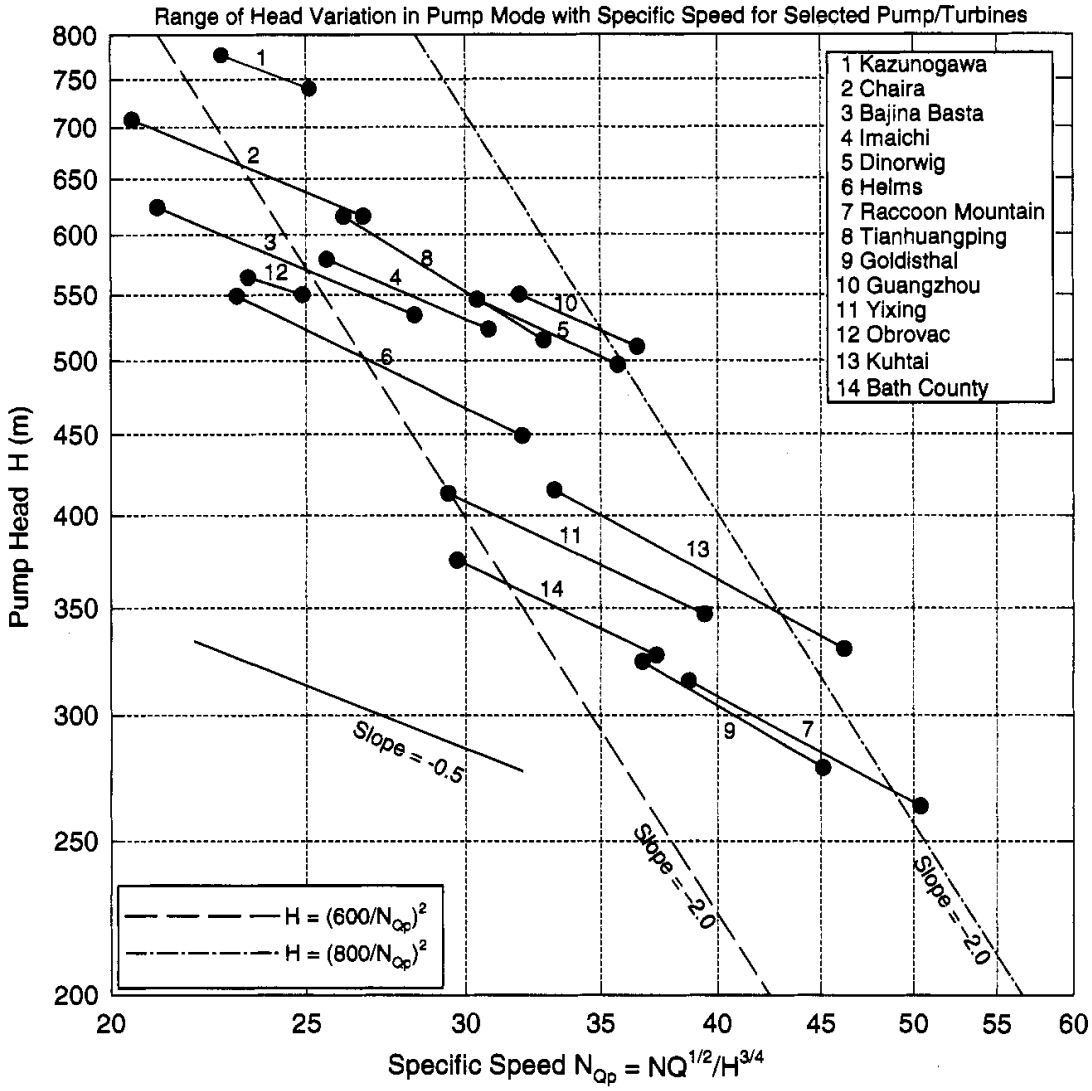


FIGURE 8 Pump head versus specific speed for selected pump/turbines

TABLE 6 Selected plants with adjustable speed motor/generators

Plant	Country	Year	Pump power (MW)	Pump head (m)	Speed range (rpm)
Yagisawa	Japan	1990	53 to 82	63 to 113	130 to 156
Takami	Japan	1993	44 to 140	86 to 118	200 to 254
Ohkawachi 3 & 4	Japan	1993	240 to 400	432	330 to 390
Shiobara 3	Japan	1995	200 to 330	330 to 387	356 to 394
Okukiyotsu 2	Japan	1996	230 to 340	445 to 494	407 to 450
Kazunogawa	Japan	2001	482	650 to 708	500+/–
Okumino 5 & 6	Japan	2000	270	520	514+/–
Goldistahl	Germany	2004	168 to 289	279 to 334	300 to 347
Limberg II	Austria	2012	240	360	–

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Finally, for heads exceeding 800 m the designer has to choose between (1) multistage pump/turbines or (2) tandem units comprised of multistage (storage) pumps and either impulse or Francis turbines. Examples of unregulated machines are summarized in Table 5.

STARTING PUMPS

The starting of storage pumps or pump/turbines in the pumping mode to an eventual synchronous speed can be accomplished in various ways. For **ternary** units with storage pump and Francis turbine rigidly connected on same shaft, rigid pump starting can be accomplished with dewatered pump casing (Waldeck II: Table 2). For **binary** sets pump/turbines are most often started with the pump casing dewatered. Plants Chiotas and Edolo are exceptions in that either a dedicated Francis turbine is used or back-to-back employing adjacent impulse turbine with the pump casing filled. Various techniques for pump starting are:

- Across the line
- Asynchronous starting
- Back to back using adjoining turbine unit or dedicated Francis or Pelton turbine
- Back to back for multistage unregulated pump/turbine with spherical valves with bypass valve for fine control
- Reduced voltage starter
- Use of pony motor with dewatered pump casing

FOUR QUADRANT CHARACTERISTICS

For transient simulation hydraulic characteristics are required in both pump and turbine regions of operation, as well as in transitory regions. Reverse flow and reverse rotation are also possible for storage pumps. For pump/turbines, however, steady-state operation in pump and turbine mode are definitely required. Typically, model studies are conducted for pump/turbines over a wide range of machine operation for wicket gate (guide vane) openings. The model data are generally represented in terms of nondimensional parameters in so-called four quadrants, with the axes being speed and flow. The same data are sometimes correlated in terms of head coefficient and flow coefficient.

Figure 9 depicts eight zones of possible operation of a pump or pump/turbine in four quadrants. As listed in Table 7 Quadrant I contains normal pumping via Zone A, while normal turbining is Zone G in Quadrant III. In a model test stand the machine can be made to operate in all four quadrants, even with those zones with negative head. For a prototype unit operating as a pump, loss of power will mean the pump starts in Zone A (Quadrant I), enters Zone H (Quadrant IV) as flow reverses, then Zone G (Quadrant III) as the machine reverses rotation, during which time the wicket gates are normally closing. If the transient persists until the pump passes zero torque (runaway) before the gates are closed then the unit enters so-called braking, called Zone F. If the servomotor were to fail, or if the pump/turbine was initially generating (Zone G in Quadrant III) there is a possibility of the machine entering Zone E (Quadrant II), called reverse pumping.

As shown in Table 7 Zones B, C, and D correspond to negative head, a condition only possible to be contrived in the laboratory model. Consequently, model data are typically only reported for positive heads, represented by Zones A, H, G, F, and E. In order to compute speed change during the transient not only head-flow data are needed, but also torque.

HYDRAULIC TRANSIENT ANALYSIS

The potential for waterhammer exists in pumped storage installations mainly due to disconnections from the electrical grid. For preliminary analyses the three worst scenarios

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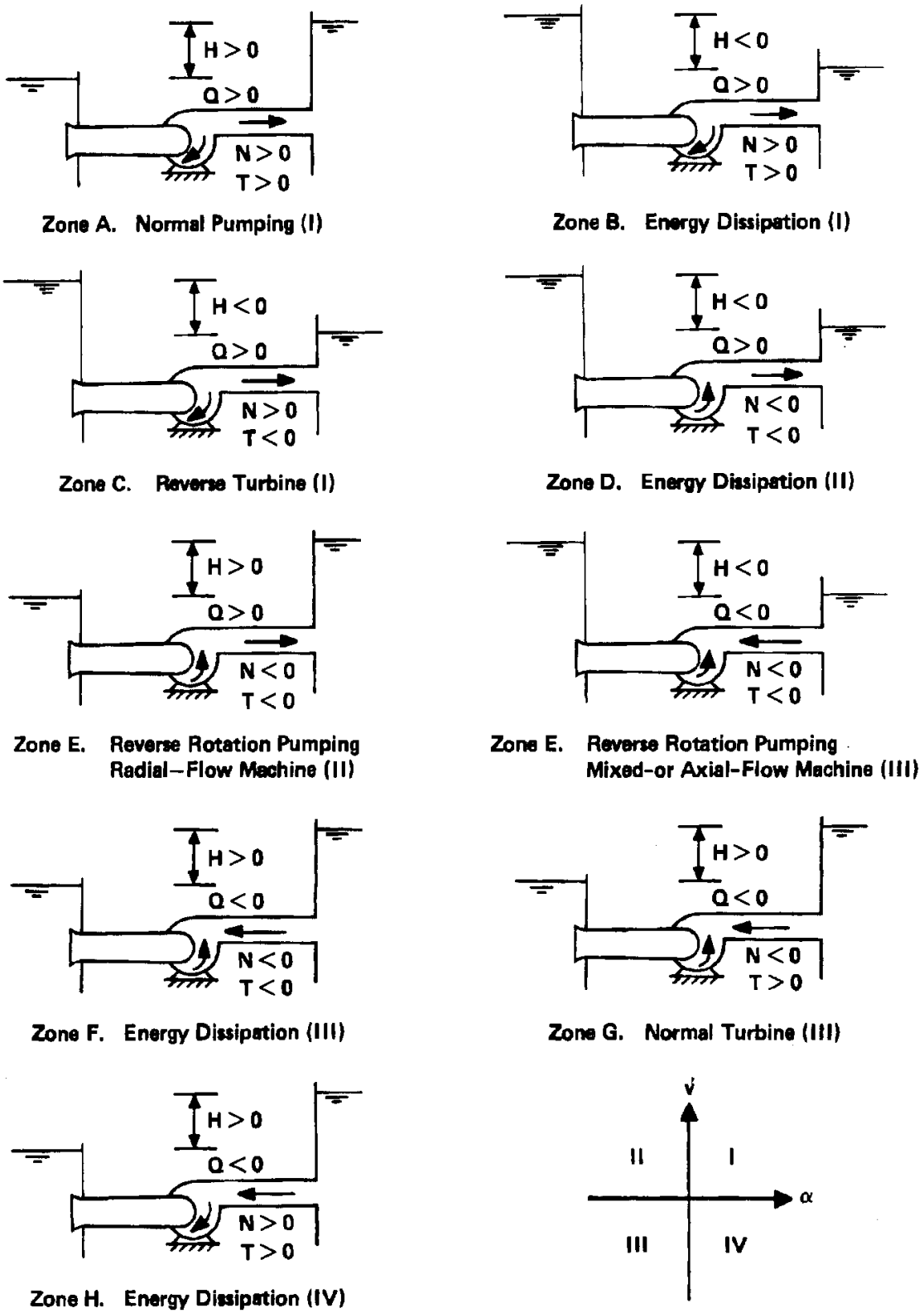


FIGURE 9 Definition of zones and signs of variables for four quadrants

are (a) full load rejection of all operating units, (b) pump power failure, and (c) load acceptance. The possibility of negative pressures in the upper tunnel and shaft must be investigated for pump power failure and load acceptance. For the layout in Figures 2a and 2c an upper surge tank may be necessary. For long tailrace tunnels load rejection and/or pump start-up from no load, or other transient scenarios, may require a lower surge tank.

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TABLE 7 Definition of zones and quadrants

Quadrant	Zone	Mode	Energy	Rotation	Flow	Head	Torque
I	A	Normal pumping	Productive	$N > 0$	$Q > 0$	$H > 0$	$T > 0$
I	B	Energy dissipation	Dissipative	$N > 0$	$Q > 0$	$H < 0$	$T > 0$
I	C	Reverse turbining	Productive	$N > 0$	$Q > 0$	$H < 0$	$T < 0$
II	D	Energy dissipation	Dissipative	$N < 0$	$Q > 0$	$H > 0$	$T < 0$
II	E	Reverse pumping	Productive	$N < 0$	$Q > 0$	$H > 0$	$T < 0$
III	F	Braking	Dissipative	$N < 0$	$Q < 0$	$H > 0$	$T < 0$
III	G	Normal turbining	Productive	$N < 0$	$Q < 0$	$H > 0$	$T > 0$
IV	H	Energy dissipation	Dissipative	$N > 0$	$Q < 0$	$H > 0$	$T > 0$

Raccoon Mountain is an example. Plants Helms (USA) and Kannagawa (Japan) needed both upper and lower surge tanks. (See Section 11.3).

For final analyses more transient cases may be warranted, as listed below

- Full load rejection with guide vane closure
- Full load rejection with main inlet valve (MIV) closure and locked guide vanes
- Full load acceptance from speed-no-load
- Pump power failure with guide vane closure
- Pump power failure with main inlet valve closure
- Pump start-up from synchronization
- Normal turbine shutdown
- Normal pump shutdown
- Reduction of generating load
- Increase in generating load
- Load rejection followed by load acceptance
- Load acceptance followed by load rejection
- Load acceptance followed by normal shutdown
- Malfunction of governor cushioning stroke, resulting in higher closure rate over entire stroke
- Guide vane closure from speed-no-load
- Governor stability
- Resonance phenomena

Load Rejection and Pump Power Failure Example Bad Creek Plant in the United States is chosen as an example for hydraulic transient simulation. The piping layout and pertinent pump/turbine data are listed in Tables 1 and 3, respectively. The vertical shaft is 278 m, followed by a nearly horizontal tunnel of 1128 m, leading to four penstocks of 121 m each. From hydraulic model test data for eleven wicket gate openings ranging from 3 to 40 mm four-quadrant curves are plotted in Figures 10 and 11, for unit flow versus unit speed, and unit torque against unit speed, respectively. Nondimensional definitions are employed for all three quantities, as follows. For a pump impeller with diameter D rotating at ω the peripheral speed is U, yielding a speed coefficient K_{u1}

$$K_{u1} = \frac{U}{\sqrt{2gH}} = \frac{\omega D}{2\sqrt{2gH}} = \frac{1}{\sqrt{\Psi}} \tag{4}$$

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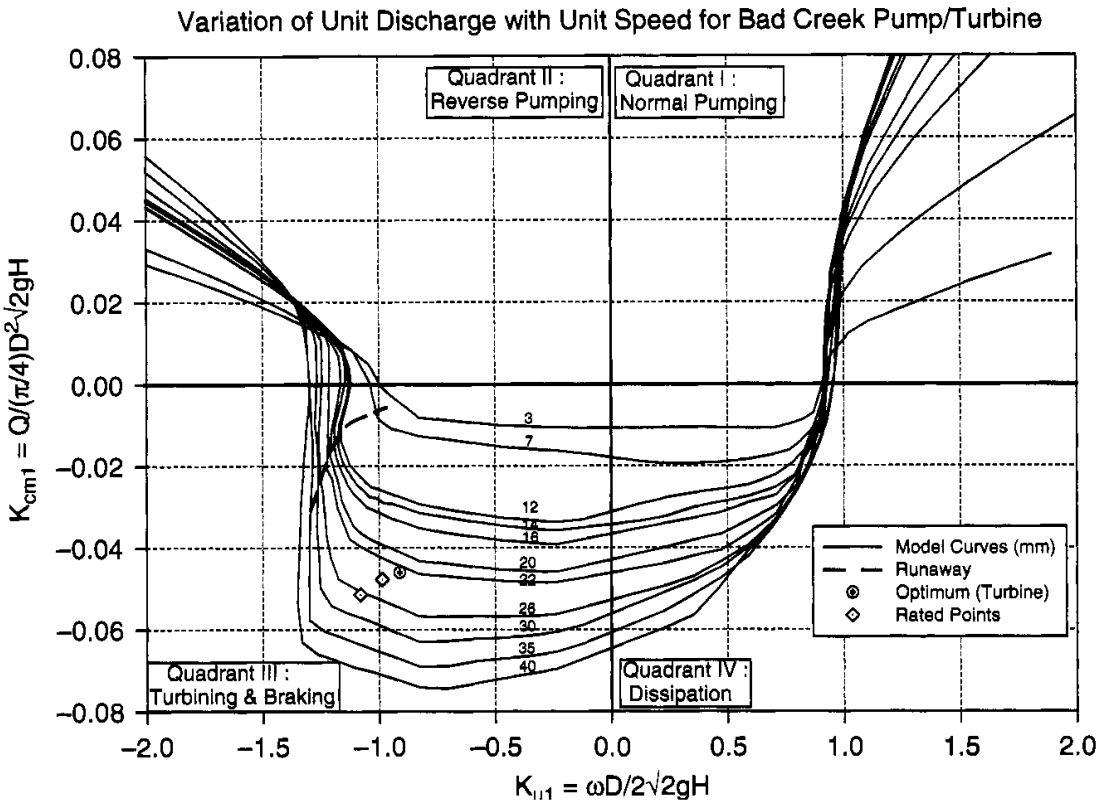


FIGURE 10 Correlation of unit discharge with unit speed for Bad Creek pump/turbine

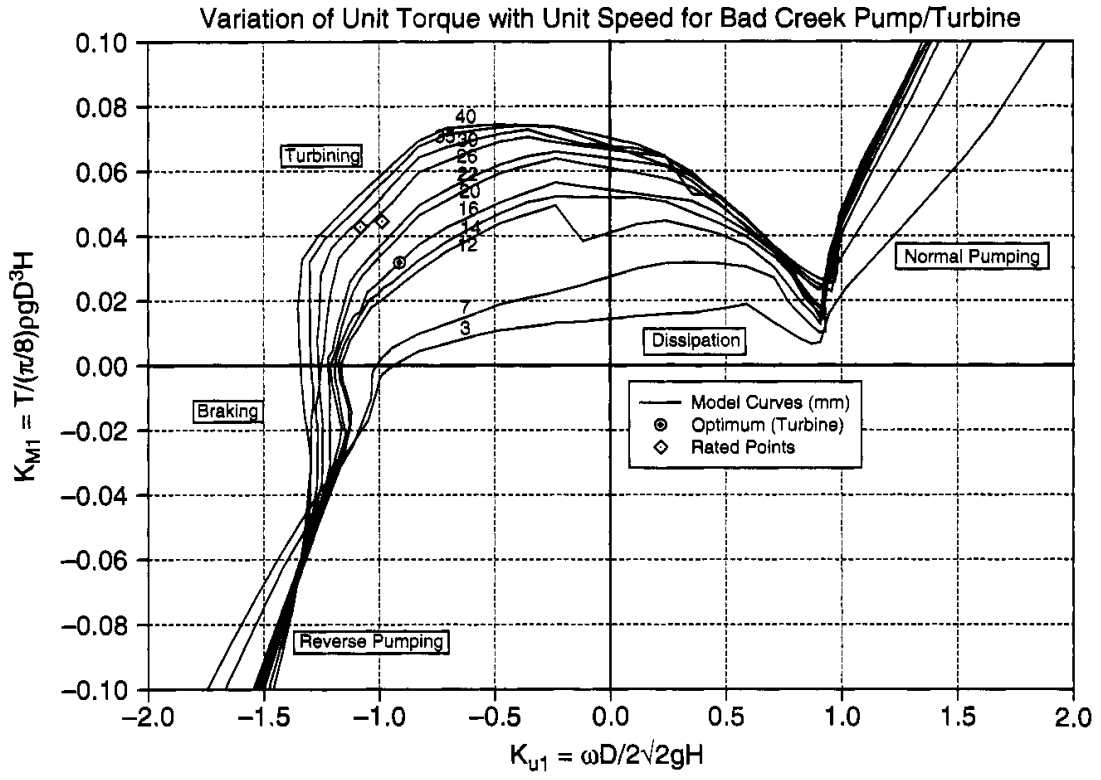


FIGURE 11 Correlation of unit discharge with unit torque for Bad Creek pump/turbine

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Dividing the flow Q by an area based on diameter D results in a unit discharge parameter K_{cm1}

$$K_{cm1} = \frac{V}{\sqrt{2gH}} = \frac{4Q}{\pi D^2\sqrt{2gH}} \tag{5}$$

The unit torque is defined as

$$K_{M1} = \frac{8T}{\pi\rho g D^3 H} \tag{6}$$

Figure 10 shows the four quadrants and Zone A (Quadrant I), Zone H (Quadrant IV), Zones G and F (Quadrant III), and Zone E (Quadrant II). The dashed runaway line in Quadrant III is the line of zero torque, separating Zones G and F. Reverse pumping is Zone E in Quadrant II. The white space between Zones A and E corresponds to negative head, which is not tested for pump/turbines. The correlation of the wicket gate opening with the servomotor stroke is plotted in Figure 12 in model units. Although the model was tested for wicket gate openings up to 40 mm, the full gate or stop position for the prototype is 35 mm in model units.

For four 300 rpm units generating at full gate (35 mm) at static head of 375 m the unit flow is 121.3 m^3/s with a power output of 358 MW. Full load rejection results are plotted in Figure 13, showing maximum speed of 475 rpm. The path of all four machines in Figure 14 falls on the 35 mm curve until the delayed wicket gate action at T = 12 s. The pump/turbines pass through runaway into the braking region, then back into positive torque. Finally, the units progress into reverse pumping before attaining a gate opening of 3 mm.

Pump power failure results are plotted as time histories in Figure 15 and transient path in unit flow unit speed four quadrant diagram in Figure 16. For this mode the flow is 67.3 m^3/s for a static head of 375 m.

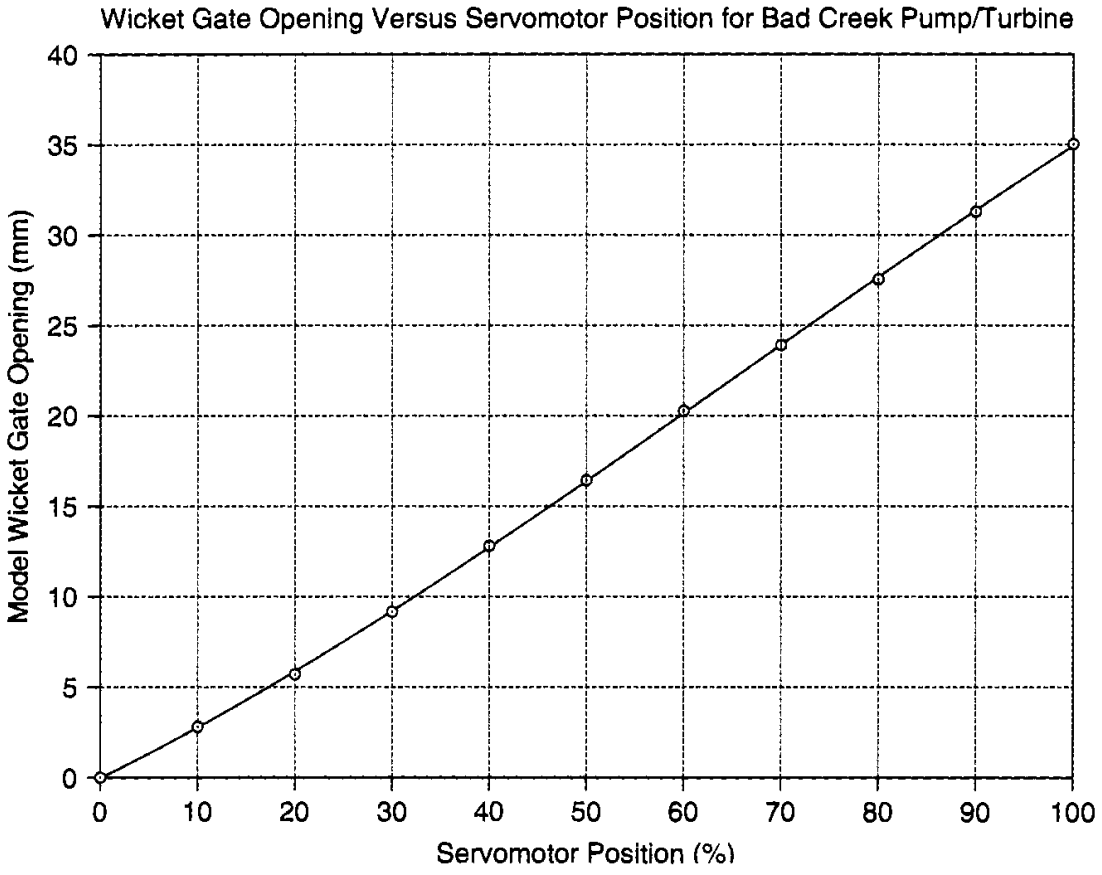


FIGURE 12 Correlation of wicket gate opening with servomotor position.

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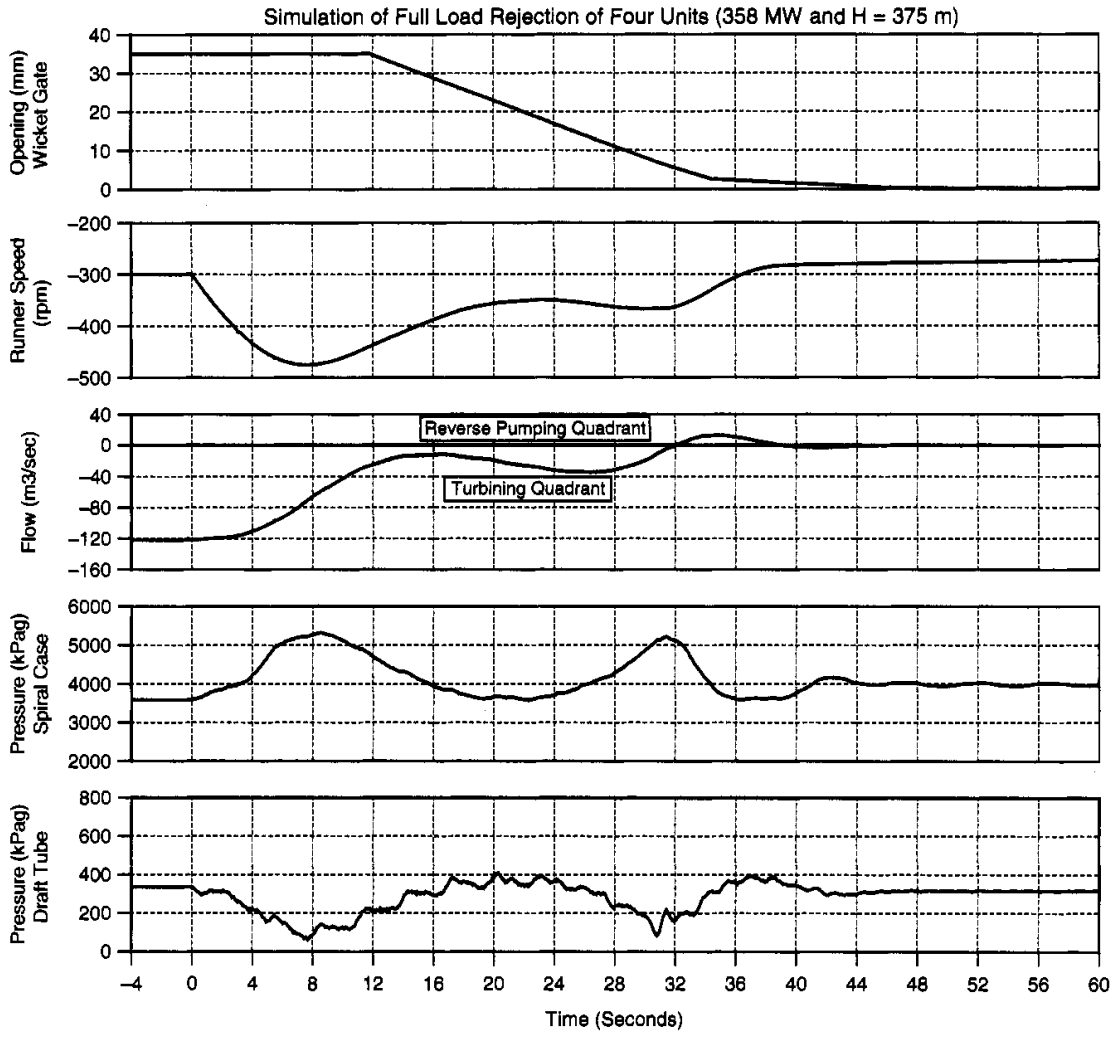


FIGURE 13 Time histories of four unit load rejection for Bad Creek

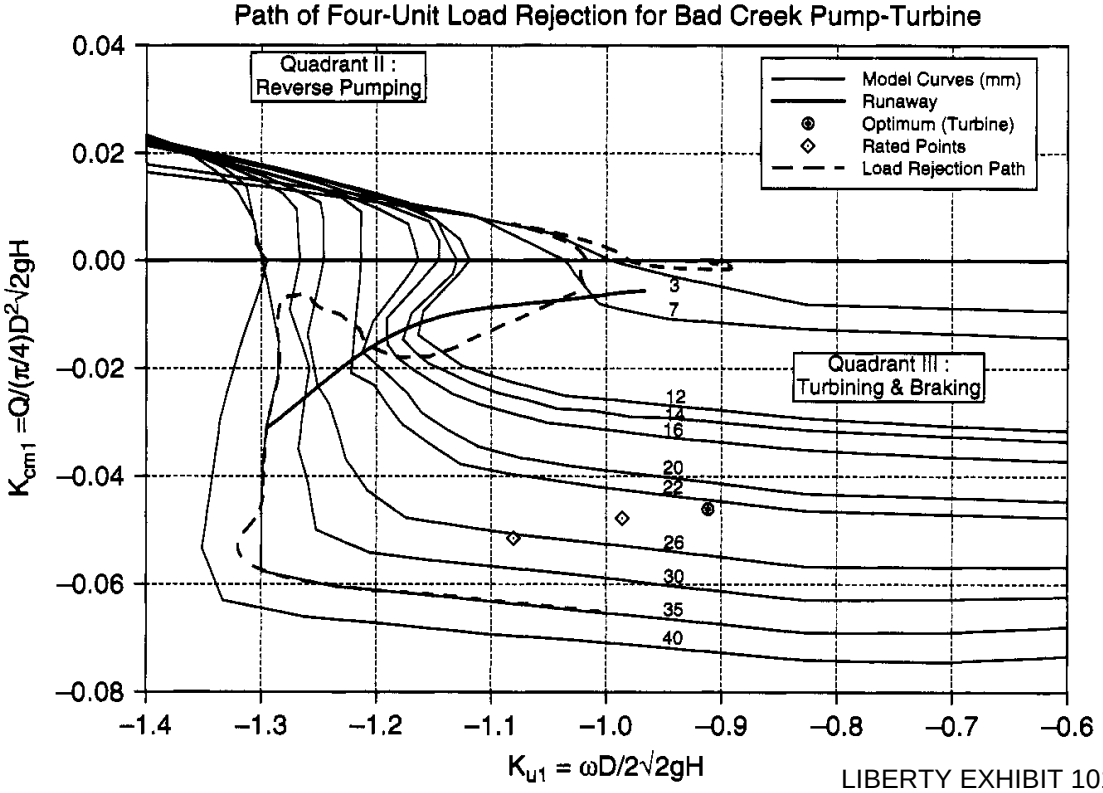


FIGURE 14 Four-quadrant path for four unit load rejection for Bad Creek

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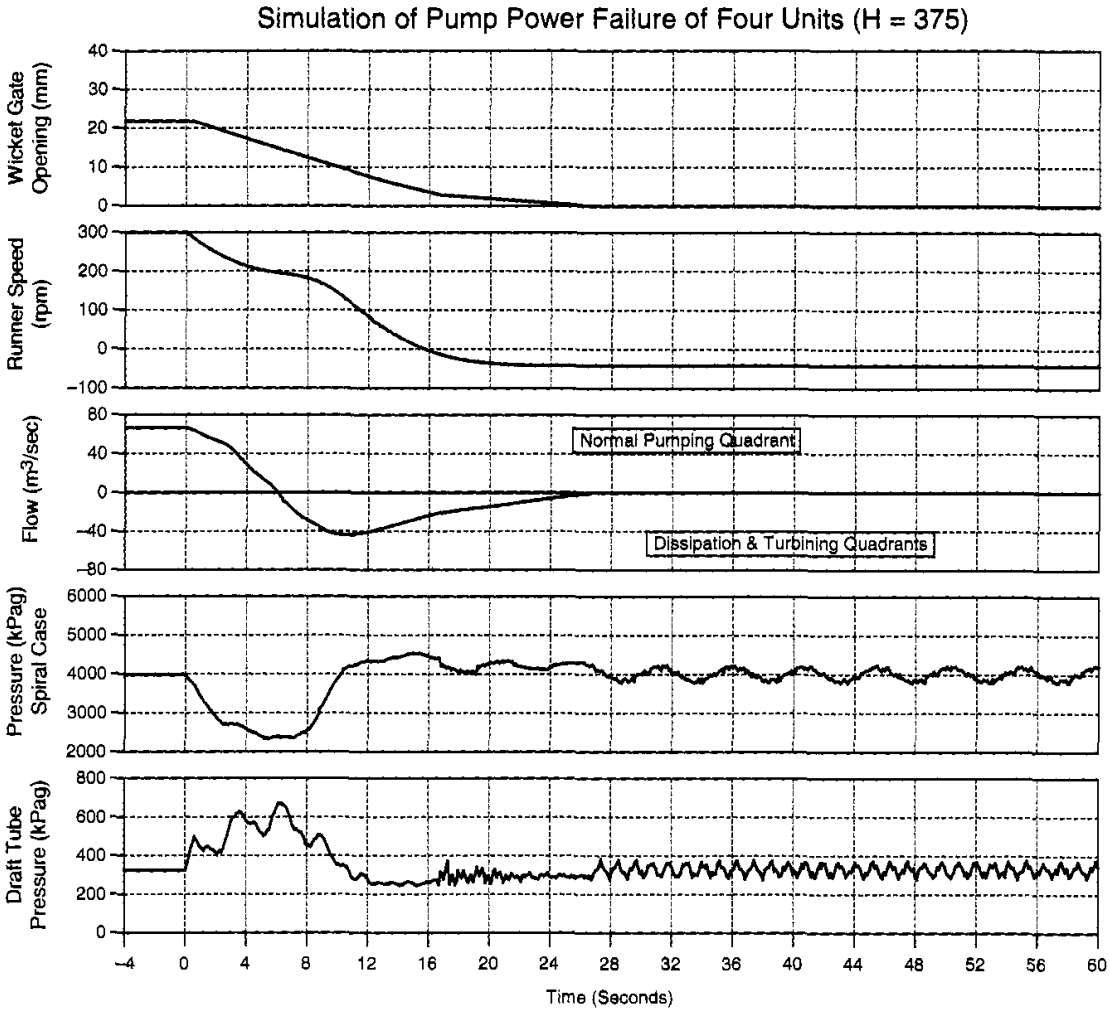
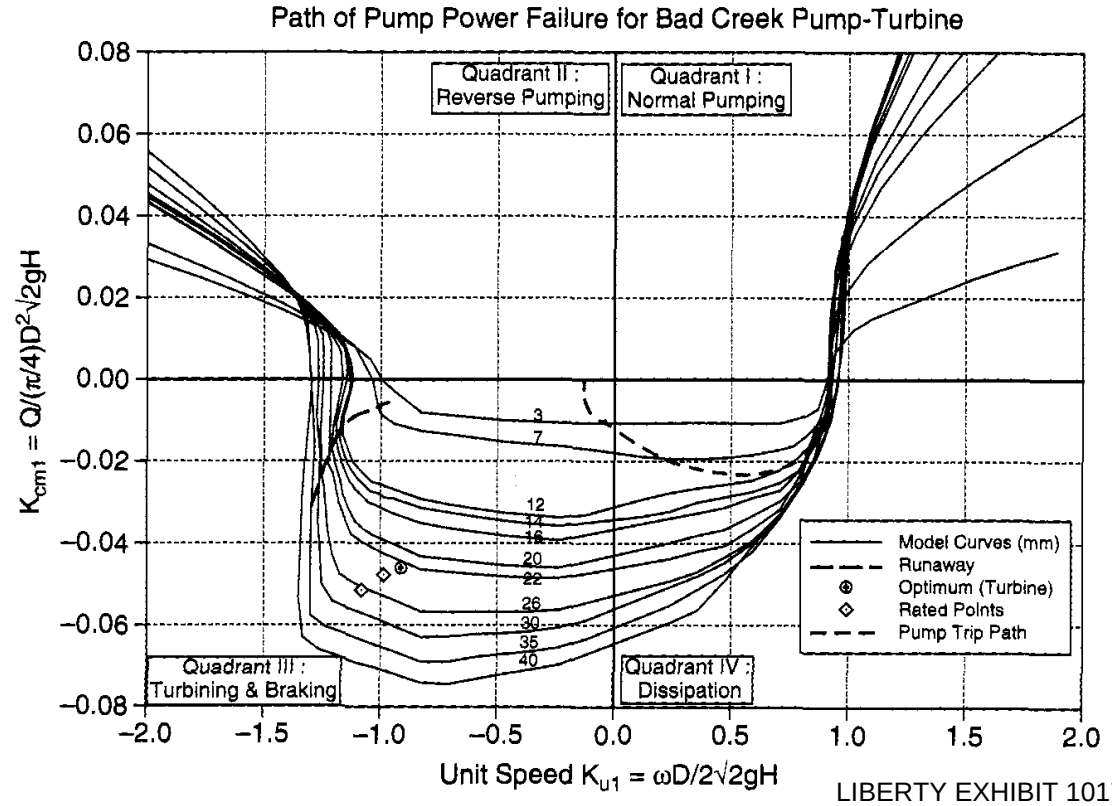


FIGURE 15 Time histories for four unit pump power failure for Bad Creek



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FIGURE 16 Four-quadrant path for four pump power failure for Bad Creek

SECTION 12.15
METERING

ROBERT H. GLANVILLE
KEN W. TAYLOR

METERING OR PROPORTIONING

Conventional reciprocating pumps can be adapted to function as metering or proportioning devices in the transfer of liquids. The principal adaptation is the addition of a means of varying the pumping rate and predicting what that rate will be, which makes the modified units suitable for use as final control elements in continuous-flow processes. Metering pumps are often employed where two or more liquids must be proportioned or where mixture ratios must be controlled. These effects are achieved by changing the displacement per stroke (by moving the crankpin by special linkages or by partial stroking) or by changing the stroking speed through the use of variable-speed transmissions or electric motors. Accurate cyclic volume compensation can also be performed using electronic calibration.

Four basic types of positive displacement reciprocating pumps, and several variations, are used for this service: packed plunger pumps, pumps with a mechanically actuated diaphragm, pumps with a hydraulically actuated diaphragm, and pumps with hydraulically actuated pistons.

Packed Plunger The packed plunger pump is the most commonly used type because of its relatively simple design and wide range of pressure capability. It is an adaptation of the conventional reciprocating transfer pump (Figure 1). Its advantages are

1. Relatively low cost
2. Pressure capability to 50,000 lb/in^2 (345 MPa) gage
3. Mechanical simplicity
4. Wide capacity range, from a few cubic centimeters per hour to 20 gpm (4.5 m^3/h)
5. High accuracy, better than 1% over a 15:1 range
6. Only slightly affected by changes in discharge pressure

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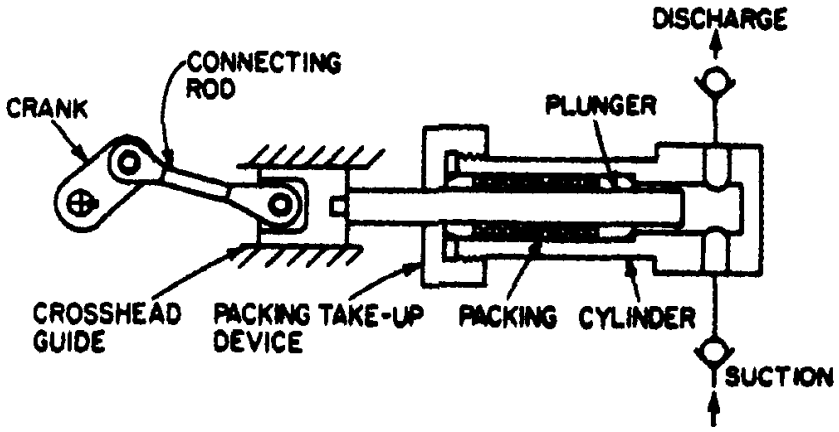


FIGURE 1 Packed plunger pump

Its disadvantages are

1. Packing leakage, making it unsuitable for corrosive or dangerous chemicals
2. Packing and plunger wear and the resulting need for gland adjustment
3. Inability to pump abrasive slurries or chemicals that crystallize

Mechanically Actuated Diaphragm The mechanically actuated diaphragm pump is commonly used for low-pressure service where freedom from leakage is important. This pump utilizes an unsupported diaphragm, which is moved in the discharge direction by a cam and returned by a spring (Figure 2). Its advantages are

1. Relatively low cost
2. Minimum maintenance at 6- to 12-month intervals
3. Zero chemical leakage
4. Ability to pump slurries and corrosive chemicals

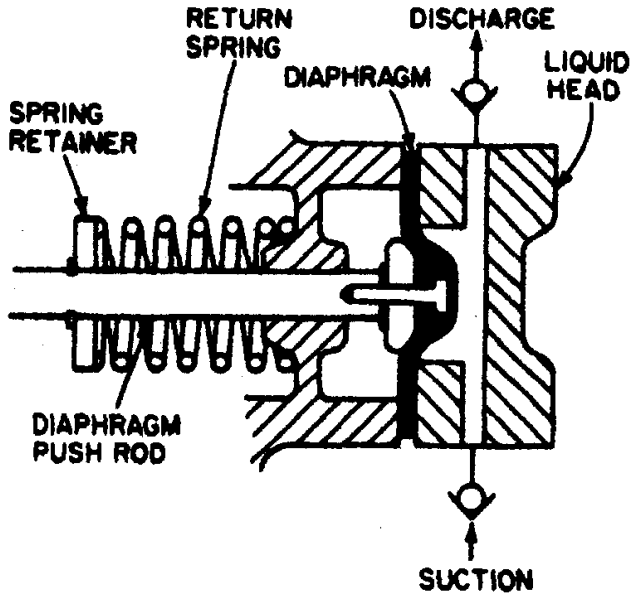


FIGURE 2 Mechanically actuated diaphragm pump

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Its disadvantages are

1. Discharge pressure limitation of 125 to 150 lb/in^2 (860 to 1030 kPa) gage
2. Accuracy in 5% range and as much as 10% zero shift with change from minimum to maximum discharge pressure
3. Capacity limit of 12 to 15 gph (0.045 to 0.057 m^3/h)

Hydraulically Actuated Diaphragm The hydraulically actuated diaphragm pump is a hybrid design that provides the principal advantages of the other types. A packed plunger is used to pulse hydraulic oil against the back side of the diaphragm. The reciprocating action thus imparted to the diaphragm causes it to pump in the normal manner without being subjected to high pressure differences. A flat diaphragm design is shown in Figure 3 and a tubular diaphragm design in Figure 4.

The advantages of the hydraulically actuated diaphragm pump are

1. Pressure capability to 5000 lb/in^3 (34.5 MPa) gage
2. Capacities to 20 gpm (4.5 m^3/h)
3. Minimum maintenance
4. Zero chemical leakage
5. Ability to pump slurries and corrosive chemicals
6. Accuracy around 1% over 10:1 range

Its disadvantages are

1. Subject to predictable zero shift of 3 to 5% per 1000 lb/in^2 (6.9 MPa) gage
2. Higher cost

Hydraulically Actuated Pistons This type of design is used when high accuracy at relatively low pressure is required. Figure 5 shows the complete pumping and metering unit, whereas Figure 6 shows a sectioned view of the metering unit. The pumping unit shown in Figure 6 can be a gear or vane type, driven by an electric motor. This is integral to a dual-measuring unit (Figure 5), mounted on top of the pumping unit. Alternatively, a centrifugal pump located remotely in the product storage tank can provide the pumping function. The pumping unit is run at a constant speed of about 850 rpm. A bypass valve in the pumping unit varies the output from 0 to 100% in response to a control valve located downstream of the meter.

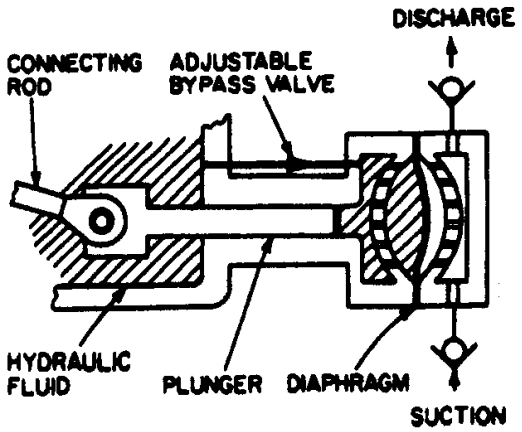


FIGURE 3 Hydraulically actuated diaphragm pump with flat, circular diaphragm

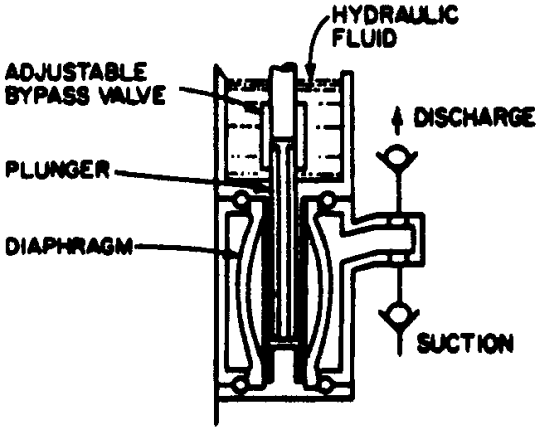


FIGURE 4 Hydraulically actuated diaphragm pump with tubular diaphragm

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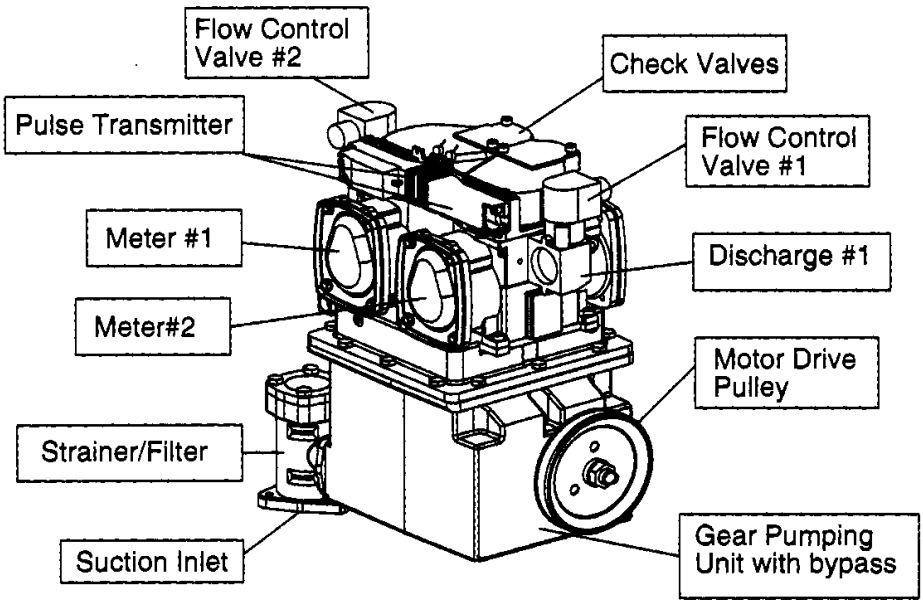


FIGURE 5 "Duplex" metering pump (Dresser-Wayne)

The metering unit shown in Figure 5 is a dual unit providing two separate measured outputs. Each meter unit has two pistons and three chambers. Scotch yokes that drive a rotary valve that ports the fluid in and out of the measuring chambers connect the pistons. A shaft encoder and calibration unit is integral to the meter to provide accurate adjustment of the pumped volume together with batch and running volume totals. The advantages are

1. High accuracy (0.25%) across complete flow range to 25 gpm (6 m^3/h)
2. Compact, totally integrated multifunction design.

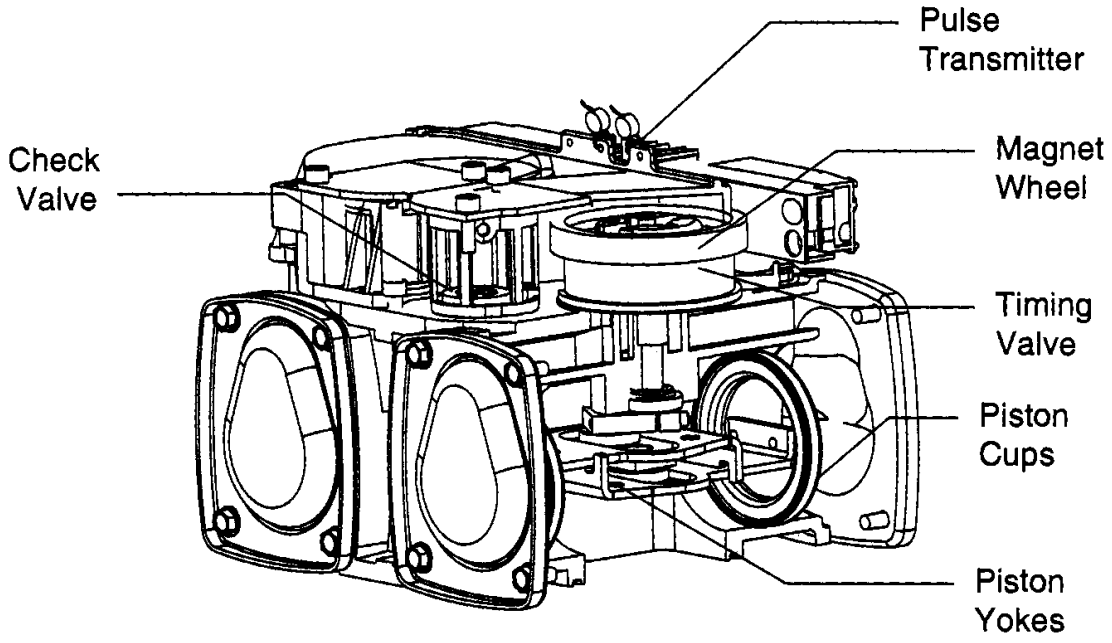


FIGURE 6 "Duplex" meter (Dresser-Wayne)

3. Minimum maintenance every 1.3–1.5 million gallons (5000–6000 m^3)
4. Low initial and ownership cost

Its disadvantages are

1. Pressure limitation of 100 lb/in^2 (700 kPa).
2. Maximum flow rate of approximately 25 gpm (6 m^3/h)
3. Only suitable for low viscosity noncorrosive products.

CAPACITY CONTROLS

There are five methods commonly used to adjust the capacity of metering pumps. The choice of which method to use is determined by the application for which the pump is intended.

Manually Adjustable While Stopped This control is generally found on packed plunger pumps of conventional design. Capacity changes are effected by moving the crankpin in or out of the crank arm while the pump is not in motion. This is the least expensive method and is used where frequent changes in pump displacement are not required.

Manually Adjustable While Running This feature is most frequently found on mechanically actuated diaphragm pumps, where it is accomplished by limiting the return stroke of the diaphragm with a micrometer screw. On packed plunger pumps, stroke adjustment while the pump is running is relatively complicated. Stroke length is set by some type of adjustable pivot, compound linkage, or tilting plate, which is manually positioned by turning a calibrated screw. On hydraulically actuated diaphragm pumps, relatively simple control is provided by manually adjustable valving that changes the amount of the intermediate liquid bypassed at each stroke.

Pneumatic Meeting pumps used in continuous processes must be controlled automatically. In pneumatic systems, the standard 3- to 15-lb/in^2 (21- to 103-kPa) gage air signal is utilized to actuate air cylinders of diaphragms directly connected to the stroke-adjusting mechanism.

Electric On electric control systems, stroke adjustment is through electric servos that actuate the mechanical stroke-adjusting mechanism. These accept standard electronic control signals.

Variable Speed This method of adjusting capacity is achieved by driving a reciprocating pump with a variable-speed prime mover. Because it is necessary to reduce stroke rate to reduce delivery, discharge pulses are widely spaced when the pump is turning slowly. Surge chambers or holdup tanks are used when this factor is objectionable.

In control situations involving pH and chlorinization, two variables exist at once; for example, flow rate and chemical demand. This is easily handled by a metering pump driven by a variable-speed prime mover. Flow rate can be adjusted by changing the speed of the pump, and chemical demand by changing its displacement.

CALIBRATION

Calibration, which is a fine adjustment of capacity, can be performed in the same way as the capacity controls previously discussed by using a fine stroke adjustment mechanism. It can better be achieved using electronics linked to the pumped volume display. A shaft encoder is connected to the meter that outputs pulses proportional to the volume pumped.

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The number of pulses is selected relative to the display resolution required. If $\frac{1}{100}$ th of a liter were required on a nominal cyclic volume of 1 liter, 105 pulses would be provided per revolution. The pulses are fed into counting and processing electronics. The electronics are put in calibrate mode and a fixed measure pumped into a calibrated vessel. The electronics counts the number of pulses received for the known volume pumped, compares that to the number of pulses it should have received, and calculates a calibration factor for that meter. This factor is stored in nonvolatile memory, and used by counter to "discard" extra pulses when the pump is in normal delivery mode. The 105 pulses per revolution is used to make sure that there are always extra pulses to discard allowing a ±5% manufacturing variation on the nominal cyclic volume. The calibrating electronics can also store batch and lifetime volumes pumped totals, and supply this information to a local or remote display.

SERVICE AND MAINTENANCE

Proportioning pumps utilizing manual capacity controls can be installed, serviced, and operated by plant personnel. Pneumatic capacity controls are more sophisticated, but after their construction and function are understood, maintenance and service should become routine. Electric capacity controls require a basic understanding of electric circuits for installation, operation, and routine service. Modular construction allows repair service by replacement of component groups, thus diminishing the task of trouble-shooting.

All proportioning pumps utilize suction and discharge check valves. These require regular maintenance and service because they are used frequently, encounter corrosion, and must be kept in good working order to ensure accurate pump delivery. Service periods are greatly dependent on liquids pumped, pump operating speed, and daily running time. Usual service periods run from 30 days to 6 months or longer. Check valves are designed to facilitate service with a minimum of downtime, thus allowing replacement of wearing parts at minimum expense. Good design allows this service to be accomplished without breaking pipe connections to the pump.

Packed plunger pumps require periodic adjustments of the packing takeup device to compensate for packing and plunger wear. Packing has to be replaced periodically, and regular lubrication of bearings and wear points is required.

The lubricating oil in speed-reduction gearing, either separate integral units or built in, must be changed at six-month intervals.

Diaphragm pumps usually require replacement of the diaphragm as part of routine service at six-month intervals. On hydraulically actuated diaphragm pumps, the hydraulic fluid must also be changed.

Pneumatic and electric controls generally present no special maintenance problems. The frequency of routine cleaning and lubrication is dependent on environmental conditions and should be consistent with general plant maintenance procedures.

INSTALLATION

Proper installation of metering pumps is very important if reliable pump operation is to be obtained.

NPSH must be kept as high as possible, and the manufacturer's recommendations as to pipe size and length, strainers, relief valves, and bypasses must be observed.

MATERIALS OF CONSTRUCTION

With the tremendous variety of liquid chemicals used in industry today, an all-inclusive guide to suitability of materials for pump construction is virtually impossible. For the great majority of common chemicals, pump manufacturers publish data on materials of

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construction. In general, packed plunger and hydraulically actuated diaphragm pumps are available as standard construction in mild steel, cast or ductile iron, stainless steel, and plastic. Mechanically actuated diaphragm pumps are usually available as standard construction in plastic and stainless steel. Almost any combination of materials can be furnished on special order.

Diaphragms are available as standard construction in Teflon, chemically-resistant elastomers, and stainless steel from various manufacturers.

See also Section 3.6, "Diaphragm Pumps," and Section 3.5, "Displacement Pump Flow Control."

SECTION 12.16
CRYOGENIC PUMPS FOR LIQUEFIED GAS SERVICE

JINKOOK LEE

Cryogenic fluids have normal boiling points below 100°F (or −73°C), and include liquid oxygen, hydrogen, nitrogen, helium, argon, methane, and ethane.

Since the start of the twentieth century, there has been increasing demand for high quality cryogenic pumps for space and missile technologies as well as commercial applications. It is always critical for cryogenic pump designers to consider the rate of thermal shrinkage of each pump component to be certain of sufficient clearances that allow the pump to run smoothly with high efficiency at very low temperatures.

LIQUEFIED HYDROCARBON

Since World War II, liquid rocket engine development has advanced significantly in military and commercial applications. On these engines, high-speed centrifugal, turbine-driven cryogenic pumps supply liquid oxygen as an oxidizer and various types of fuel such as liquid hydrogen, kerosene, H_2O_2 to be burned in the engine. The inducer suction specific speed N_{ss} for liquid hydrogen pumps is reported to run in excess of 35,000 (rpm, $gpm^{1/2}$, $ft^{-3/4}$) and as high as 70,000 (rpm, $gpm^{1/2}$, $ft^{-3/4}$) because this liquid is near its thermodynamic critical point. (See Section 12.18.2.)

After the oil shocks in the 1970s and the increasing worldwide demand for energy coupled with the international clean air policies set in the 1980s, gas is regarded as the key transition fuel of the twenty-first century because of its environmental benefits in reducing greenhouse gas emissions. The U.S. government's Energy Information Administration (EIA) predicts that world gas consumption will nearly double by 2020, with its share of the total energy supply increasing from 23% in 1999 to 28% in 2020 and that, by 2030, developing countries will account for more than 60% of the increase in world energy. The cryogenic fluids in this service are primarily liquefied natural gas (LNG), which is normally a mixture of more than 90% methane at −162°C or −260°F, ethylene, and ethane; along with the liquefied petroleum gases (LPG) propane and butane. Therefore, demand of LNG and LPG as clean energy sources will increase the demand for cryogenic pumps in the import, transportation, and export gas markets.

Over its 45-year history, the submerged motor-driven cryogenic pump has been used in the LNG business because cryogenic fluids are excellent dielectrics with low lubricity.

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A portion of the pumped fluid is guided through the motor gap between the rotor and stator to cool the motor and flows downward to lubricate the bearings. Motors are running at 2-pole and 4-pole speeds at 50 and 60 Hz depending on the application, such as pumps in a land-based receiving terminal or marine cargo pumps in LNG carrier tankers.

What Is Natural Gas? Natural gas is a flammable gaseous mixture made up essentially of hydrocarbons (chemical compounds that contain carbon and hydrogen), which is found along with coal and petroleum. Natural gas is a fossil fuel, which exists in a gaseous form in either sedimentary rock in the earth's crust or in layers of organic matter that has accumulated at the bottom of bodies of water from the Precambrian to the Conozoic Era.

Natural gas is obtained through the four stages of exploring, confirmation of its existence, drilling, and mining. Natural gas, depending on where it is obtained, can contain as much as 90% methane (CH_4) with small quantities of ethane (C_2H_6) and even smaller amounts of propane (C_3H_8), butane (C_4H_{10}), pentane (C_5H_{12}), and other alkines. Natural gas is extracted and refined into fuel that provides approximately 25% of the world's energy supply.

OVERVIEW OF LNG PRODUCTION AND SUPPLY CHAIN

The simplified LNG production process and supply chain can be summarized as follows:

Offshore Gas Production → Onshore Gas Processing → LNG Export Terminal → Shipping → LNG Receiving Terminal → End User

Gas Production Facilities Natural gas upon being extracted and refined is then liquefied. This processed natural gas is known as LNG. LNG takes up six hundredths of the volume of natural gas, which allows for efficient storage and long-distance transportation.

Liquefaction Methods

1. *Turbo Expander Cycle*

   When natural gas is compressed under high pressure and expanded using turbines, its temperature drops sharply and it enters a liquid state.

2. *Cascade Cycle*

   The cascade cycle uses various refrigerants to liquefy natural gas. First, propane is liquefied by using ethylene as a refrigerant. Then, natural gas is liquefied by using this liquefied propane as a refrigerant.

3. *Multicomponent Refrigerant Cycle*

   In this cycle, natural gas is condensed and expanded by using a mixture of nitrogen and hydrocarbons as a refrigerant until the temperature drops to the liquefaction point.

Liquefaction Facilities

STORAGE TANKS LNG storage tanks at the natural gas liquefaction facilities have the same structure as the ones at the receiving terminals. Storage tanks at the LNG fields are used for temporary storage before LNG is shipped to its destination. LNG storage tanks have double walls like a thermos. Their exterior is made of a poststressed concrete wall, which is designed to absorb external shocks and maintain the tank's cohesiveness. The interior wall is a sealed stainless steel membrane specially designed to store LNG. Between the interior and exterior walls, polyvinyl chloride (PVC) is inserted to insulate the tank and so maintain a low temperature, thereby preventing the LNG from reverting

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back to a gaseous state. The primary (in-tank) pumps in the storage tanks pump liquefied gas to recondensers, and the vaporized gas in the storage tank is sent by pipeline to the boil-off gas (BOG) compressors for reliquefaction.

LOADING SYSTEM After LNG is produced and stored in tanks at production sites, it travels to the final destinations by either moss or membrane-type LNG tanker vessels. The type of vessel is determined by the size of storage tank. Loading systems used to fill the tanker vessels include LNG pumps, pipelines, loading arms, and BOG facilities to handle the gas generated in the process of loading.

GAS RECEIVING TERMINAL FACILITIES The size of an LNG receiving terminal ranges from less than 1 million tons per annum (tpa) to over 10 million tpa of gas sent out. Typical end uses for the imported gas are as follows:

- Power generation for metropolitan areas
- Industrial/domestic fuel for customers in metropolitan areas
- Power generation for regional use
- Industrial/domestic fuel through pipe line for regional use
- Feedstock for manufacturer of other chemicals
- Alternate source of fuel for transportation (CNG/LNG vehicle)
- Others

A typical LNG receiving terminal simplified flow diagram is shown in Figure 1.

UNLOADING SYSTEM At the LNG unloading facilities, the LNG tanker is towed to the production site. Upon docking, the ship is moored to a jetty to minimize movements during the unloading process. Once securely moored, three LNG unloading arms are connected to one NG unloading arm, and LNG is transported to ground storage tanks via pipelines

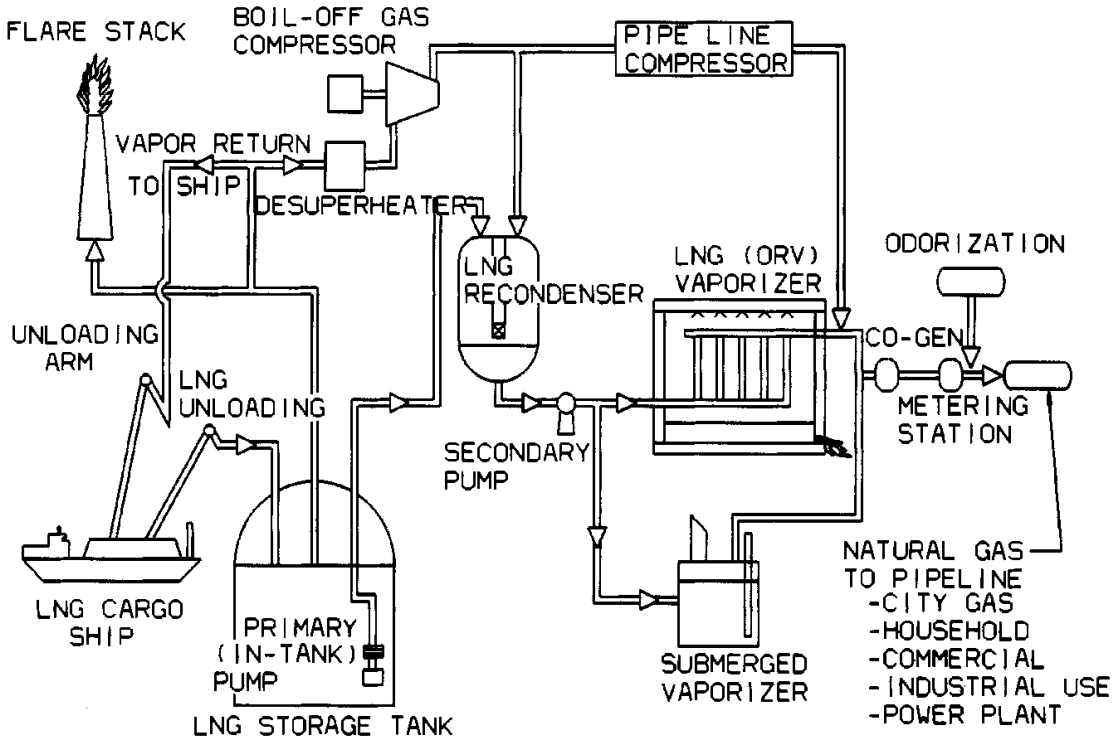


FIGURE 1 Typical LNG receiving terminal simplified flow diagram

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using cargo pumps on the tanker. The NG unloading arm is connected to LNG unloading arms to maintain the pressure inside the storage tank on the LNG vessel. The pressure inside the vessel drops when LNG is unloaded, but the NG arm maintains this pressure by supplying boil-off gas generated at the production site. One LNG tanker vessel can unload its capacity of between 125,000 m^3 (approximately 57,000 tons) and 135,000 m^3 (about 62,000 tons) of LNG within an average of 12 h.

The LNG receiving platform and a typical marine cargo pump are shown in Figures 2 and 3. The LNG unloading system also includes facilities for disconnecting the ship-to-shore interface at the end of the unloading operation, and for undocking the ship prior to its return voyage.

LNG UNLOADING ARMS As the typical unloading time for a full-size tanker (125,000 to 140,000 m^3) is 12 to 15 h, this requires peak unloading rates of about 12,000 m^3/h. This can be accomplished by two 16-in unloading arms, each capable of transferring 6000 m^3/h. The transfer pipe portion of the unloading arm is made of stainless steel, capable of withstanding cryogenic temperatures below 162°C. Typically, the shipboard cargo pumps provide a head of about 100 to 140 m. The jetty unloading platforms, which typically have four 16-in LNG arms and one 16-in vapor return arm, are shown in Figures 4 and 5.

STORAGE SYSTEM The storage system provides a buffer between the LNG discharged from the ships and the vaporizers that regasify the LNG. The system consists of one or more specially designed tanks. Ships carrying the LNG arrive at the terminal at specified intervals. The minimum required storage capacity is the volume of LNG discharged from the largest ship expected at the terminal. While the typical capacity of an LNG tanker is around 140,000 m^3, it is anticipated to grow to 200,000 m^3 in the near future. In practice, the installed storage is larger than this minimum. The extra storage provides a cushion to account for scheduled and unscheduled delays in ship arrival. A typical land-based LNG receiving tank is shown in Figures 6 and 7.

Most large LNG tanks in service today, including all the large LNG tanks in the United States, are of the single-containment, or double-containment design and built in accordance with American Petroleum Institute (API) Code 620, Appendix Q, BS7777, and Recommended Practice In-Ground Storage Tank (RPIS).[4,5] Typical design temperature inside

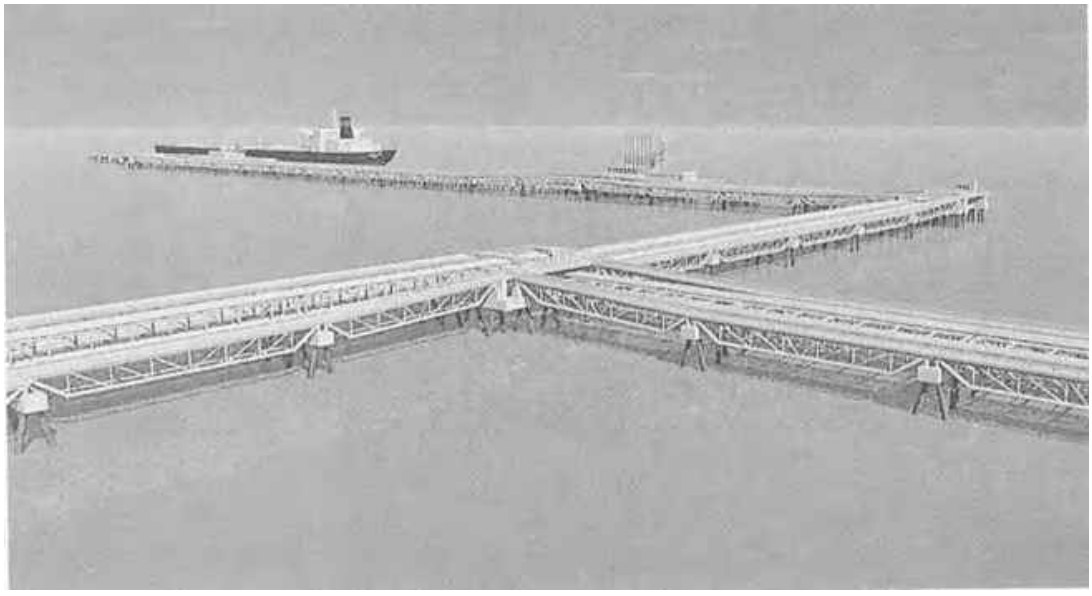


FIGURE 2 Typical LNG receiving platform

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FIGURE 3 Typical shipboard cryogenic pump with flow rates to 1500 m^3/h (Courtesy of Carter Cryogenics)

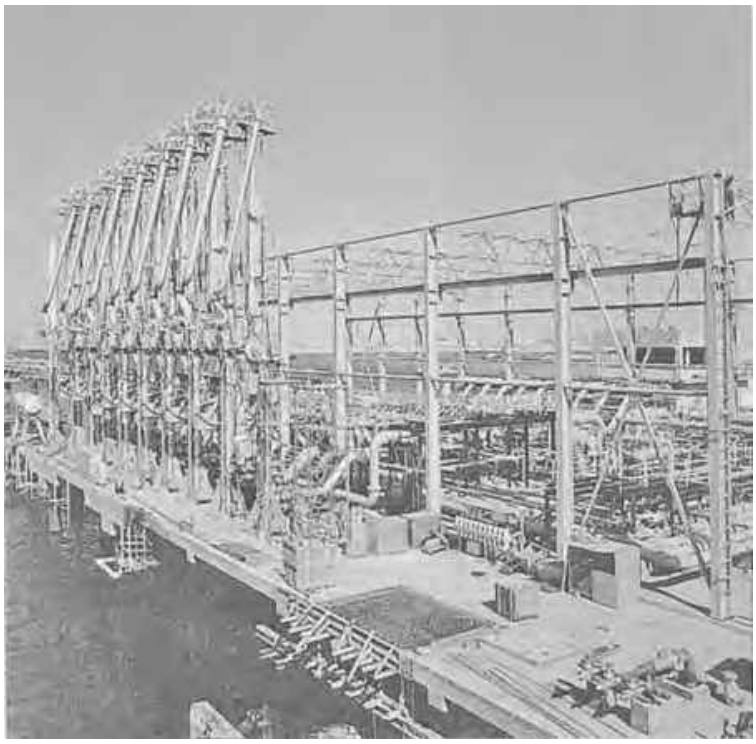


FIGURE 4 Typical receiving jetty arms

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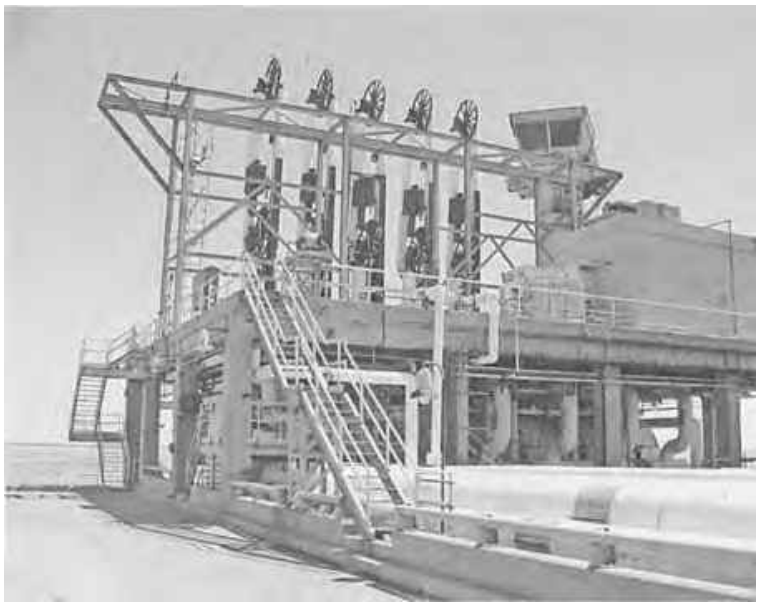


FIGURE 5 Typical receiving jetty arms

of the tank is –170°C; the range of design pressure is from –5 to 450 mbarg; and operating pressure is from 50 to 350 mbarg.

BOG COMPRESSOR BOG compressors literally compress the boil-off gas from LNG storage tanks and pipelines. LNG boils at an ultra-low temperature of –162°C, and can easily boil due to heat transmitted from the outside of tanks and pipelines. BOG compressors reliquefy this vaporized gas by raising pressure. Ethane and propane have the relatively higher boiling points of –88.6°C and –42°C, respectively. Therefore, BOG from LNG is almost entirely methane.

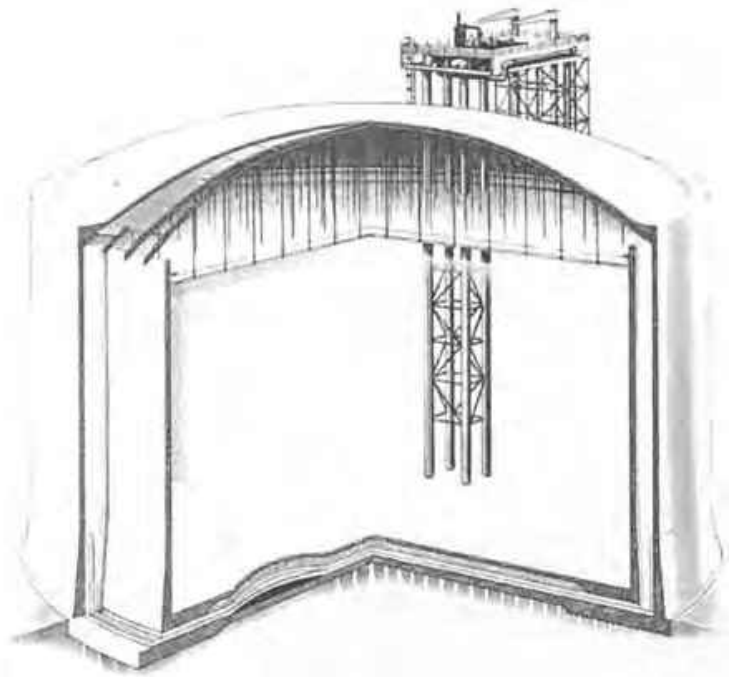


FIGURE 6 Typical land based LNG receiving tank

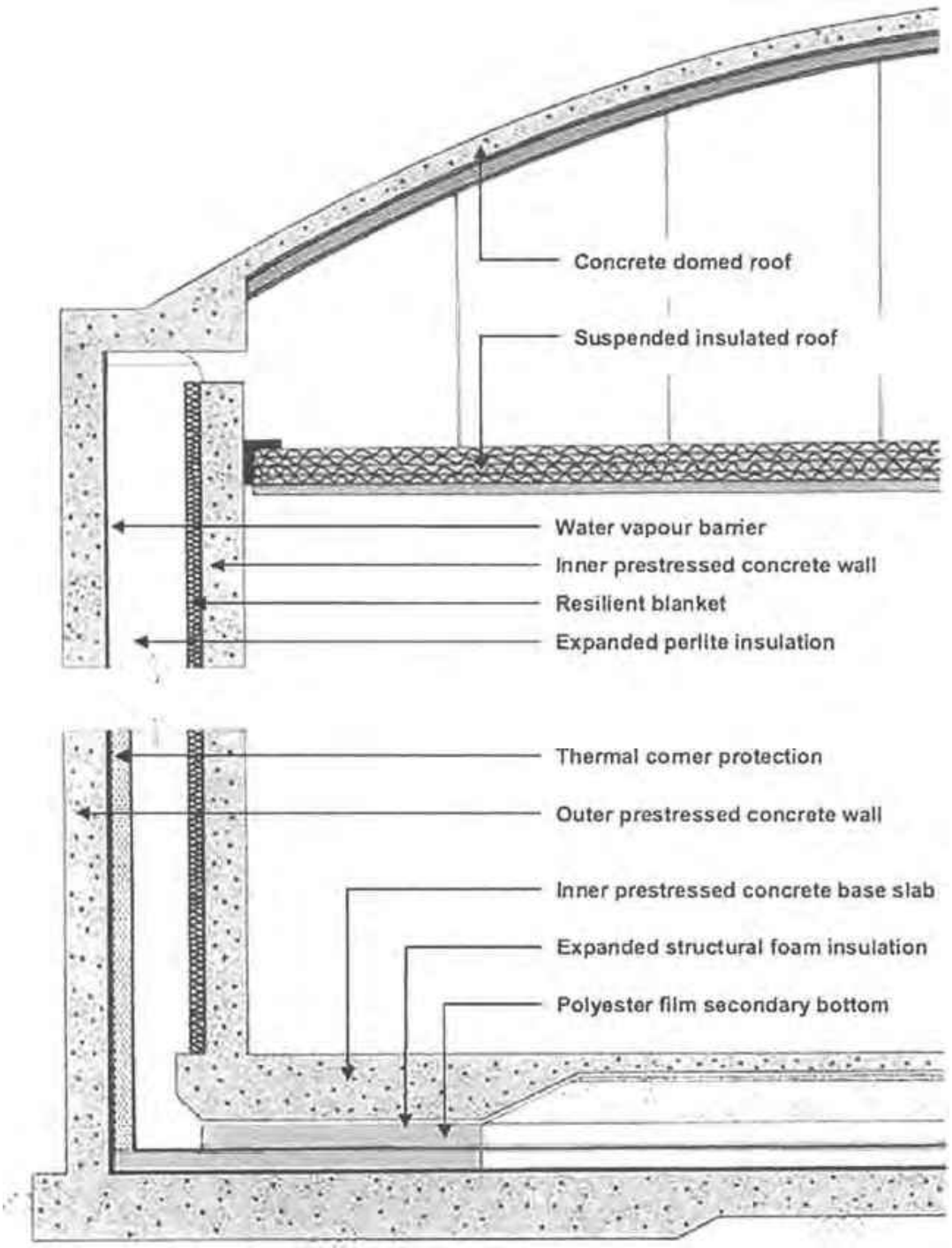


FIGURE 7 Typical all concrete primary containment

RECONDENSERS A recondenser liquefies BOG compressed by the BOG compressor by blending it with LNG. LNG, compressed to a higher pressure by primary (in-tank) pumps in the storage tank, and the BOG from the BOG compressor each separately flow into the condenser and are mixed. The LNG with its low temperature (–150°C to –160°C) and BOG with a relatively high temperature (5°C to 35°C) mix and condense to LNG (–130°C to –150°C) through a heat exchanger reaction in the recondenser, and is then transported to the secondary (ex-tank or sendout) pumps. LNG boils at a higher temperature than –162°C when the pressure is high.

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LNG PUMPING SYSTEM

In-Tank Pump LNG tanks operate at very low pressure, just slightly above atmospheric pressure. The gas from the terminal needs to be delivered at an elevated pressure. Since pumping the liquid is easier and less expensive than compressing gas, the LNG is pumped to the required sendout pressure prior to its regasification. The pump-out system may consist of one or two stages of pumps, depending on the required battery-limit pressure of the gas and on the configuration of the BOG handling system. Pumping of cryogenic liquids, especially at the high rates required in LNG facilities, is specialized technology. In most cases, in-tank pumps are driven by 3-phase, 2-pole, or 4-pole motors at speeds corresponding to 50 and 60 Hz. In general, land-based in-tank pumps operate at 50 Hz. However, with a variable frequency drive (VFD), they can operate at any speed. In modern terminals the first-stage pumps almost always are installed inside the storage tanks, and are referred to as in-tank pumps as shown in Figures 8, 9, 10, and 11.

Ex-Tank Pump A secondary pump (ex-tank pump or sendout pump) pumps compressed LNG from low-pressure primary LNG pumps and recondensers to high pressure. An extremely high pressure level is required to deliver natural gas via a long pipeline network across the nation to the end users. As it is easier to increase the pressure of a liquid than a gas, the liquefied form of LNG is compressed normally up to 80 bars by secondary

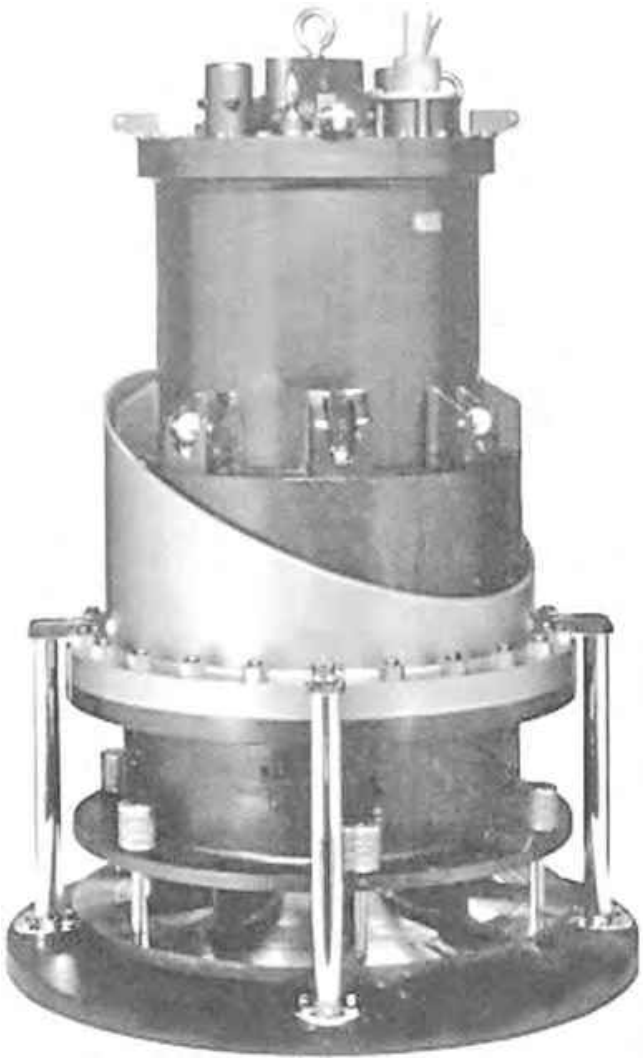


FIGURE 8 Typical removable in-tank cryogenic pump with flow rates to 2200 m^3/h (Courtesy of Carter Cryogenics)

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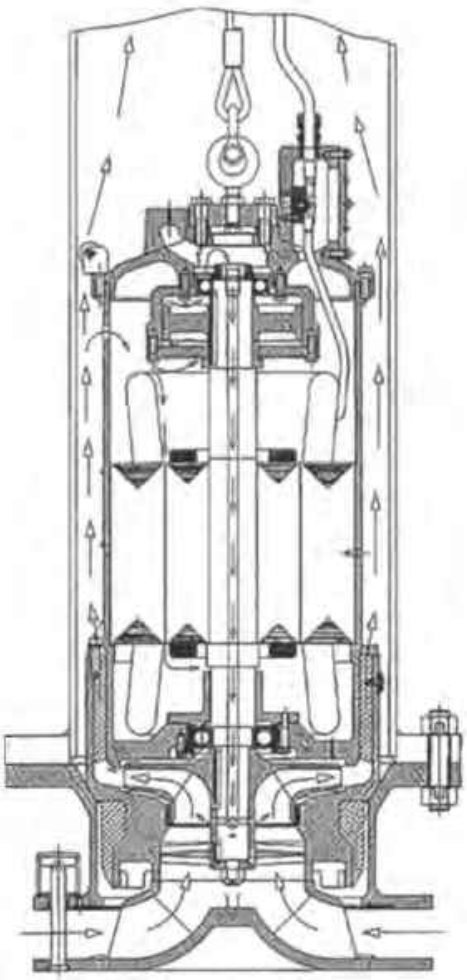


FIGURE 9 Cross-sectional view of typical single-stage cryogenic in-tank pump showing flow passages through the motor (Courtesy of Carter Cryogenics)

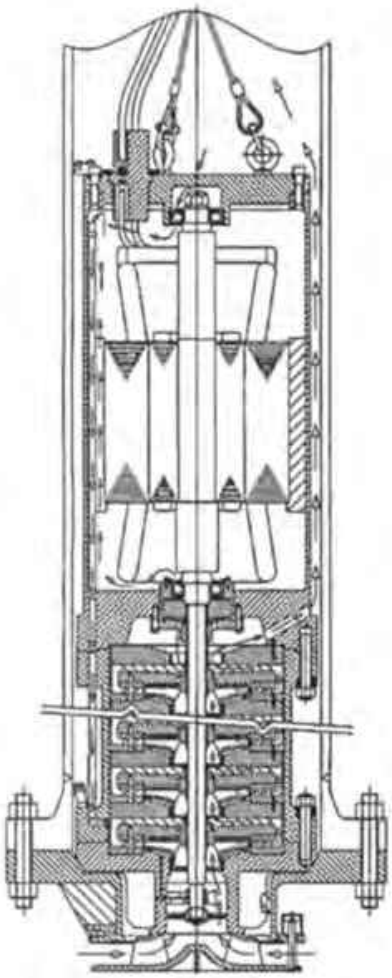


FIGURE 10 Cross-sectional view of typical in-tank multistage cryogenic pump (Courtesy of Carter Cryogenics)

pumps. In addition, fewer calories are consumed in vaporizing gas when the LNG pressure is high. Secondary pumps supply highly compressed LNG to high-pressure vaporization facilities. The number of secondary pumps is determined by the amounts to be delivered through the network pipelines.

These second-stage pumps, when required, are located outside the tanks, in the process area. Second-stage pumps discharge at a pressure sufficiently high to satisfy the battery limit pressure at the terminal fence. Cross-sectional views of single and multistage suction vessel-mounted ex-tank submerged motor pumps are shown in Figures 12, 13, 14, and 15.

Vaporizer A seawater open rack vaporizer supplies vaporized LNG from secondary pumps to the pipeline. There are two types of vaporizers:

- Open rack vaporizer (ORV) that uses the temperature of seawater which is higher than that of LNG
- Submerged vaporizer (SMV) that uses the combustion of gas

The ORV pumps and pours seawater to vaporize LNG, while moving through the thin-layered heat exchanger from bottom to top. The heat exchanger of an SMV is submerged in a water tank, and a burner installed in the tank burns oxygen and natural gas to heat water and vaporize LNG.

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FIGURE 11 Typical removable in-tank multistage cryogenic pump (Courtesy of Carter Cryogenics)

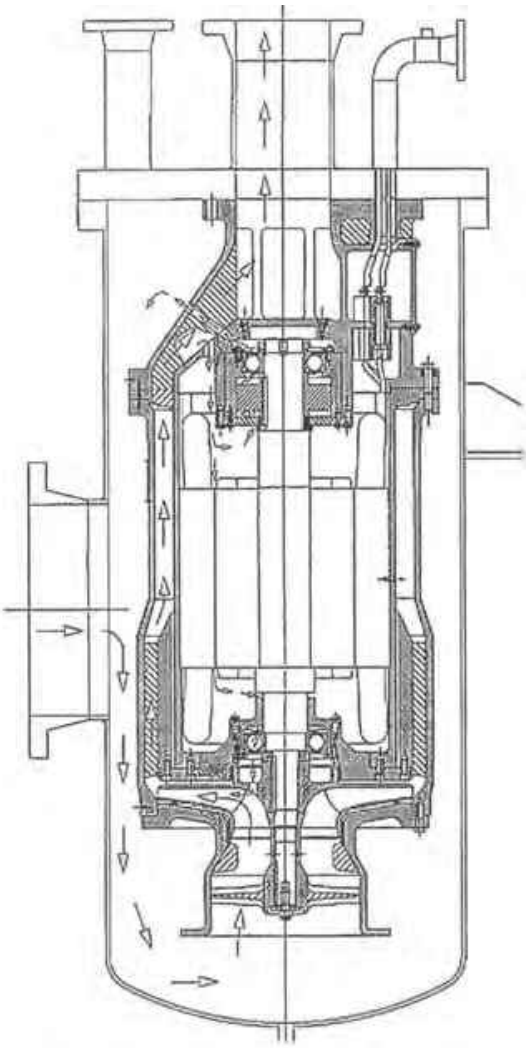


FIGURE 12 Typical suction vessel-mounted ex-tank submerged motor pump—single stage (Courtesy of Cater Cryogenics)

HISTORY AND DEVELOPMENT OF CRYOGENIC PUMPS

It was reported that the first commercial production of LNG was at a liquefaction plant in Cleveland, Ohio, USA, in early 1940s. A LNG carrier, the "Methane Pioneer," made the first successful transport of a commercial cargo of LNG from Lake Charles, Louisiana across the Atlantic Ocean to England in 1959.

This voyage carried an almost token quantity of 5000 m^3 of LNG. This vessel was fitted with deep-well pumps, which were found to be unreliable. An investigation was undertaken to find possible types of pumps that could be used in the transfer of LNG in a safe, simple, and reliable manner. This program identified one of the most suitable types to be a fully submerged electric motor pump based on submerged motor fuel pumps used in the aerospace industry. After long and exhaustive testing, the very first application was realized in late 1961 when a submerged motor pump was installed in the LNG carrier Beauvais followed by the first commercial vessels, the "Methane Progress" and the "Methane Princess" in 1964. These units were found to operate successfully. Similar pumps were later installed in other LNG carriers.

Figure 16 shows the history and development of cryogenic pumps over the last 40 years.

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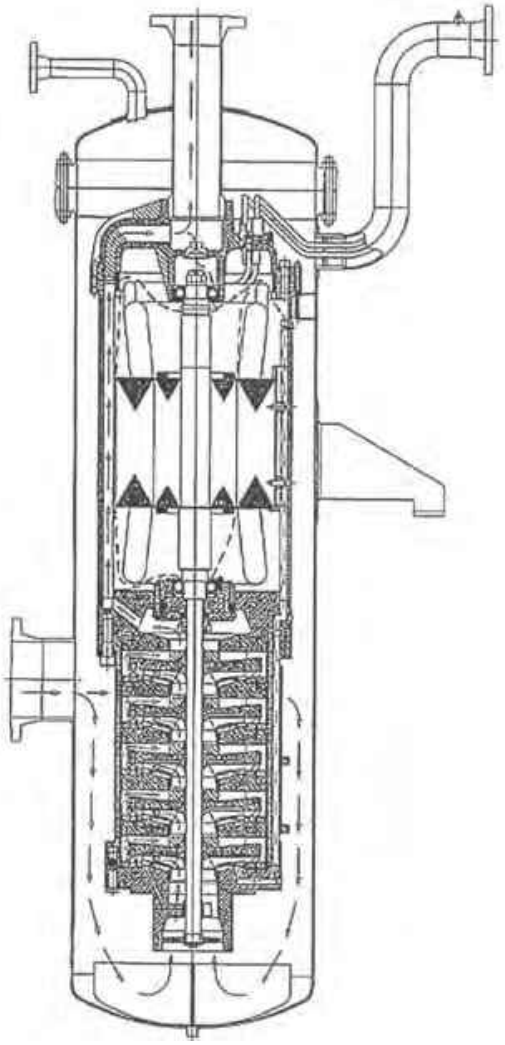


FIGURE 13 Typical suction vessel-mounted ex-tank submerged motor pump—multistage (Courtesy of Carter Cryogenics)

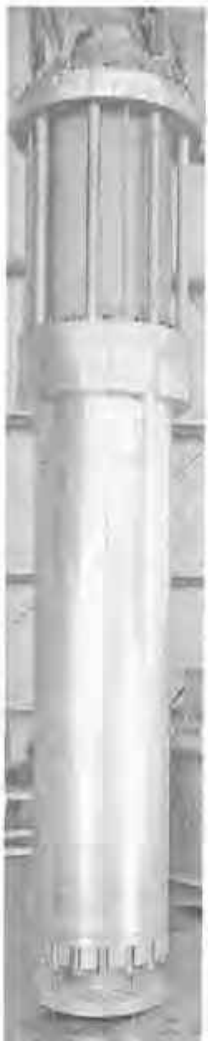


FIGURE 14 Typical multistage ex-tank submerged motor pump—with a maximum of 25 stages and over 8 m tall, 1000 m^3/h, and 5000 HP (Courtesy of Carter Cryogenics)

DESIGN CONSIDERATIONS FOR CRYOGENIC PUMPS

Cryogenic pumps such as LNG pumps incorporate many special features that make them different from other pumps in industrial and process plant services. Some of these features and requirements are as follows:

- Submerged electric motors
- Suction-vessel mounting with suction and discharge connections
- Retractable in-tank mounting for installation within tanks
- Special materials of construction
- Testing at very low temperature and at the high-power levels expected in service
- Cooldown and start-up considerations
- Warm up for teardown inspection
- Handling considerations for pump size and weight
- Required operation at low NPSH and often two-phase flow conditions which need high performance inducers and high-efficiency impellers

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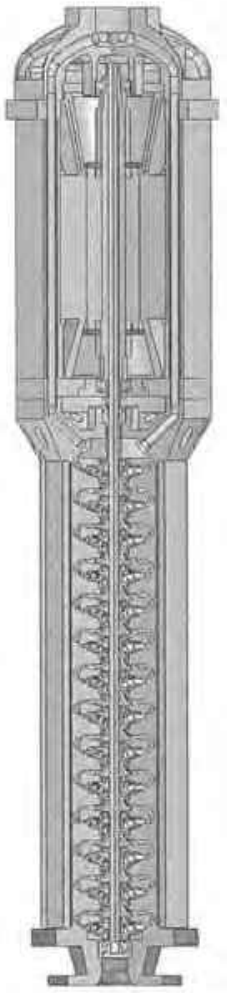


FIGURE 15 Typical multistage ex-tank submerged motor pump (Courtesy of Carter Cryogenics)

Removable In-tank Pump System When the removable pump system is installed in accordance with U.S. National Electric Code, it is suitable for use in Class I Division 1 locations, with leak-free pump maintenance. Figure 17 depicts the components usually supplied as part of the system.

The key to a safe, removable pump system is the foot valve. A patented foot valve ensures that no flammable gas or liquid enters the column during pump installation or removal. The foot valve is attached using a standard pipe flange at the bottom of the pump column. During pump installation, the pump weight opens a spring-loaded valve poppet, admitting liquefied gas into the column. During operation, pump discharge is isolated from pump suction by a conical seal between the pump housing and the foot valve. The tapered seal-type system is self-centering and requires no additional seals, such as O-rings. When the pump is removed, the foot valve poppet closes, isolating the liquefied gas from the column. The pump can be safely removed from the tank with no emission to the atmosphere. For applications where no leakage of liquefied gas into the column can be tolerated after a pump is removed, a redundant purged seal foot valve can be used. During installation and removal, pump facilities should support and protect electrical power cables. These consist of an electrical-power assembly and two antirotation stainless steel wire ropes (support cable and lift cable). The support cable ensures that electrical cables are never exposed to any load. The lift cable facilitates pump removal and is attached to the electrical-cable assembly by means of a one-touch clamp that requires no special tools. This clamp precludes the possibility of anything falling into the pump column during pump installation or removal. Spreader bars center the removal system and prevent excessive movement.

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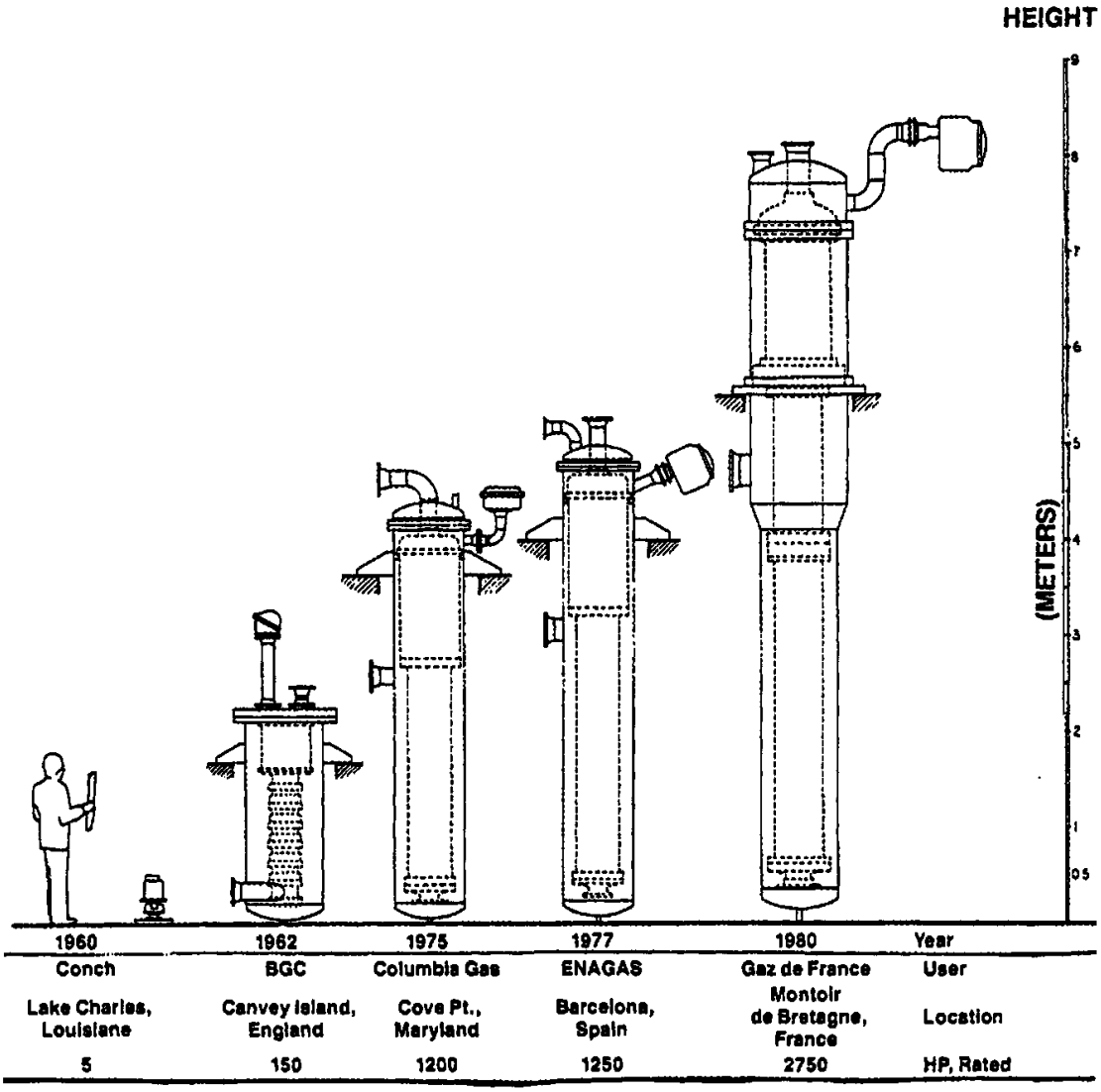


FIGURE 16 Typical history and progress of cryogenic pump development (Courtesy of Carter Cryogenics)

Materials of Construction Head plates, lift-shaft covers, and terminal headers are designed and fabricated in compliance with ASME Pressure Vessel Code. Impeller and pump casings are fabricated from ASTM A356-T6 aluminum alloy. Impellers are hard-coat anodized to provide an erosion-resistant integral hard surface. The strength, light weight, and heat-transfer properties of this material are well suited to cryogenic applications. Inducers are typically machined from AMS 7075-T6 solid aluminum billets. Five-axis machining provides better control of surface finishing and contour, as required in advanced designs to meet low NPSH and possibly higher-gas-volume-fraction two-phase-flow pump inlet conditions. This also enhances strength. After machining, the surface is hard-coat anodized, for cavitation resistance and durability. Some sample cryogenic pump materials are listed in Table 1.

Submerged Electric Motors A submerged cryogenic motor must be highly efficient, because the pump operates in –162°C cryogenic liquid and any heat generated due to the inefficiency of the motor will be the direct source of BOG. Therefore, a small amount of pumped fluid is actually routed through the motor to wet the rotor and stator as a coolant and as well as to serve as a lubricant for the bearings. In normal operation, the motor temperature rarely rises more than 5°C. Like many other hydrocarbon liquids, LNG is not corrosive or electrically conductive. Submerging the motor in LNG totally isolates the motor

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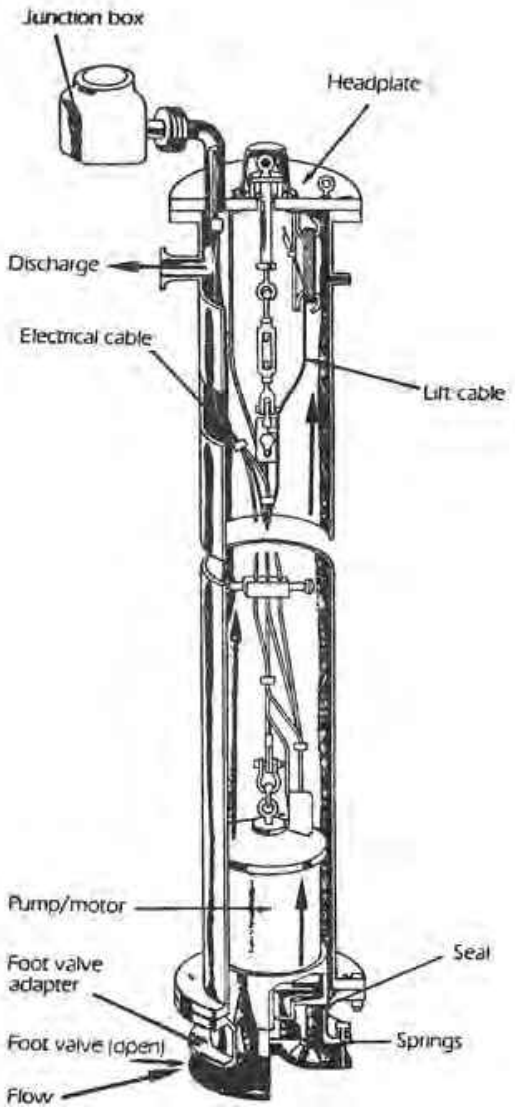


FIGURE 17 Typical removable in-tank pump system (Courtesy of Carter Cryogenics)

TABLE 1 Sample materials for cryogenic pump components (Courtesy of Carter Cryogenics)

Part	Equivalent Material	Specification	ASTM
Head plate	Stainless steel	304	A240/182/312
Lift-shaft cover	Stainless steel	304	A240/182/312
Terminal header	Stainless steel	304	A276
Impeller	Aluminum, A356-T6	AMS 4218	B26
Impeller wear ring	Hard-coat anodize	MIL-A-8625	Type III
Pump casing/diffuser	Aluminum, A356-T6	AMS 4218	B26
Casing wear ring	Bronze	SAE 660	B148
Motor housing	Aluminum, A356-T6	AMS 4038	B209-65
Inducer	Aluminum, 7075-T6	AMS 4038	B209-65
Suction casing	Aluminum, A356-T6	AMS 4218	B26
Bearings	Stainless steel	AISI 440C	A276
Rotor shaft	Stainless steel	17-4PH	A461

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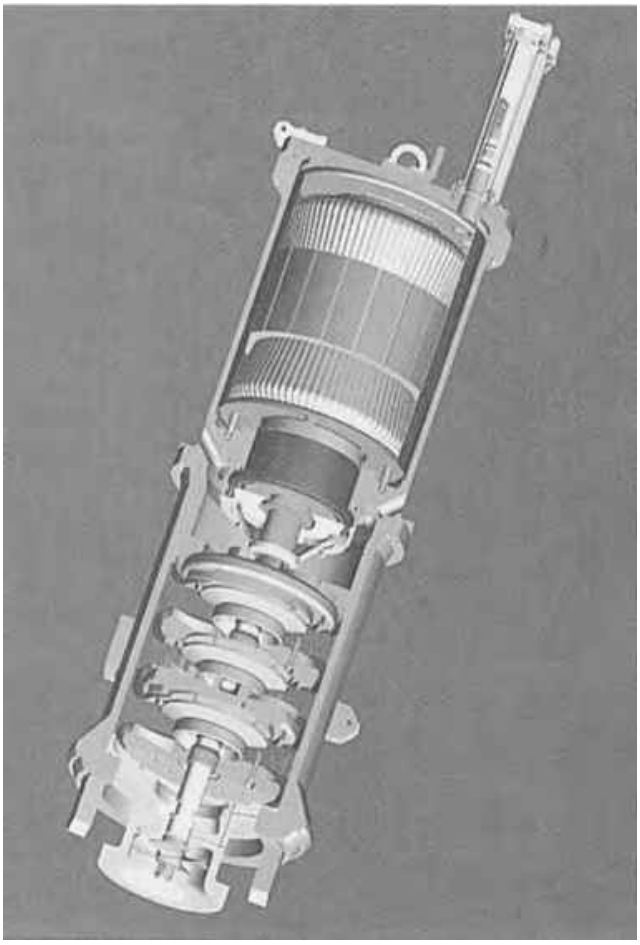


FIGURE 18 Typical submerged pump/motor assembly (Courtesy of Carter Cryogenics)

and its connections from any air or oxygen, which eliminates the possible danger of fire or explosion. Figures 18, 19, and 20 show the typical cryogenic submerged pump/motor assembly, motor rotor, and motor performance, respectively.[2,3]

Motor Stator Design for Stability and Moisture Resistance For the stability of motors above 200 kW, a form-wound stator system has been developed. Smaller motors are often random or "mush" wound for economy. All stators are vacuum pressure impregnated (VPI) to provide a moisture-impervious assembly. This keeps the insulation free of moisture even if the pump is disassembled when cold.

Rotor Design for Strength and Stability Rotors are made of standard electrical silicon-steel laminations with cast or fabricated aluminum bars and end rings. The rotor is mounted to the shaft with a controlled-interface fit. The assembly is quenched in liquid nitrogen (LN_2) for stability and then dynamically balanced in two planes to ISO standard 1940/1 Grade 2.5 to minimize vibration.

Hydraulic Design In recent years, hydraulic components such as the inducer, interstage guide vanes (IGV), impeller, and crossover diffusers are designed in most cases by commercially available computer programs and in-house-developed programs. The ability of computational fluid dynamics (CFD) to make reliable predictions has advanced rapidly and can enable avoidance of expensive design and testing practices. As discussed in Section 2.1.2, commercial CFD codes are available to use as an aid in the design process.

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FIGURE 19 Typical rotor of cryogenic pump motor (Courtesy of Carter Cryogenics)

Design procedures and integrated elements of systems are as follows:

- Kickoff meeting to review design specification
- Preliminary design review
- Detail (or critical) design review
- Sophisticated analysis tools are applied to both design and design verification such as
  - Rapid CFD for hydraulic design
  - Finite element analysis (FEA) for stress analysis
  - Rotor dynamics for vibration analysis

This system has specialized five-axis machining software, design for assembly (DFA) and design for manufacture (DFM) tools, and a smooth transfer of data to computer aided design (CAD) packages.

Inducer Designs for NPSH Performance Most cryogenic pumps require inducers to meet low pump inlet NPSH requirements over the wide flow ranges required. There are several types of inducers designed to meet the requirements at the pump inlet, such as fan inducers, Archimedes-type or spiral inducers, and the high performance inducer (HPI) illustrated in Figure 21a (side view) and Figure 21b (frontal view).

The suction specific speed N_{ss} of these inducers often reaches 80,000 to 100,000 (rpm, $gpm^{1/2}$, $ft^{-3/4}$). Even these values have been exceeded in tests of the HPI, which pumped down to 0.1 m of NPSH at the intake of an LPG pump at a flow rate of 1000 m^3/h.[1]

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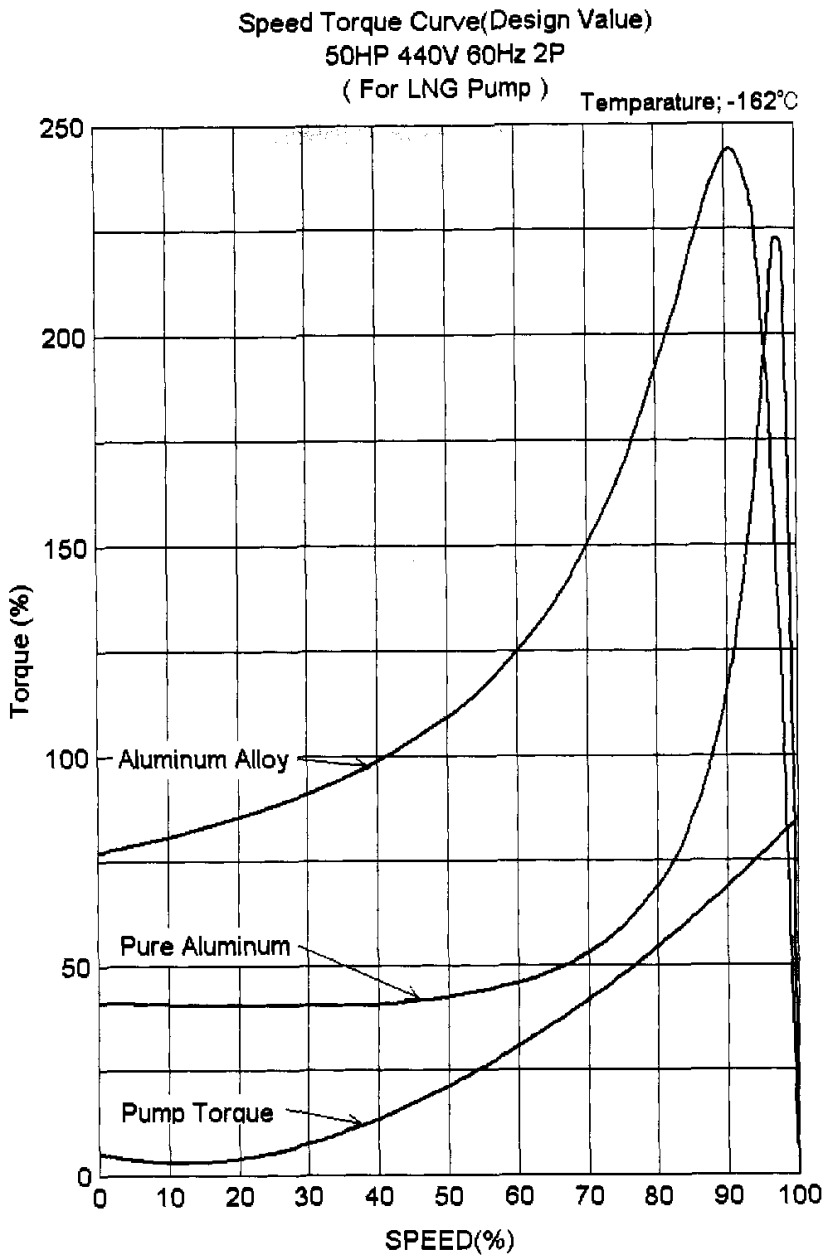


FIGURE 20 Typical cryogenic pump submerged motor performance (Courtesy of Carter Cryogenics)

The HPI type of inducer has also demonstrated the capability of ingesting two-phase flow having a gas- (or vapor-) to-liquid volume ratio (V/L) of up to 0.5 in normal cases and sometimes up to 1.0 in extreme aerospace applications, particularly in modern large commercial jet engine main fuel pumping systems (see Section 12.18.1).

Impeller Design Most cryogenic pump impellers are single-suction, radial-flow, and closed-type impellers with one-piece construction and serrated wear rings. Single-stage pumps can be fitted with back wear rings to assist in thrust balancing. A typical 3-D solid model of a radial-flow impeller is shown in Figure 22a (without the front shroud in place) and Figure 22b (with the shroud in place).

A typical performance map with different sizes and sets of impellers is shown in Figure 23. At a given specification requirement, head rise versus operating flow rate at a chosen pump speed can be easily found from performance charts of this kind.

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(a)

FIGURE 21A Typical High Performance Inducer (HPI) (Courtesy of Carter Cryogenics)



(b)

FIGURE 21B Typical high performance inducer (HPI) (Courtesy of Carter Cryogenics)

Diffuser Design Multistage pumps are equipped with several types of diffusers. When combined with radial-flow impellers and radial diffusers, this design does not generate the unbalanced hydraulic forces that are common in axial diffusers. The inherent stability prevents premature wear of guide bushings and wear rings. The radial diffuser stacks together with other diffusers in a way that ensures concentricity with the rotating elements. The design permits closer running clearances to enhance efficiency. A radial diffuser is shown in Figure 24.

The continuous crossover diffuser, shown in Figure 25, has been incorporated in the design of a recent multistage pump. CFD tools were utilized in the design process, which enabled designers to maximize overall pump performance and reduce the total length of the shaft, resulting in minimized total height and weight of the pump.



(a)

FIGURE 22A Typical 3-D solid model of radial-flow impeller (Courtesy of Carter Cryogenics)



(b)

FIGURE 22B Typical 3-D solid model of radial-flow impeller (Courtesy of Carter Cryogenics)

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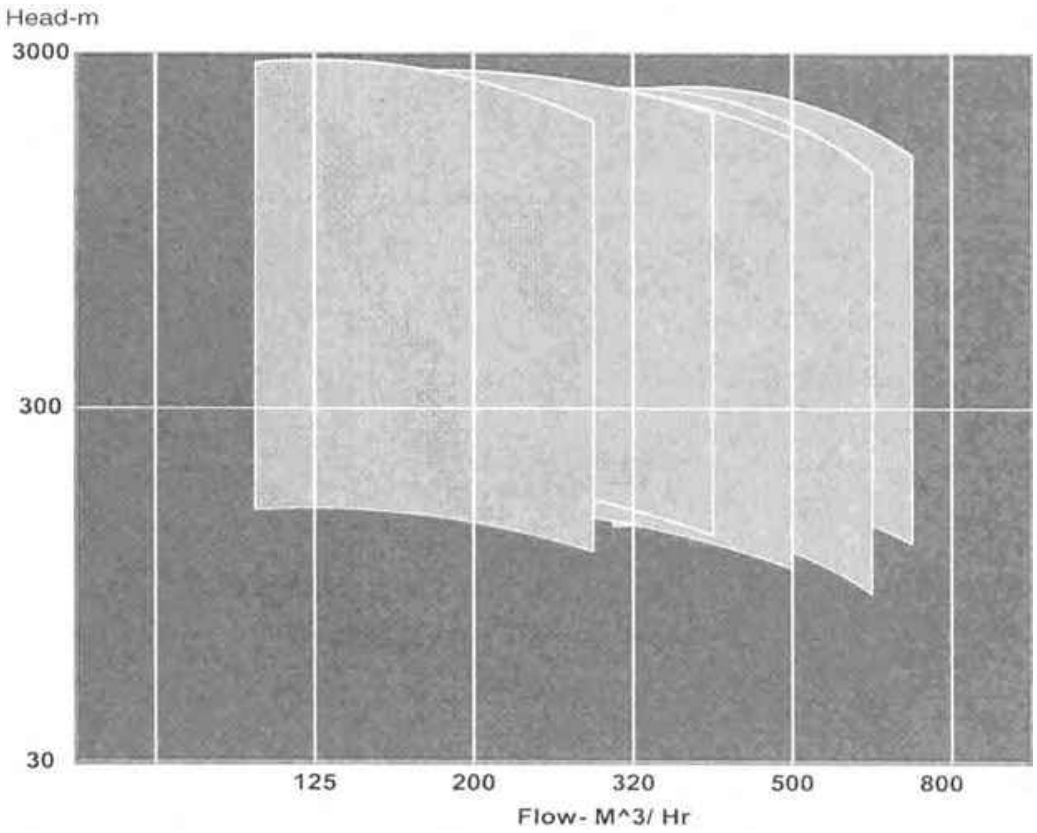


FIGURE 23 Typical envelope of pump performances obtained with different sets of impellers and pump speeds (Courtesy of Carter Cryogenics)

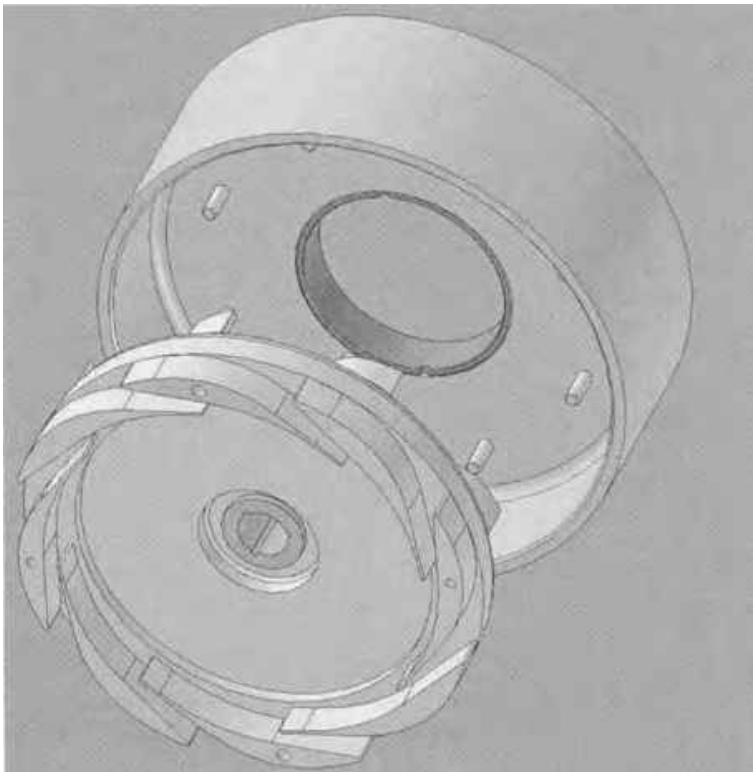


FIGURE 24 Typical radial-flow diffuser (Courtesy of Carter Cryogenics)

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FIGURE 25 Typical continuous crossover multistage diffuser (Courtesy of Carter Cryogenics)

Rolling-Element Bearings Cryogenic fluids have poor lubricating qualities, precluding the use of commercially available bearings. This has led to the development of durable rolling-element bearings that can accept thrust in either direction to handle spurious load reversals.

Pumps coupled to variable frequency drives (VFDs) can be supplied with ceramic ball bearings to prevent arcing due to eddy currents. The absence of heat in the bearings, combined with a thrust-equalizing balance drum, gives the bearings exceptional durability.

Thrust-Equalizing Balance Drum The thrust-equalizing balance drum balances thrust across the entire operating flow range, removing all axial bearing loads. The system is modular, and can be located either upstream or down stream of the motor. It keeps the motor at discharge pressure, preventing vapor from forming in the motor housing. This suppresses any hotspots and the possibility of liquid boiling, which could compromise cooling.

Terminal Header Submerged motor cryogenic pumps are provided with a terminal-header assembly to supply electrical power safely to the pump. The terminal-header assembly employs two hermetic seals, in series. Each seal is rated for maximum system pressure and minimum process temperature.

Extensive lab testing assures seal-assembly reliability and integrity. Testing includes:

- Seismic shock
- 24-h thermal exposure
- Cycling test: 100 thermal and pressure cycles over the full operating range
- 200-bar hydrostatic proof-pressure test with zero leakage tolerance
- Dielectric tests

Each terminal header is subject to acceptance tests for electrical-insulation resistance and proof pressure. Each seal is tested for integrity with a mass spectrometer to ensure that leakage is less than 1×10^{-6} sccm (standard cubic centimeters per minute). End-user connections are provided for purging, pressure sensing, or gas detection. Figure 26 shows the location of terminal headers for the pot-mounted and removable in-tank-mounted pumps.

Flexible Cryogenic Electrical Power Cables Electrical-power cables are typically configured as flexible nonasbestos insulated cables designed for use in cryogenic fluids at temperatures as low as –196°C. Each power cable is protected with a stainless steel braid sheath over its entire length and has both low (800 V) and medium voltage (6000 V) ratings. Figures 27 and 28 show samples of cryogenic cable.

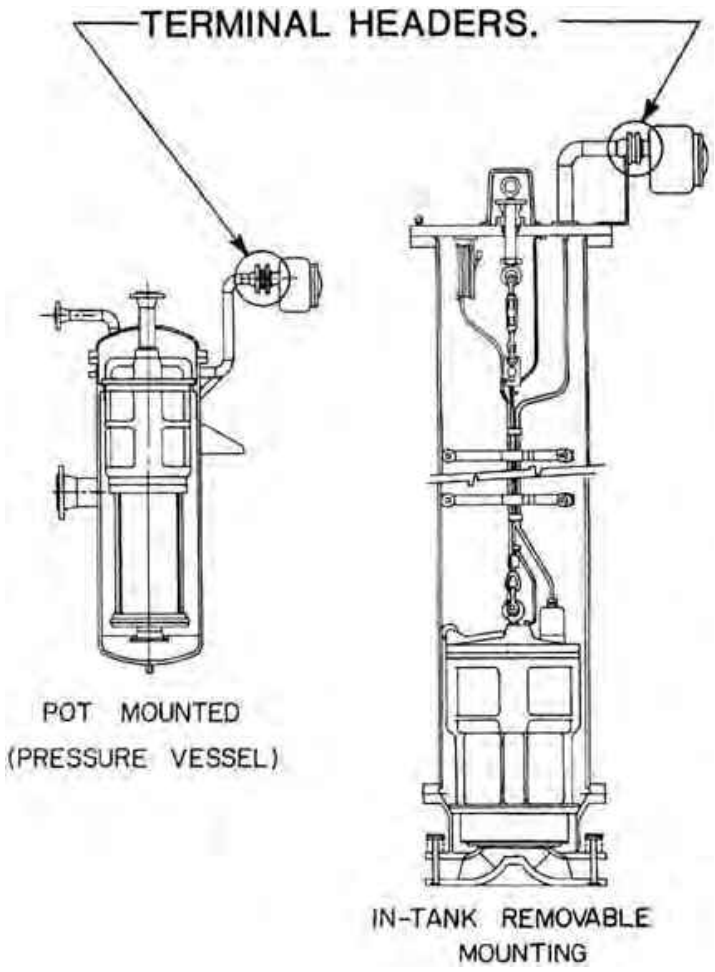


FIGURE 26 Terminal-header applications (Courtesy of Carter Cryogenics)

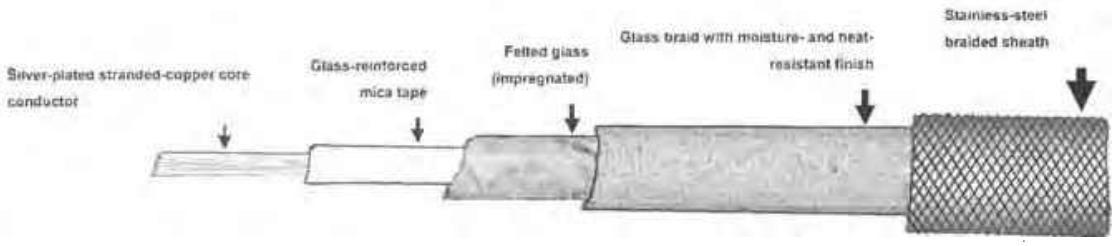


FIGURE 27 Typical layers of the cryogenic cable (Courtesy of Carter Cryogenics)



FIGURE 28 Typical stranded copper and the first three layers of insulation (Courtesy of Carter Cryogenics)

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CRYOGENIC TESTING

The manufacturer should provide full-scale testing of submerged motor cryogenic pumps. A typical facility for this purpose is illustrated in Figures 29 through 32. It has storage capacity for LNG, LPG, LN_2, and other fluids. During performance testing, all measurements are monitored by a high-speed data-acquisition system. This system graphically displays both raw and reduced data against predicted performance in real time. All instrumentation is tractable to recognized standards, with an overall measurement accuracy of ± 0.25%.

Figure 29 shows an ex-tank pump being lowered to the underground tank by crane for test. The cryogenic test loop during test is shown in Figure 30. After testing, the pump is moved by crane to the disassembly area for visual inspection as depicted in Figure 31. As seen in Figure 32, upon arriving at the disassembly area, the pump is heated up to the room temperature for teardown and visual inspection of any signs of abnormality.

Because of thermal contraction of the pump material in severe cryogenic conditions, the pump tested should be carefully examined after disassembly for any sign of rubbing by rotating parts such as inducer and impeller against nonrotating parts such as the inducer shroud or crossover discharge vane housings. Clearances of any mating parts at room temperature are not the same as those in the test tank at –162°C, but an unnecessarily larger gap will contribute to poor pump performance. Therefore, in the design stage, special care must be taken to specify the correct clearances for the mating parts of the pumps, which have many variations in size, flow rate, speed, and head-rise requirements.



FIGURE 29 A secondary cryogenic pump being lowered to the underground tank for testing (Courtesy of Carter Cryogenics)

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FIGURE 30 Cryogenic test loop during cryogenic pump test (Courtesy of Carter Cryogenics)

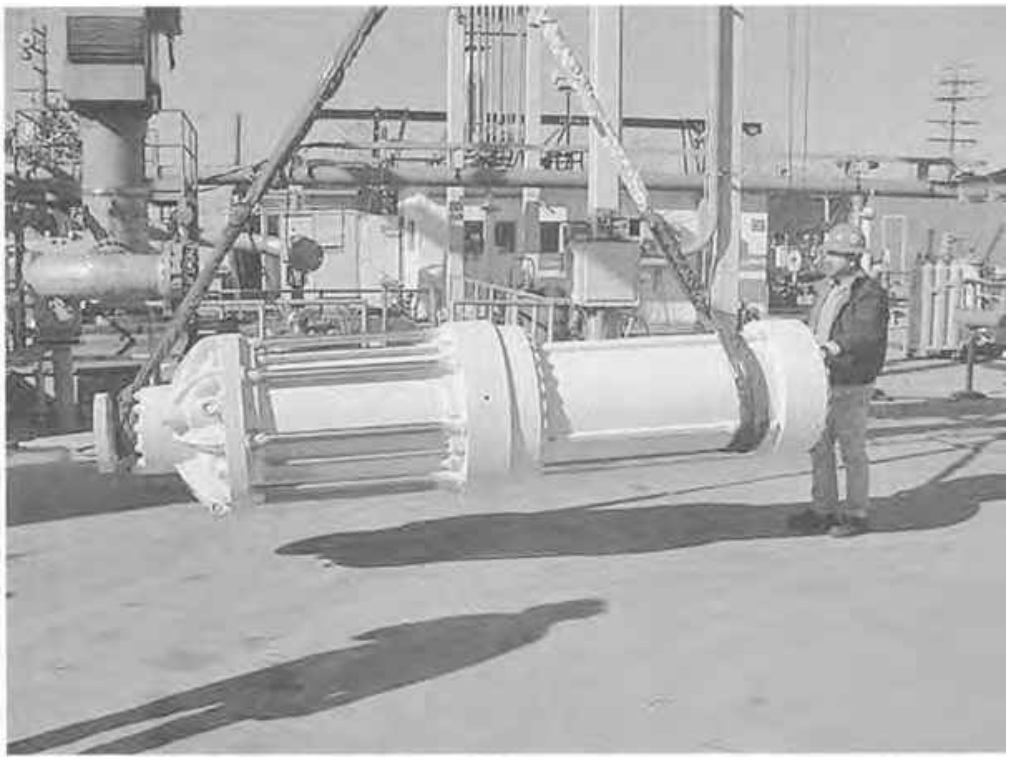


FIGURE 31 After testing, the cryogenic pump is transferred to undergo teardown inspection (Courtesy of Carter Cryogenics)

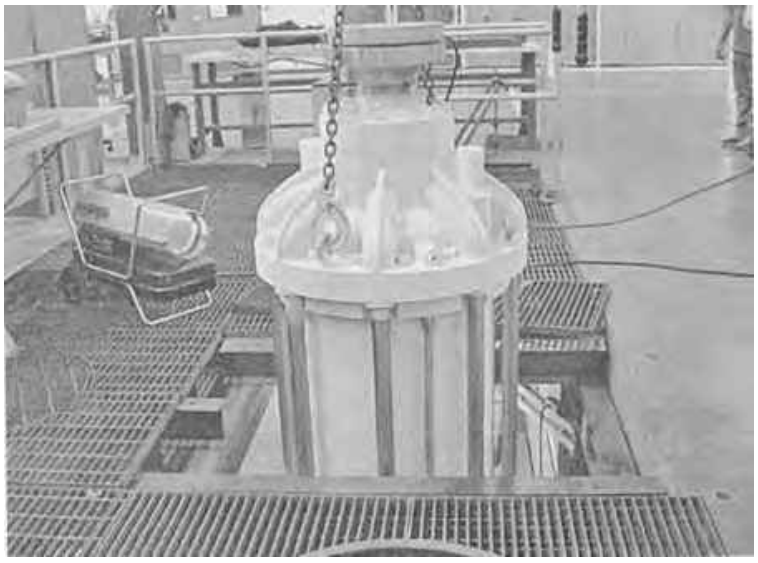


FIGURE 32 After testing, the pump is warmed to room temperature for teardown inspection (Courtesy of Carter Cryogenics)

OTHER TECHNICAL CONSIDERATIONS

Methodology for Seismic Qualification of Pumps

REQUIREMENTS

1. Equipment should keep its physical integrity, and earthquakes should not produce any unrecoverable deformation that could prevent proper operation. This requirement extends to the mounting brackets and the anchorage or fastenings to the support structure.
2. Earthquakes should not produce a loss of the pressure boundary that could lead to spillage of the LNG.

SEISMIC ASSESSMENT The seismic response of each pump should be combined with the effects of normal operating loads and compared with the allowable loads, stresses, and deformations.

FREQUENCY ANALYSIS Since the seismic load depends on natural frequencies, the first step of the methodology is to compute natural frequencies. In all cases, fluid-added masses should be taken into account. (See Section 2.1.4.)

SEISMIC RESPONSE Once the natural frequencies and mode shapes are known, the seismic response of the pumps should be computed. Three response spectra (two horizontal and one vertical) should be introduced for each analysis.

The response spectrum method gives the expected maximum values of the response during the earthquakes. Results include maximum section forces and moments, maximum deflections, maximum values of reaction forces, and maximum forces at connections or kinematical constraints.

The response spectrum method does not take into account nozzle loads. The effects of nozzle loads are added to the results of the response spectrum analyses by means of conventional static analysis.

In the case of the in-tank pumps, the response spectrum method does not consider the hydrodynamic forces on the column derived from the seismically induced movement of the whole LNG tank. These forces should be computed from maximum hydrodynamic pressures

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within the tank and should be introduced as static forces over the outer surface of the column.

RESPONSE TO NORMAL OPERATING LOADS The effect of normal operating loads should be added to the seismic response in order to assess the structural integrity and operability of the equipment. For computing the response to normal operating loads, the same finite element methods already used for frequency and response spectrum analysis should be employed. The response should be obtained by conventional static analyses.

Methodology With respect to the seismic qualification of submerged motor cryogenic pumps it should be noted that these pumps are mechanical devices that are inherently rugged from a seismic point of view. Operational loads are commonly more severe than seismic loads and, as a consequence, it is very rare that a well-designed pump will suffer from seismic damage if it is correctly attached to the supporting structure. Documented seismic experience of actual pumps in real earthquakes shows that only in a very few cases pumps were taken out of operation by the earthquake; and, in these cases, the failure was associated with the failure of the supporting pad or a rupture in the anchorage pipes.

With this background in mind, it seems reasonable to focus the qualification effort "outside" the pump-motor assembly, looking at the path of the inertial loads from the pump-motor assembly all the way to the anchorage to the supporting structure.

Inside the pump-motor assembly it should be sufficient to look at the deformation of the shaft due to earthquake inertial loads and at the additional forces on the bearings caused by the quake. However, in all probability, the results of the analysis will show that shaft deformation and bearing forces caused by the specified earthquakes are negligible.

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SECTION 12.17
PORTABLE TRANSFER OF HAZARDOUS LIQUIDS

P. A. NOLTE

Transfer systems for hazardous chemicals must not only take into account the best means to transport the chemical based on the chemicals fluid characteristics, but must also pay particular attention to both safety and environmental issues. The chemical can be something as common as gasoline, or a very poisonous restricted-use pesticide such as Paraquat. When the requirement of portability is added to the chemical handling system, design parameters must now take into account variables such as power source, environmental conditions, and chemical restraints. Several power sources have been incorporated through out the years for the portable transferring of hazardous chemicals. One of the earliest is simple hand power. Others consist of gasoline powered, ground-driven, compressed air, and battery operated pumping systems.

Major markets that require the portable transfer of chemicals include petroleum, agricultural, construction, pest control, and lawn care. Systems must be adaptable to changing environmental regulations, chemical container designs, and application requirements.

AGRICULTURAL REQUIREMENTS

The need for portable chemical transfer systems in the agricultural market arises from farmers working fields with large equipment, usually several miles away from local chemical dealerships and storage facilities. It is usually more economical to bring the chemical to the piece of application equipment while it is in the field, rather than transporting the application equipment back and forth from the field. Chemical is delivered to the field in several different ways, depending on the volume required. Tanks can vary in size from 120 gallon (450 liter) herbicide tanks that can be transported by pickup trucks, to large 1000 gallon (3800 liter) tanks that are delivered by trailers or large custom trucks. With the growth of the custom applicator business, the sophisticated chemical transfer rigs have

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focused on speed of transfer, flexibility to supply the chemical required, and the ability to cover large areas. Some large aerial applicator rigs have the ability to allow spray helicopters to land directly on the rig, thus allowing quick, portable, and convenient transfer of spray solution from tanks into the helicopter.

The majority of hazardous chemical transferring in the agricultural market deals with the usage of either fertilizers or pesticides. A pesticide is a group of chemicals that consist of insecticides, herbicides, or fungicides. Insecticides focus on the control of insects, herbicides are chemicals used to control of grasses and weeds, and fungicides are chemicals used to control crop damaging bacteria and/or fungus. During the last 20 years, the usage of chemicals in the agricultural market has been greatly affected by growing environmental concerns and worker safety. Environmental concerns have caused chemical formulations to use less aromatic solvents. As an alternative, many chemicals are now viscous water-based suspensions that generate pumping challenges. Chemical systems have also seen new regulations that control items such as chemical spillage from fittings, tank sizes, and design and testing parameters. This, in turn, has affected the needs and requirements of the chemical handling systems.

Fertilizers present a transferring challenge due to the large volumes that are used, and the corrosive nature of most solutions. Two types of power modes are commonly used in the market for the transferring of fertilizers: gasoline-powered centrifugal pumps and ground-driven positive displacement pumps. Gasoline-powered centrifugal pumps used are usually lightweight aluminum or plastic units with 2 in (50 mm) ports. The pumps are coupled to a 3 to 5 hp (2 to 4 kW) gasoline engine and are capable of transferring rates up to 150 gpm (568 l/min). These high flowrate gasoline powered pumps are used to transfer fertilizer and water from large 1000 to 2000 gallon (3785 to 7570 liter) transport tanks into the application equipment's onboard storage tanks, 300 to 500 gallons (1135 to 1890 liters) in size. Ground driven pumps are mounted on the implement equipment such as planters and cultivators. The pumps are used to transfer and meter the fertilizer through a network of tubing that runs to the backside of cutting blades. This allows the knifing of the fertilizer into the ground. Ground driven pumps are popular for this type of application because the volume dispensed is directly proportional to the ground speed of the implement. Popular pumps that have been adapted for ground driven applications are positive displacement piston, diaphragm or multi-tube peristaltic pumps.

Pesticide transfer is now rigorously regulated because of environmental concerns and an increased emphasis on decreasing worker exposure to the hazardous formulation of a large percentage of these chemicals. The Environmental Protection Agency (EPA) has guided the agricultural chemical industry away from using 2.5 gallon (9.5 liter) throwaway containers to larger, returnable, reusable, chemical-specific containers. These "Mini Bulk" tanks range in size from 15 to 200 gallons (50 to 750 liters), depending on the chemical usage rate. This change in the way farm chemicals are packaged to the end user has generated new requirements for portable pumps. Current chemical delivery systems have evolved around the usage of 12-volt power systems. Due to regulator restrictions that forbid the cross contamination of agricultural chemicals, the delivery systems are usually designed for a particular chemical or container. An adequate pumping rate for most 12-volt portable system is a rate of 5 to 10 gpm (18 to 38 l/min). Agricultural chemicals come in a wide range of viscosities, varying from 1 to 500 centipoise. Some are true liquids, whereas most are a suspension or mix. Typically, positive displacement pumps of the diaphragm, piston, or gear type are used for portable chemical transferring. Gear pumps are usual limited to the low viscosity chemicals with no suspended solids. Piston and diaphragm pumps are best suited to handle the higher viscous chemicals, many of which have suspended solids. Chemicals with solids usually will require tank circulation, and this must be designed into the transfer system.

Figure 1 shows a pumping system on a 120 gallon (450 liter) tank that consists of a pump, 12-volt motor, a digital meter system, and no-drip hose coupling. The pump is permanently mounted to the shipping container at the time of tank manufacture. The motor, meter, and hose are removable items that are normally supplied by the chemical dealer. This allows the drive unit and metering device to be transferred from tank to tank, after a tank is emptied. The pump is a 6-chambered plastic diaphragm pump and is mounted to the tank by means of a bolting flange. The mounting flange also contains a built-in vent to

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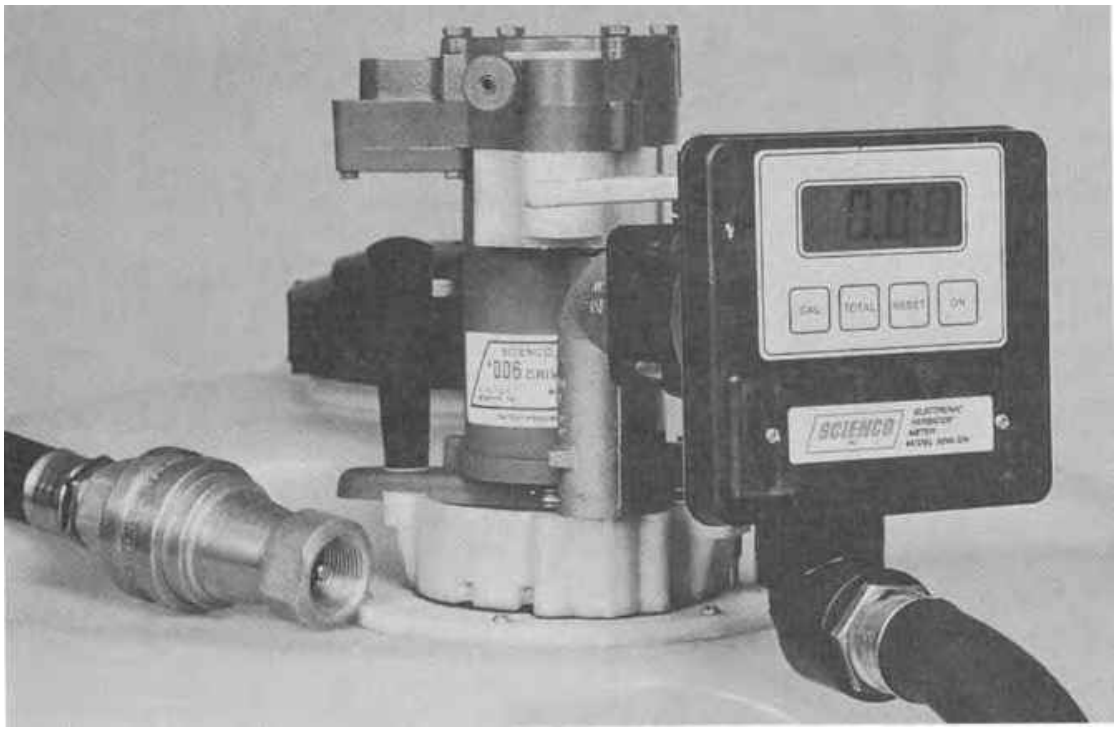


FIGURE 1 Portable 12-volt chemical pump (Flowserve Corporation)

allow air in during fluid pumping. An internal bypass valve in the pump is connected to the inside of the chemical tank, allowing chemical circulation within the tank during bypassing. This pump also allows manual opening of the bypass valve to allow drainage of chemical from the meter and hose before removal. When the pump is mounted to the tank during shipping, the complete system must meet and pass specific Department of Transportation (DOT) tests. These tests are specified in the United States by DOT's HM181 specifications. Depending on the size of the container and the hazardousness of the chemical, these tests can require drop tests of several feet at 0°F (minus 17.8°C), vibration, stacking, and pressure tests.

Figure 2 shows a typical 120 gallon (450 liter) chemical "Mini-Bulk" tank with the pump connected to the tank by a coupler (not permanently attached). The coupler system provides a means for attaching and detaching the pump and the tank, a means for venting, and a dip tube assembly allowing suction from the bottom of the tank. The pump assembly is coupled through the tank coupling for chemical dispensing and is then removed for transporting. The pump shown is a stainless steel, 12-volt diaphragm pump, with a stainless steel flow meter. The chemical dealer fills the mini-bulk tank with chemical. The grower purchases the chemical, uses the pump to dispense chemical into an application rig, and then returns the tank empty to the dealer.

PETROLEUM AND CONSTRUCTION REQUIREMENTS

One of the original areas requiring the use of portable pumps is the transfer of gasoline and diesel fuel. The need to refuel and maintain equipment in remote locations has generated a large need for portable transfer equipment. Road and building construction, mining, and farming all involve heavy equipment that must be refueled and maintained on a regular schedule. One of the earliest and simplest methods of portable transferring was with hand pumps. Hand pumps are positive displacement pumps of piston, vane, or diaphragm-type design. Of course, the power supply is very limited with hand pumps, and they are typically used only on thin liquids and small jobs. On large pieces of equipment,

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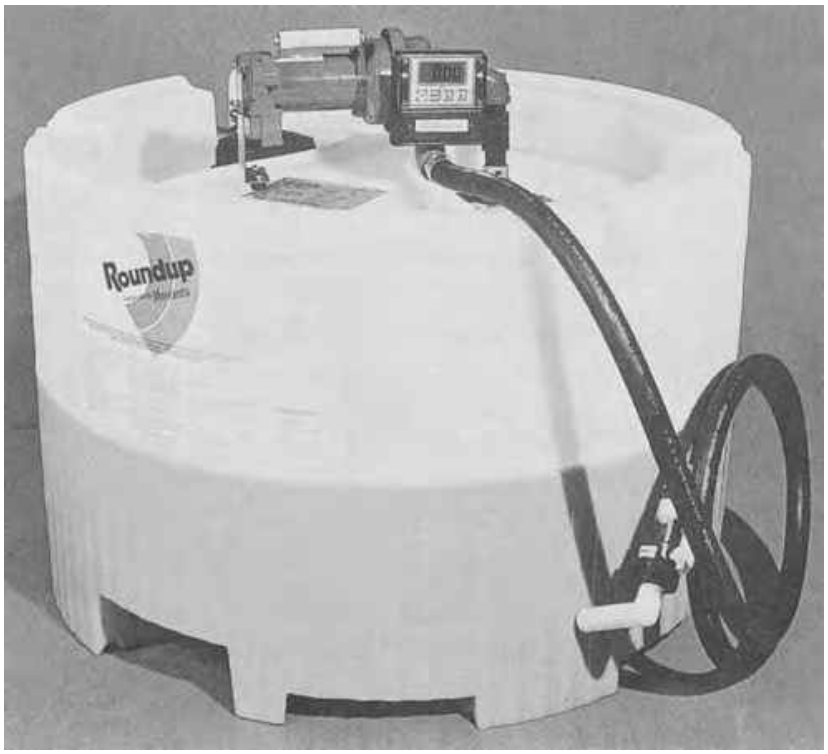


FIGURE 2 120 gallon (454 l) "Mini-Bulk" Container and pump (Flowserve Corporation)

the need is to transfer large volumes of fluids, such as water, antifreeze, diesel, gear lube, grease, and transmission fluid.

Figure 3 shows a portable maintenance vehicle with air powered piston pumps. In the construction industry, compressed air is a common power source for transferring product. A typical truck consists of a gasoline-powered air compressor, several tanks of petroleum products plus water and antifreeze. The size and type of pump will normally depend on the size of the equipment requiring servicing and the fluid being pumped. For thin fluids, such as diesel oil or gasoline, high volume, low pressure ratio pumps (approaching 1:1) will be used. Occasionally, on units that have the option of a power take-off (PTO), a high volume centrifugal-type pump will be used. For the oils and lubricants, higher pressure ratio pumps (3:1 to 5:1) are required because of higher transfer pressure requirements. To transfer grease, high pressure ratio pumps—to 20:1—are required. The pumps are normally coupled with a hose reel, 50ft (15 m) of hose, and a control nozzle.

LAWN AND PEST CONTROL

Lawn and pest control companies require a portable means of transferring chemicals in order to provide treatments. Liquids are used to treat yard infestations; in addition, the majority of homes and buildings are treated for termites during construction. It is common for lawn care companies to use PTO-driven centrifugal pumps capable of delivering 10 to 15 gpm (38 to 56 l/min) through a 100 ft (30 m) hose. Specialty requirements, such as tree spraying, will require a higher pressure system, normally a gasoline-powered piston pump. For insecticide treatment, the power source is commonly either electricity or gasoline. Typical pumps are roller pumps, capable of approximately 4 gpm (15 l/min) at 50 lb/in^2 (3.45 bar).

These industries have taken major steps to limit chemical exposure to the general public and to their operators. Efforts have been made by many to eliminate the need of transporting large 100-plus gallon (378 liter) tanks of mixed chemicals. One method that allows

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FIGURE 3 Flatbed service truck (Aro Fluid Products)

the tanks to hold and transport only clean water is to inject the chemical into the discharge only at time of use. The chemical is kept in small, 1 to 5 gallon (3.5 to 20 liter) containers with 12-volt metering pumps attached, which allow direct injection into the clean water line. Other systems use small proportioning pumps. These units are two-stage piston pumps powered by clean water. The smaller piston injects a set volume of chemical into the water supply for each stroke of the larger water piston.

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SECTION 12.18
AEROSPACE

12.18.1
AIRCRAFT FUEL PUMPS

J. E. CYGNOR
Y. J. CHU
D. PALOMBO

The operating characteristics and reliability of aircraft fuel pumps are absolutely critical with respect to the performance and the safety of flight of today's gas turbine-powered commercial and military aircraft. With a large commercial airliner carrying upwards of 50,000 gal (189,000 L) of fuel on a long range flight, making fuel the single heaviest load the aircraft must handle, the importance of intelligent and safe processing of this fuel load is clearly evident.

The aircraft fuel pump system is divided into two separate but related systems: the airframe fuel pump system and the engine fuel pump system. Two systems are required because of the significantly different functional requirements of each system. The airframe system is a low pressure system that operates continuously at low values of inlet NPSH and utilizes booster pumps located throughout the airframe fuel system to supply pressurized vapor free fuel to the engine fuel system. The engine fuel system is a high-pressure system that further pressurizes the fuel for delivery to the engine combustor fuel nozzles and other engine systems. The requirements of these systems and the pumps in them are covered by a wide range of government and industry specifications. Minimally, these define the performance, fuel types, operating and ambient conditions, drive source, reliability, life, weight, installation, quality, materials, and test requirements to which the fuel pump must conform.

The detailed nature of these specifications further emphasizes the critical nature of aircraft fuel pumps. The fuel pump designer's task is to provide fuel pumps that meet these specifications with the lowest weight, best efficiency, and at a competitive cost. Both centrifugal and positive displacement types of pumps are used in performing the necessary fuel pumping functions on essentially all large gas turbine-powered aircraft.

Because of the specific installation and performance requirements for airframe and engine fuel pumps, except the small piston-powered engine in general aviation, each pump is an individual and custom design for each specific application. As such, there are no catalogue type standard pump configurations or designs for medium or large aircraft applications.

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OVERALL AIRCRAFT FUEL SYSTEM

Figure 1 presents a basic schematic of the fuel pump components in the overall aircraft fuel system. The division between the airframe and engine systems is shown.

The primary function of the airframe fuel system is to provide a pressurized fuel feed to the engine fuel pump system under all operating conditions. In the event of the emergency conditions of nonoperating or failed boost pumps, a bypass located either in the airframe plumbing or the boost pump allows the engine pump system to receive sufficient fuel to provide takeoff and climb thrust. Other functions required of the airframe fuel pump system are as follows:

- Provides fuel transfer between various fuel tanks to maintain continuous engine feed and to adjust the center of gravity of the aircraft.

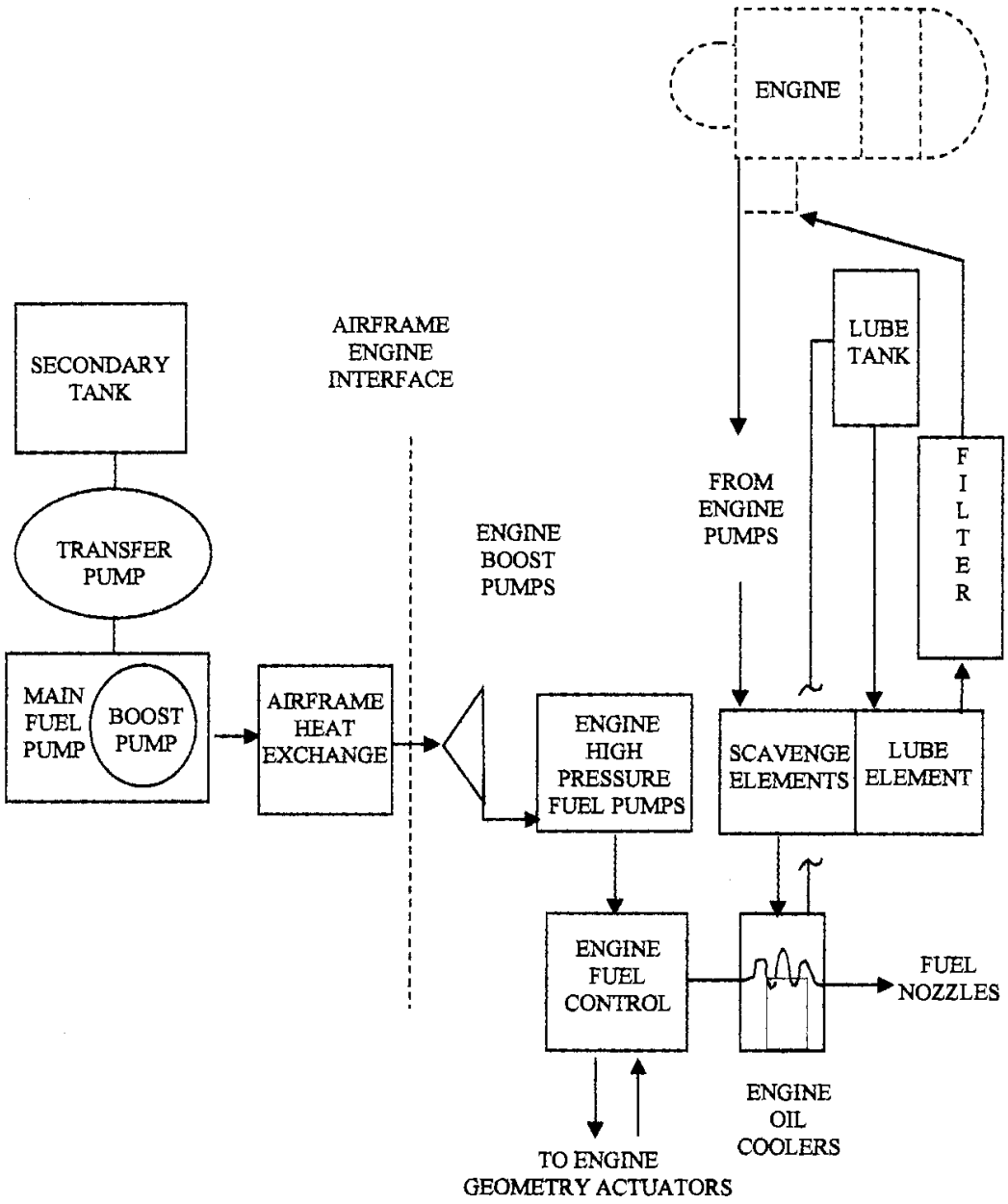


FIGURE 1 Basic schematic of fuel system

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- Provides fuel jettison in flight in the event it is necessary to quickly reduce the aircraft weight.
- Assists in the defueling of the aircraft on the ground.

The primary function of the engine fuel system is to receive the fuel from the airframe fuel system, under normally pressurized conditions or emergency conditions, and provide pressurized fuel to the engine. This fuel is used for burning in the combustors and for powering actuators that control the engine's variable geometry.

Because fuel is the only consumable fluid carried by the aircraft, it represents the sole heat sink on board the aircraft. (For extreme conditions, some aircraft have carried an additional disposable heat sink fluid.) The on-board fuel and the air ingested by the engines or scoops provide all of the cooling necessary for the proper functioning of the various airframe and engine systems. The impact of this on the fuel pumps is a significant increase in the temperature of the fuel on which the pumps must operate.

GENERAL DISCUSSION OF AIRFRAME FUEL PUMPS

The primary type of pump used for providing the fuel boost and fuel transfer functions on an airframe is a centrifugal pump element driven directly by a fuel-flooded electric motor. Positive displacement elements are rarely used and alternate drive means such as hydraulic motors and air or fuel driven turbines have only been used on a selected few military aircraft. On small general aviation aircraft, jet pumps with the motive flow provided by the engine fuel pump are popular.

Helicopters commonly incorporate a special type of fuel system called a "suction feed" fuel system. Because a helicopter is altitude-limited, it is not necessary to provide a pressurized engine fuel inlet condition. In fact, for safety considerations, it is desirable not to pressurize the fuel lines between the fuel tanks, located in the lower sections of the airframe, and the engines, which are located in the upper sections of the airframe. For these applications, the engine fuel pumps are designed to continuously operate with a fuel inlet pressure lower than the fuel tank pressure by the height of the fuel lift between the fuel tank and the engine and the incurred fuel line pressure losses.

AIRFRAME FUEL PUMP DESIGN

The design of an airframe fuel pump is generally characterized by the configuration in which it is mounted in the airplane. The two common airframe pump types are classified in this way; namely,

- In-tank mounted
- Spar mounted

Each type of pump has advantages as well as disadvantages. Although they are all motor driven, each fits different applications and different types of aircraft. To minimize weight, all are made of aluminum except for highly stressed parts such as shafts and fasteners.

The in-tank type of pump is one of the most widely used pumps in the aerospace industry and is normally a cartridge canister pump as shown in Figure 2. Since the pump is installed at the bottom of the tank, it may take advantage of the fuel head inside the fuel tank to help its performance at altitude. (See performance discussion further on.) Therefore, it is often used as a boost pump to feed the engine or in-tank transfer pump for center of gravity (CG) balance. The flow rate or capacity of the in-tank-mounted pump varies greatly from tens of gallons per minute (or m^3/h) for boost or transfer to an order of magnitude more for jettison application. Pressure rises range from 10 to 50 psi (10.69 to 3.45 bar). Because it is structurally simple and does not require a priming element, it is a

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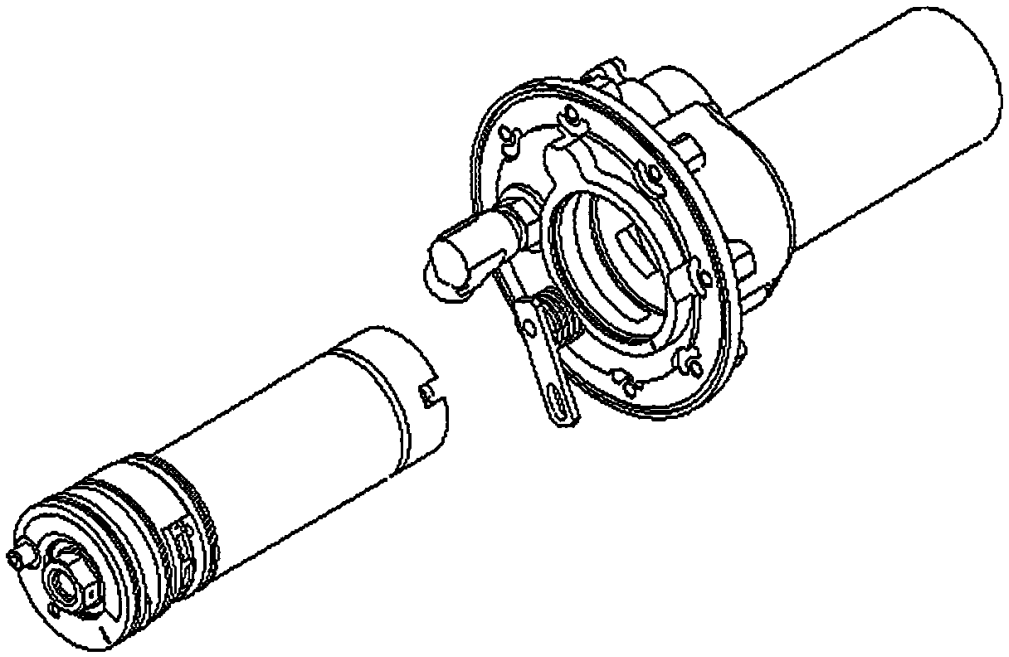


FIGURE 2 Typical in-tank pump (Courtesy of Parker Hannifin Corporation)

preferred choice for the fuel system engineer. However, there may be three distinctive drawbacks of this type of pump:

- It may weaken the wing structure, as most of the in-tank type of pumps are of the cartridge-canister style, and an access hole is required for the pump cartridge at the bottom of the wing.
- It may not achieve the pump-down level required, because the cartridge-canister pump requires some floor clearance to allow the fuel to flow into the impeller at the pump inlet, therefore, the fuel quantity below the pump impeller eye may become the "unpumpable volume." To minimize this volume, an ejector or jet pump (Chap. 7) may be deployed at the bottom of the tank.
- The design phase requires close scrutiny to ensure safety, because the entire pump including the motor is inside the fuel tank and therefore must be explosion-proof. (See discussion below on safety requirements.) This concern is less challenging if the pump is fully submerged in the fuel, because the fuel serves to cool the pump and to quench any possible ignition source.

The spar-mounted pump has flow rates and pressure rises that are similar to those of the in-tank pump, and it offers some attractive features which may compensate for the shortcomings of the in-tank pump. As shown in Figure 3, it consists of three elements: the cartridge, mounting plate, and snorkel. The electric motor, which is included in the pump cartridge, is located outside the fuel tank in a dry area; it is inherently better than the in-tank pump in terms of safety. The spar itself is not subjected to the wing deflection. Furthermore, the inlet snorkel of the pump can be deployed to the lowest spot of the fuel tank to achieve the optimum pump-down level. Since the fuel must be lifted to the main impeller, the altitude performance of this type of pump may not be as good as the in-tank type.

MOTOR DRIVE OF AIRFRAME FUEL PUMP

The drive motor of the pump deserves some special attention in this section, because of the rapid change in motor technology as applied to aircraft fuel pumps. As shown in Figure 4, motors for aircraft fuel pumps have evolved into a wide range of types.

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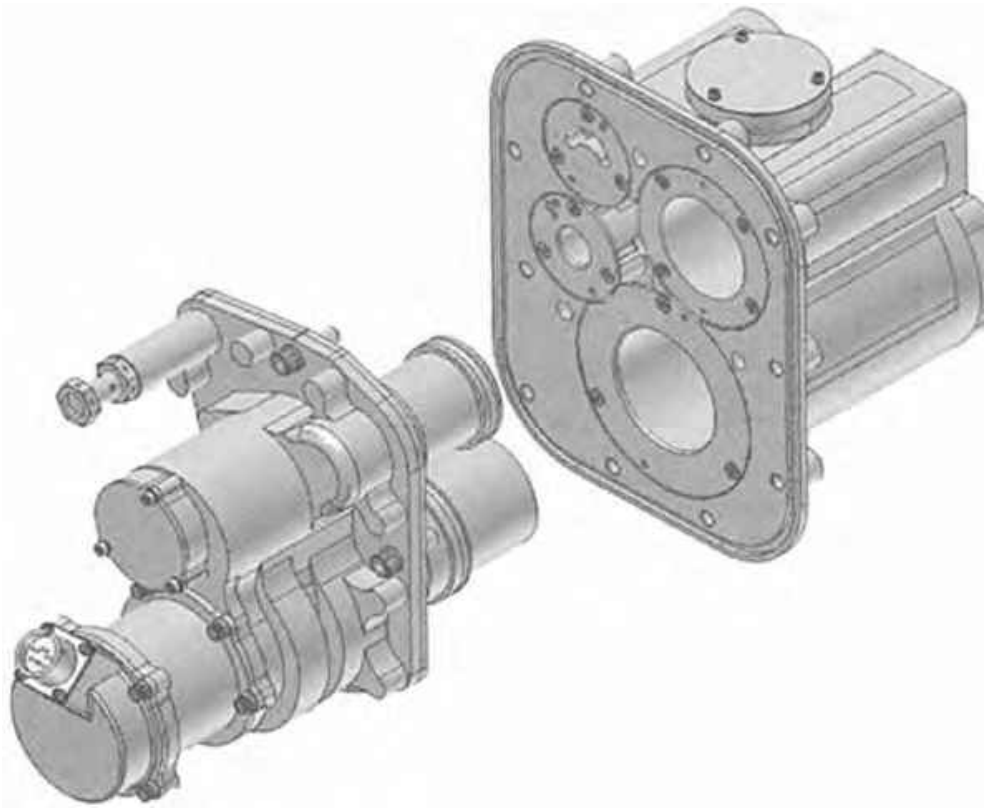


FIGURE 3 Spar-mounted pump with preferred snorkel shown

Prior to 1980, almost all motor-driven pumps in aircraft fuel applications were either the low voltage brush-type motor or constant-frequency ac induction motor. However, the variable-frequency (VF) pump started to become popular in 1990, as did also the electronic-drive motor-driven pump. Although the 28-V brush-type dc pump and 400-Hz ac induction motor-type pump still dominate the field, the growing number of electronic-drive

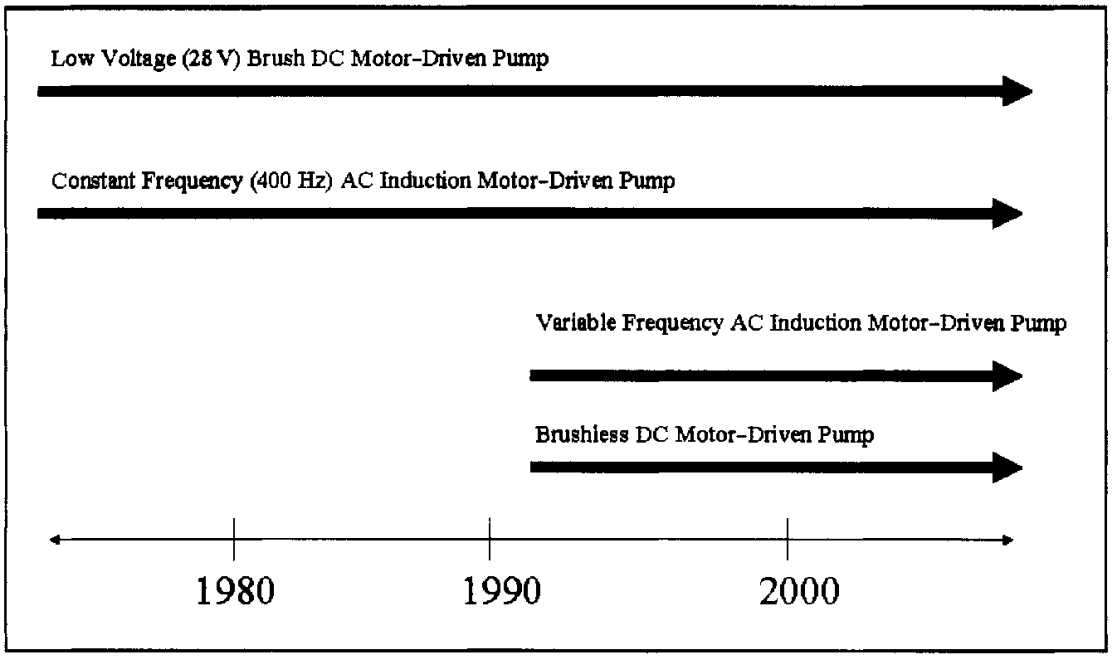


FIGURE 4 Evolution of airframe motor-driven fuel pump

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motor-driven pumps (brushless dc motor-driven pump) will make it a major player in the future. In fact, the brushless dc motor-driven pump has been used in almost every airplane since 2005. The description of each type of motor drive for airframe fuel pumps is shown below:

Low-Voltage Brush-Type Motor-Driven Pump This is one of the most popular types of fuel pump used in the aircraft industry. Most of these pumps are used in the piston engine-powered aircraft of General Aviation. More than half a million pumps of this type have been produced in the industry. These pumps are powered by either battery or low-voltage power bus. In most of the applications, there is no speed control on the motor; therefore, the pump speed varies greatly with the load of the pump, and so do the flow rate and pressure rise of the pump. In order to extend the life of the motor, the motors used in this type of pump are fuel flooded. This not only eliminates the life-limited dynamic seal between motor and pump but also provides constant cooling and lubrication flow for motor and bearings. Due to weight and power constraints, this type of pump is seldom used at a power level above 1 hp (746 W). For smaller aircraft, it is widely used for the boost or feed pumps. The strengths and weaknesses of this pump are as follows:

- Strengths
  - Reliable
  - Inexpensive
  - Simple power interface
- Weaknesses
  - Suitable only for low power application
  - Limited life expectancy
  - Limited dry run capability
  - High in-rush or starting current
  - Performance varies due to the load change

Although this type of pump is quite simple, its performance is very sensitive to load variation. As shown in Figure 5, a small change in the pump load (expressed here as torque) may cause some change in the pump speed; consequently, the pump performance may vary. Since it contains brushes for commutation; it is not recommended for prolonged dry run. In some applications, additional resettable thermal protection devices may be required to accommodate this requirement. Because there is a limited life expectancy for brushes in the motor, the pump needs to be overhauled after a limited number of hours of operation. The starting current of the pump may be undesirable for electrical systems. It is not uncommon to see more than 600% of normal running current at pump start-up for a very short period of time. Similarly, if the pump encounters a locked-rotor condition, the high current could destroy the motor unless the thermal-protection device of the motor is tripped.

Constant Frequency (CF) AC Induction Motor-Driven Pump This type of pump is primarily driven by a 400-Hz, 115-V ac (line to neutral), or 200-V ac (line to line) motor. Both three-phase and single-phase motors of this category are widely used in aircraft applications. However, the three-phase one is very popular, being used in almost every large transport—both commercial and military. The advantages of using a three-phase motor are compactness, high power-to-weight ratio, and superior reliability. A properly designed three-phase motor and mating impeller can pump up to hundreds of gpm in a package that is a few pounds in addition to overall motor weight. (m^3/h = 0.227 gpm.)

Unlike the brush-type motor, there is no contact between the stator and rotor; hence, the life of the induction type motor is extremely high and commonly reaches hundreds of thousands of hours before overhaul. As shown in Figure 6, when the load (in terms of torque) varies from T_1 to T_2, the speed variation is small. Therefore, the performance of the pump is very predictable. The features of this type of motor are summarized as follows:

- *Simple and robust.* No extra parts are required besides the rotor and stator.
- *Stable operating speed over wide torque range.* Load of the pump has little effect on the pump speed.

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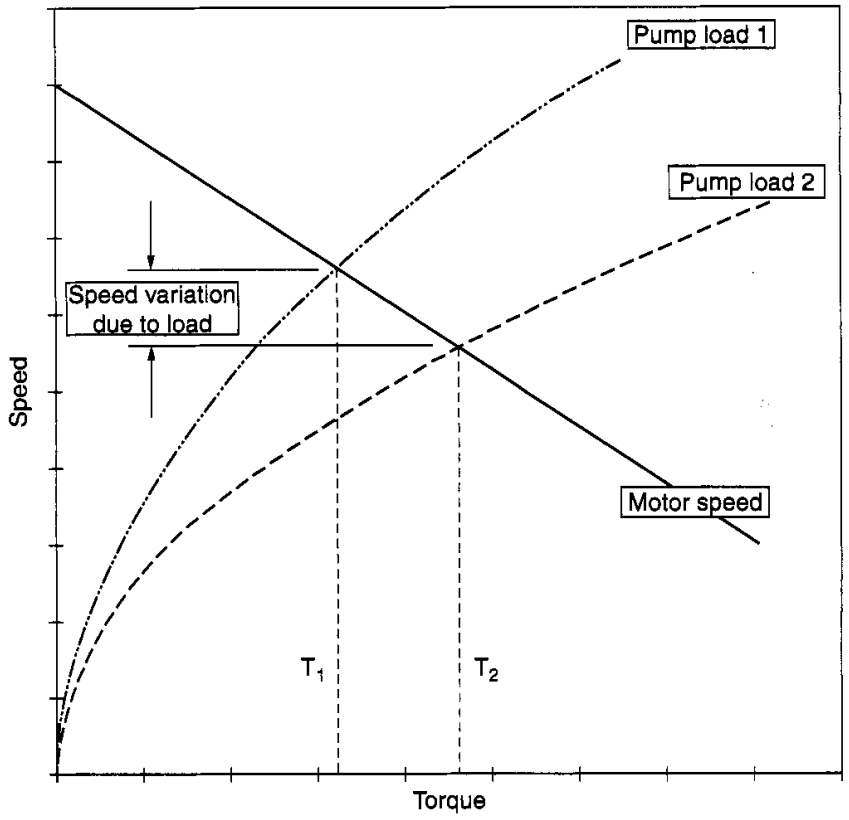


FIGURE 5 DC motor-driven pump load

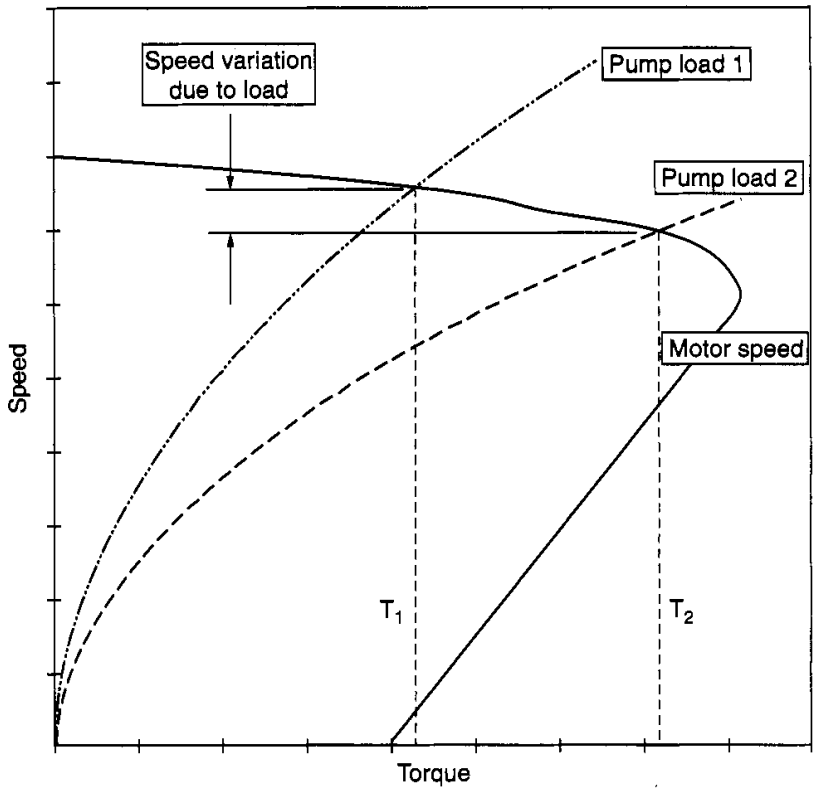


FIGURE 6 Constant frequency ac induction motor-driven pump load

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- *Dry runnable.* The properly designed ac induction motor may be capable of running dry for an extended period without overheating of the motor.
- *High efficiency.* Peak efficiency is around 80 to 85%; and pumps are commonly designed to operate at both this peak and at peak power factor, thereby consuming minimum energy and current for maximum hydraulic output.
- *Low starting or in-rush current.* Unlike the brush-type motor, the starting current of the ac induction motor is seldom over 350% of its steady-state operating current.

Variable Frequency (VF) AC Induction Motor-Driven Pump The increasingly popular VF motor-driven pump is basically the same as the CF-type pump; however, it allows the elimination of the constant-speed drive from the aircraft electrical system. In the VF power system, the pump power supply is directly connected to the aircraft generator without the constant speed drive in between. The frequency of the power supply to the pump motor is directly proportional to the engine speed. Normally, the VF motor-driven pump operates with the power frequency from 300 to 800 or 900 Hz. However, since the pump power consumption is proportional to the cube of the speed, such a large range of motor frequency results in a larger range of the pump load (in terms of power.) Consequently, an oversized motor may be required to cope with the large change of the load if the speed of the motor is not allowed to slip substantially below synchronous speed at high input frequency. As a result, there are two types of VF-powered pump in the aircraft industry:

- The high-slip ac induction motor pump
- The electronic-drive motor-driven pump (brushless dc motor-driven pump with power factor correction module)

Balancing the pump load (torque) without significantly increasing motor weight and the pump power at high frequency becomes an interesting trade-off. As shown in Figure 7, for a given family of performance curves of VF motors, the pump speed can vary greatly depending upon how and where both the pump load curves intersect with each other. One way to deal this wide frequency range in the VF motor without penalizing the motor weight is to use a "weak" motor and allow the motor speed to slip substantially from its synchronous speed. By doing so, the pump load will be allowed to increase only to a limited extent. It is also not unusual to see the speed of the motor slip in the range of 30 to 40% when it is

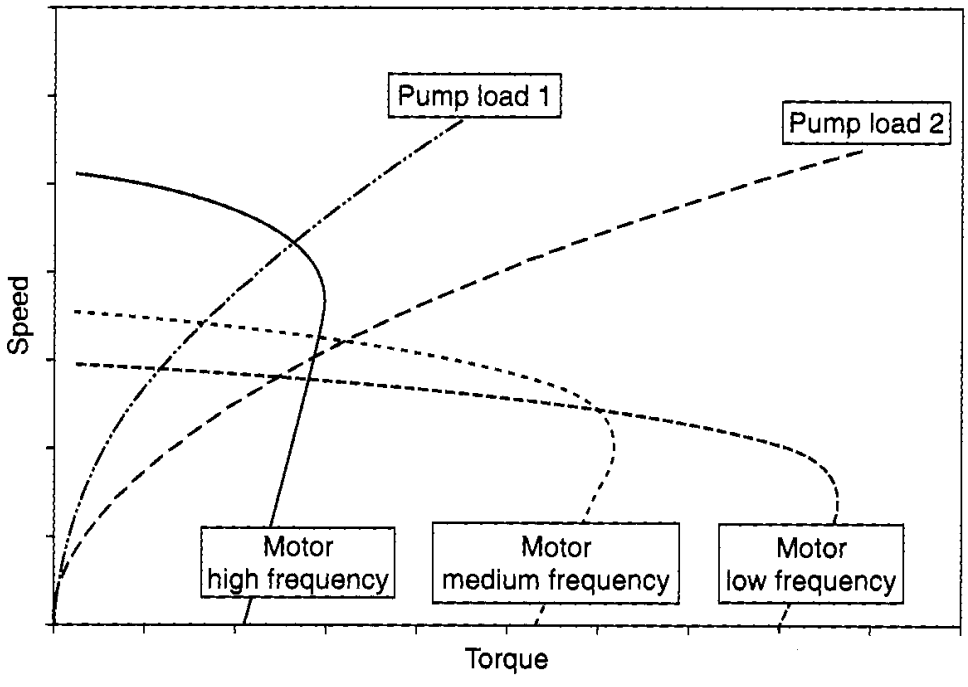


FIGURE 7 Variable frequency ac induction motor-driven pump load.

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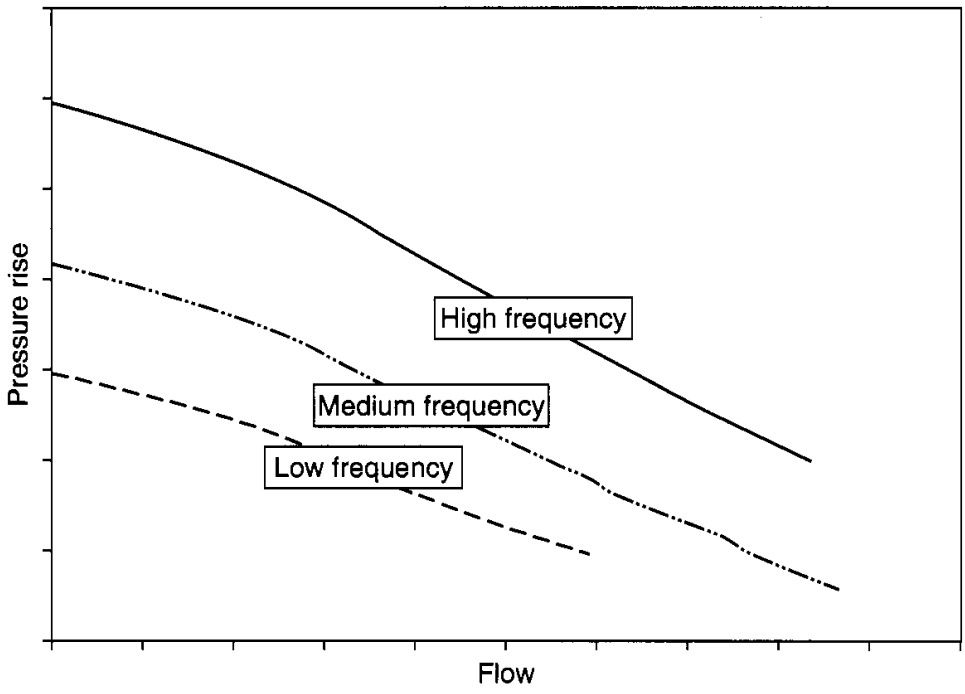


FIGURE 8 Performance curves of variable frequency ac induction motor-driven pump load 1

operated at high end of frequency range (800 to 900 Hz). If the pump load is either small enough or the motor is robust enough at high frequency, then the pump performance will be as shown in Figure 8. However, if the motor is weak and the slip in motor speed is excessive at high frequency—such as illustrated by the curve *Pump Load 2* in Figure 7–then pump performance will behave as shown in Figure 9. From this discussion emerge the inherent weaknesses of the VF motor-driven pump with a high-slip induction motor:

- *Poor power factor and motor efficiency through the entire frequency range.* This situation becomes extreme when the pump operates at the low end of the frequency range; for

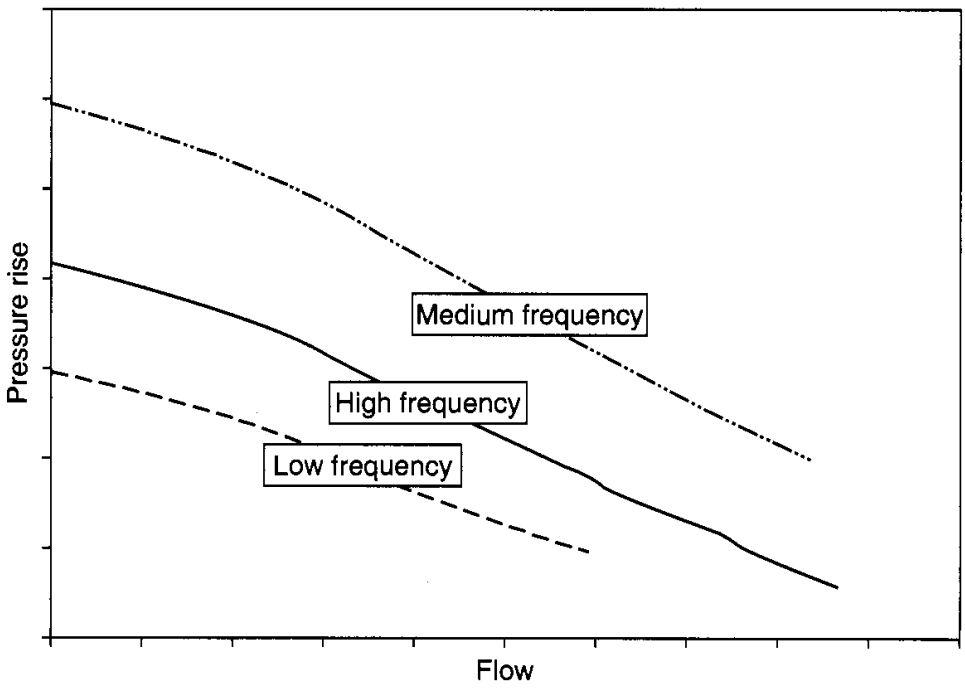


FIGURE 9 Performance curves of variable frequency ac induction motor-driven pump load 1

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example, power factor can be as low as 30 to 40%. Sometimes, it only reaches 30 to 40% in power factor at 360 Hz if the full frequency range is from 360 to 900 Hz.

- *Overheating of dry run at low frequency.* When operated at low frequency, a VF motor is saturated; so that, regardless of the load on the motor, it consumes the same amount of current even under dry run conditions. Consequently, if the pump encounters a dry run condition at low frequency, the motor may easily become overheated.
- *High locked-rotor current at low frequency.* The locked-rotor current of a VF motor is inversely proportional to the square of the motor frequency. Therefore, the attendant wide range of locked-rotor current from low frequency to high frequency becomes critical for the electrical system.
- *Low starting torque at high frequency.* The starting torque of a VF motor is inversely proportional to its frequency. This may become an issue if the pump has a high static load. If the starting torque of the motor is not capable of overcoming the static load of the pump, the motor may stall.

Brushless DC Motor-Driven Pump The brushless dc motor has been widely used in other industries for several decades; and, due to advancement in the power electronics of the motor controller as well as a sharp decline in controller cost, it has more recently gained momentum in application to aircraft fuel pumps. It is distinguished from the other pump types by the following unique features:

- It can vary the pump speed while remaining efficient; hence, the output of the pump can be "on demand" to precisely match the aircraft mission requirement. Therefore, it may reduce the power consumption of the pump.
- The brushless dc motor is approximately 5 to 15% more efficient than either the constant frequency ac induction motor or the brush-type dc motor. Moreover, when the pump runs dry, it consumes only a fraction of power consumed by the brush dc motor or ac induction motor.
- It can incorporate some intelligence in the motor controller such as over temperature protection, over current protection, failure reporting/isolation, health diagnosis and prognostic functions. Therefore, it may save some maintenance cost.
- It can provide a "soft start" of the pump and so reduce the in-rush current at pump start-up, or it may offer faster response to the control signal.
- It can interface with other components in the fuel system such as pressure switches or pressure transducers to supplement or replace the functions of those components and so simplify the fuel system.

On the other hand, although the stator and rotor of the brushless dc motor may be as robust as those of the ac motor, there is a weakness of the brushless dc motor-driven pump. This is the loss of reliability of the electronic components and solder joints on the circuit board of the motor controller if these are exposed to high temperature. Most industrial grade electronic components are rated at 40 to 85°C. The cost associated with enhancing the reliability and ruling out infant mortality of the electronic components remains an issue in the application of brushless dc motor-driven aircraft fuel pumps.

CONTROL OF BRUSHLESS DC MOTOR-DRIVEN FUEL PUMPS

Due to its unique features and the importance of the brushless dc motor in aerospace application, the electronic motor controller deserves some special attention. Generally, the electronics can be split into two sections, the input power filter and the motor drive.

The diagram in Figure 10 depicts the major components in the drive electronics. The input filter would be everything to the left of the dc link capacitor. (Current research may lead to the elimination of the dc link—in an integrated ac-to-ac converter or "matrix converter.")

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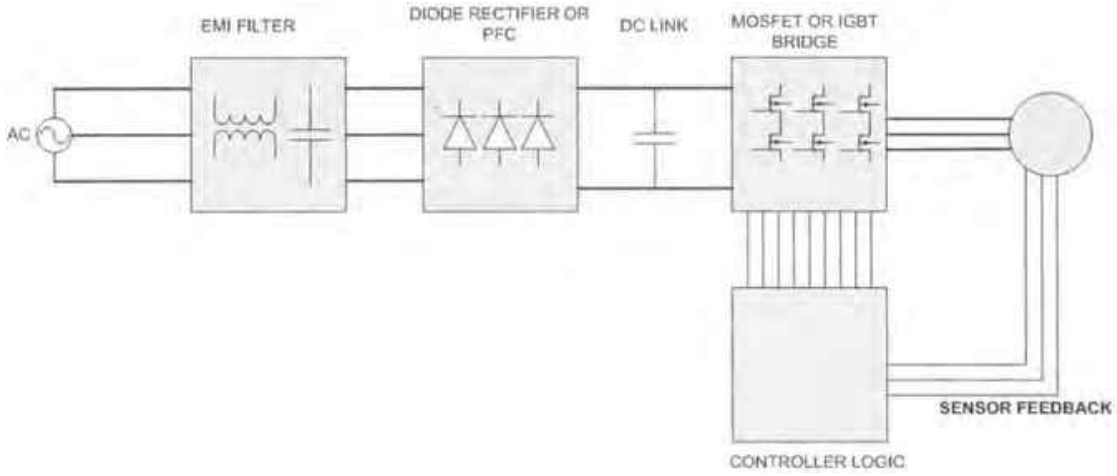


FIGURE 10 Diagram of a basic brushless motor drive

Input Power Filter At a minimum, the input power filter must prevent conducted energy from the motor drive from polluting the input power source. The two sources of emissions are called common mode (CM) and differential mode (DM). Motor controllers usually have transistors that switch at very high frequencies, which implies that the voltage V across them have very fast rise and fall times t. CM noise originates from the stray capacitance between the power devices and the pump chassis driven by the dV/dt across the power transistors. The source of the DM noise is from the ripple current created by the high frequency switching of the motor controller. The filter for these CM and DM noise sources is usually a collection of inductors and capacitors which create a high impedance to these high frequency noise sources.

If the input to the pump is ac, then the input filter, at a minimum, must convert the ac into dc but may also have the task of power factor correction (PFC). Power factor is the ratio of real power to apparent power (distortion PF) multiplied by the cosine of the angle between voltage and current (displacement PF).

Consider the circuit in Figure 11 and the waveforms in Figure 12. The diode bridge and capacitor form a type of peak detector, with R1 representing the load. The diodes become forward biased (conducting current) only when the voltage from the anode to the cathode exceeds the voltage drop of the diode plus the voltage that is stored in the capacitor. When the input ac voltage falls below this point, the diode stops conducting and the current in the ac line falls to zero. The resulting current waveform, although perfectly in phase with the input voltage, has high peaks containing many higher order harmonics. These harmonics create reactive currents in the line which contribute to RMS heating of the power lines to the controller, necessitating over sizing of the wire feeding the pump. If the input to the pump is dc, then this section is omitted.

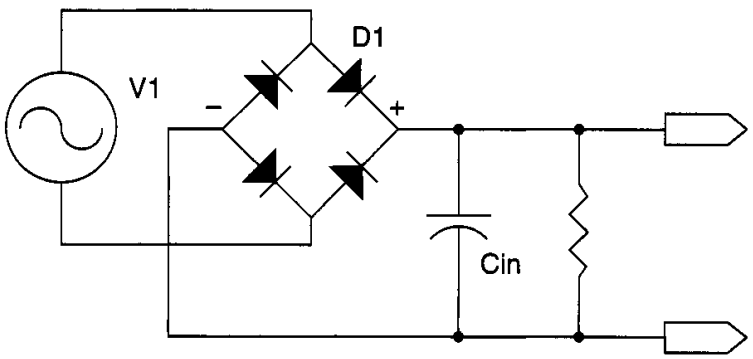


FIGURE 11 Simple diode bridge and capacitor bank

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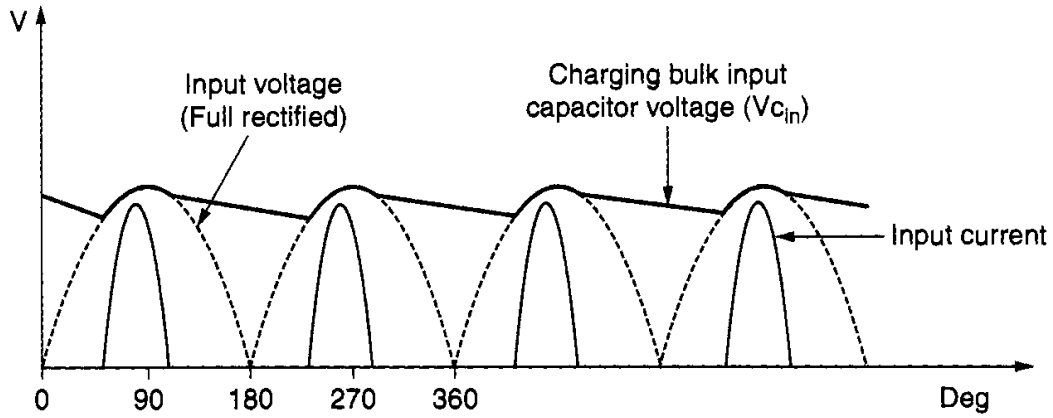


FIGURE 12 Waveforms of circuit in Figure 11

Power factor correction can be accomplished with pure passive components (capacitors, inductors, and transformers) or with a combination of active components (high frequency power electronics) and passive components. The use of active components running at high frequency may allow smaller passive components, but their advantage is somewhat mitigated by the fact that additional CM and DM noise sources are created, thereby complicating the design of the input power filter.

Motor Driver The function of the motor driver for a brushless dc motor is to replace the function of a mechanical brush and commutator. As the permanent magnet rotor rotates with respect to the stator, the controller must continually reverse the current flow in the individual coils to always keep torque in the same direction. Commutation of the motor is just the beginning. The complexity of the drive is a compromise between its desired functionality and the complexity of development in an aerospace environment. At one end of the spectrum, a brushless motor-driven pump can appear like a dc brush motor pump, simply spinning at a speed proportional to the input voltage as soon as power is applied. At the other end of the spectrum, the control electronics can adopt sophisticated control algorithms to vary the speed of the pump based on local sensors and commands from a digital communication bus.

A brushless motor controller must sense the rotor position with respect to the stator position in order to properly energize the stator coils. At the onset of a motor drive design for a brushless motor one must make a choice as to whether the controller will use embedded sensors in the motor for position feedback or use a "sensorless" scheme. A sensorless motor controller detects the rotor position by looking at the Back-EMF or current of the stator phases. With a sensorless design, no delicate electronic devices are mounted near or on the motor and only three phase wires connect the motor to the controller. On the surface, it may appear that the sensorless approach is the best option. But the complexities of developing in an aerospace environment, and almost always the necessity for either software or other programmable devices to realize a sensorless design can make the choice of adding sensors to the motor more economical over the life of the program.

There are instances that require motor position feedback as well. A high resolution position feedback device (resolver or encoder) may be required for very precise position or velocity control, operation at very slow speed or an extremely wide dynamic operating range. Digital signal processors (DSP) and control algorithms are becoming more and more sophisticated, which may obsolete the need for position feedback, even with the most challenging applications. If the environmental conditions permit, a simple hall-effect-type sensor can be integrated directly into the stator, providing a low-cost and reliable low resolution position sensor, suitable for commutation and velocity feedback.

However, if the design permits, adding microprocessors to a controller adds incredible flexibility. For instance, a pump can read a local pressure or flow sensor and vary the pump speed accordingly. Communication busses (CAN, RS-232, and so on) can be added to interrogate the pump's current status, health, operating time, historical running conditions, faults, and other information.

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Another consideration of a brushless motor controller is whether to run the motor as a "six-step" trapezoidal or with field-oriented control (FOC). A six-step drive is very simple in operation, but often can create torque ripple in the motor which may manifest itself as extra noise. FOC applies a sine wave to the motor phases, and with the proper motor (sine wave Back-EMF), a very smooth drive can be realized, with low vibration.

PERFORMANCE CRITERIA OF AIRFRAME BOOST PUMPS

The fact that aircraft boost pumps operate at very low net positive suction heads is evident in their operating environment. Fuel tanks are vented to ambient pressure and commercial airlines fly up to 45,000 ft altitudes where the standard atmospheric pressure is 2.14 lb/in^2 (.148 bar). Altitudes are even greater for supersonic aircraft. Tank pressurization above the ambient pressure is employed only on some specialized military aircraft.

The performance criteria that specifically define the design of the centrifugal pumping element in an airframe boost pump are presented in the following discussions.

MAXIMUM/MINIMUM PRESSURE RISE AND MOTOR SPEED

The pressure rise limits of the pump are defined by the pump specification and are determined by airframe and engine system needs.

The motor speed range is selected by the pump designer to be compatible with the overall requirements of the pump, in particular the altitude requirements. Because of the need to minimize size and weight, the maximum possible speed is employed. This has, in recent years, resulted in the almost exclusive use of axial inducers in airframe boost pumps. They are used either alone, as purely axial pumping elements, or in conjunction with mixed-flow sections depending upon the flow and pressure requirements. Axial inducers are a pumping element that have proven to provide the best suction performance because of their low blade loading and gradual pressure rise characteristic. The inducers on airframe boost pumps incorporate features such as blade sweep back, blended suction surfaces, and thin blade inlet edges. The inlet blade angle at the tip usually ranges from eight to ten degrees measured from a plane perpendicular to the axis of rotation. These pump elements have demonstrated the capability of operating at suction-specific speeds in excess of 40,000 measured in units of gpm, feet of fluid, and rpm (that is, the universal suction specific speed Ω_{ss} exceeds 14.6).

FUEL TYPE

Two types of fuels are in general use worldwide in aircraft gas turbine engines. They are referred to as "wide cut fuels" and "kerosene-based fuels."

Both types of fuel are composed of a complex mixture of a range of individual hydrocarbon compounds. As the name suggests, the wide cut fuels have a wider range of hydrocarbons than kerosene-based fuels. The composition of the fuel is controlled, rather loosely, by the fuel specifications through defining limits on such factors as distillation range, density, flash temperature, heat of combustion, vapor pressure, freezing point, additives, and limits on certain compounds. Wide cut fuels are characterized by relatively low density, low freezing point (−65°F/−53.9°C), high vapor pressure and low flash point (−45°F/−42.8°C). Kerosene-based fuels on the other hand are characterized by a relatively high density, higher freezing point (−40°F/−40°C), low vapor pressure, and high flash point (140°F/60°C). The U.S. designations for wide cut fuels for commercial and military uses are Jet B and JP-4, respectively. The designations for kerosene-based fuels are Jet A1 for commercial uses and JP-5 and JP-8 for military uses. Jet A1 predominates in production because of the huge demands of the airlines. Jet A1 is used in commercial service worldwide

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primarily for safety reasons because of its high flash point. Jet B is only used when its low freezing point is required; for example, in northern Canada in the winter. The U.S. Air Force previously used JP-4 as their standard fuel, but for availability and safety reasons have switched to JP-8, which is a military version of Jet A1. The navy uses JP-5 because of its high flash point for safety considerations on board aircraft carriers.

The knowledge of the true vapor pressure of the fuel used is necessary in the design and test evaluation of the centrifugal pump element. Because the fuels are a mixture of a range of hydrocarbons, the direct determination of the true vapor pressure is difficult. It is determined indirectly, for a desired temperature, through the Reid vapor pressure (RVP). The RVP is an average vapor pressure determined under a defined set of conditions, established by test at 100°F (38°C). The true vapor pressure (TVP) at this temperature is slightly higher than the RVP. Moreover, the TVP-versus-temperature relationship is determined by the RVP (see Reference 1).

Another key property of the fuel that directly influences the design and performance of airframe boost pumps is the solubility of air in the fuel. This is defined by Henry's law through a solubility coefficient. The ullage volume above the fuel in the aircraft tanks is generally vented to the ambient pressure. The maximum amount of dissolved air in the fuel will occur on the ground. The fuel will be in an air-saturated condition and can then contain as much as 14% dissolved air by volume.

Any reduction in the pressure of the fuel will result in the release of this air in accordance with Henry's law and expansion of this air to a volume in accordance with Dalton's law of partial pressures. In addition, as the pressure is reduced, the vapor pressure of the light end hydrocarbon constituents of the fuel is reached. They too will vaporize, adding to the volume of vapor evolved. Reference 1 provides a detailed review of all of the properties of aircraft gas turbine engine fuels.

ALTITUDE CLIMB PERFORMANCE

The altitude climb performance required of an airframe boost pump is specified in terms of altitude achieved versus time in minutes, flow rate required with respect to altitude, the fuel tank temperature with respect to altitude, and the minimum pump pressure rise required versus altitude. The temperature versus altitude variation is due to the cooling of the fuel through heat transfer to the cold ambient atmosphere and boiling of the fuel that is experienced in a climb event.

Immediately upon takeoff, as the aircraft gains altitude, the dissolved air in the fuel will begin to evolve. The rate of this air evolution will depend upon the climb rate of the aircraft. Presently, there is no accurate way to determine this rate of air evolution. This volume of evolved air is handled in the design of the pump by judicious oversizing of the inlet based upon experience and empirical design parameters. Further assistance in handling the volume of vapor is provided in tank bottom mount pumps through pump element inlet tip vapor vents and discharge hub vapor vents. Snorkel inlet pumps can receive vapor handling assistance through the liquid ring reprime element. Any remaining air and fuel vapor mixture is compressed as it passes through the increasing pressure within the inducer and is redissolved into the fuel. Therefore, the pressurized fuel delivered to the engine is free of vapor.

As the altitude increases, a tank pressure will be reached that will equal the vapor pressure of the fuel. At this point, especially with wide cut fuels, the vapor release will become quite violent (boiling). Here, the rate of vapor evolution will be determined by the capacity of the tank vent system. As the light ends leave the fuel, the vapor pressure of the fuel will decrease. This is referred to as "weathering" of the fuel. After the maximum cruise altitude is reached, the vapor pressure of the fuel will reach equilibrium with the altitude pressure.

The altitude climb test is the most difficult test to perform on airframe boost pumps because it must be a single fuel pass test without recirculation to accurately reproduce an actual climb condition. This requires approximately 1500 gal (5,680 L) of fuel for one test

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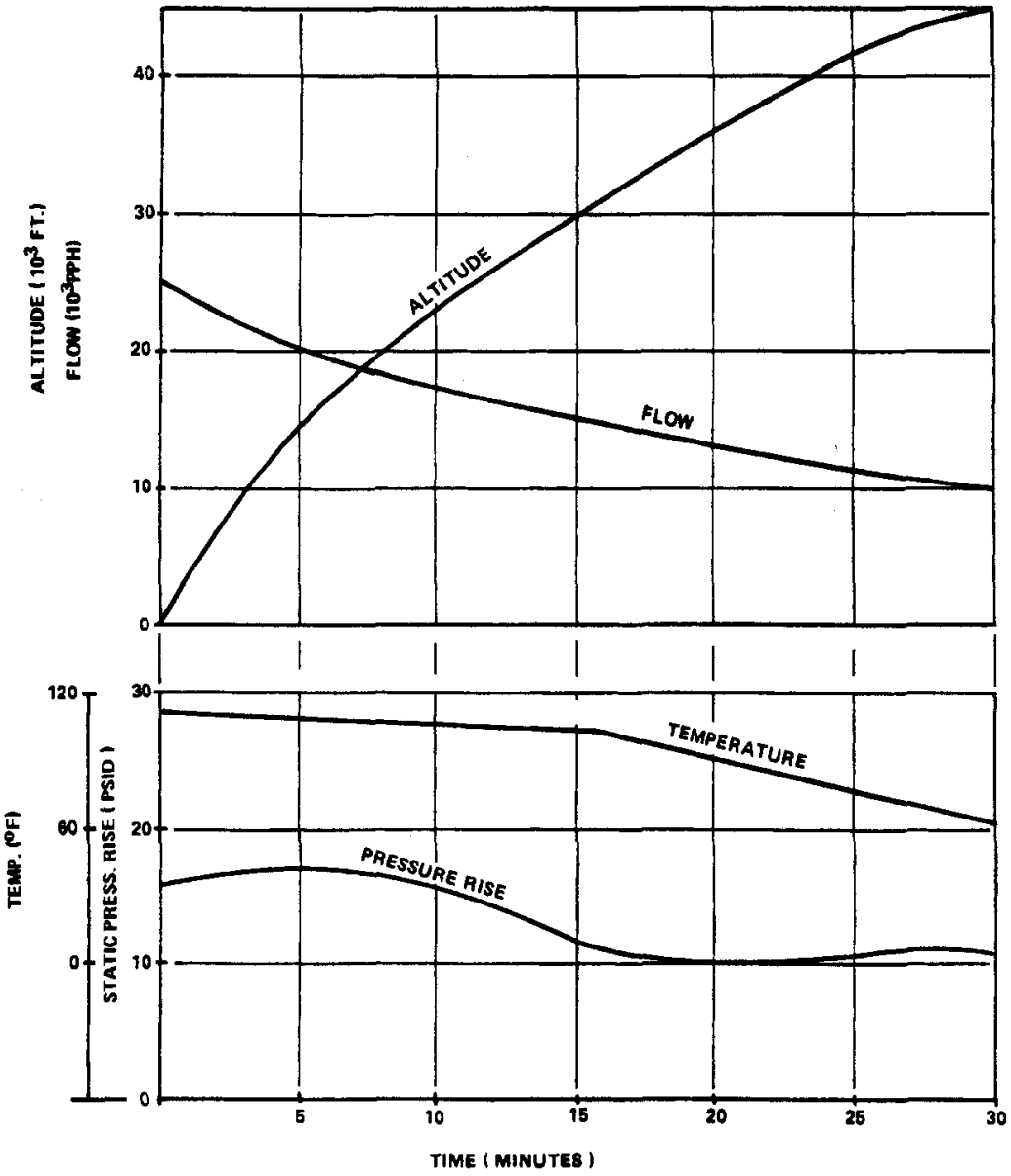


FIGURE 13 Main boost pump altitude climb (kg/h = PPH × 0.4536; m = ft × 0.3048; bar = psi × 0.06895; °C = (°F − 32) × 0.556)

on an average commercial airliner fuel boost pump element. Figure 13 presents the results of an altitude climb test on a typical main fuel boost pump.

CONSTANT ALTITUDE PERFORMANCE

This requirement for an airframe boost pump is specified in terms of a required fuel flow range at a given temperature and at a constant altitude. The minimum required pressure rise is also specified. The submergence of the inlet of the pump (for bottom mount types) or the snorkel inlet (for snorkel inlet types) is also defined and is measured in the range of a few inches. For altitude pressures lower than the initial fuel vapor pressure, the fuel is weathered to achieve equilibrium with the altitude pressure. The fuel is recirculated in this test. Figure 14 presents the constant altitude performance of the typical main fuel boost pump that has the climb performance of Figure 13.

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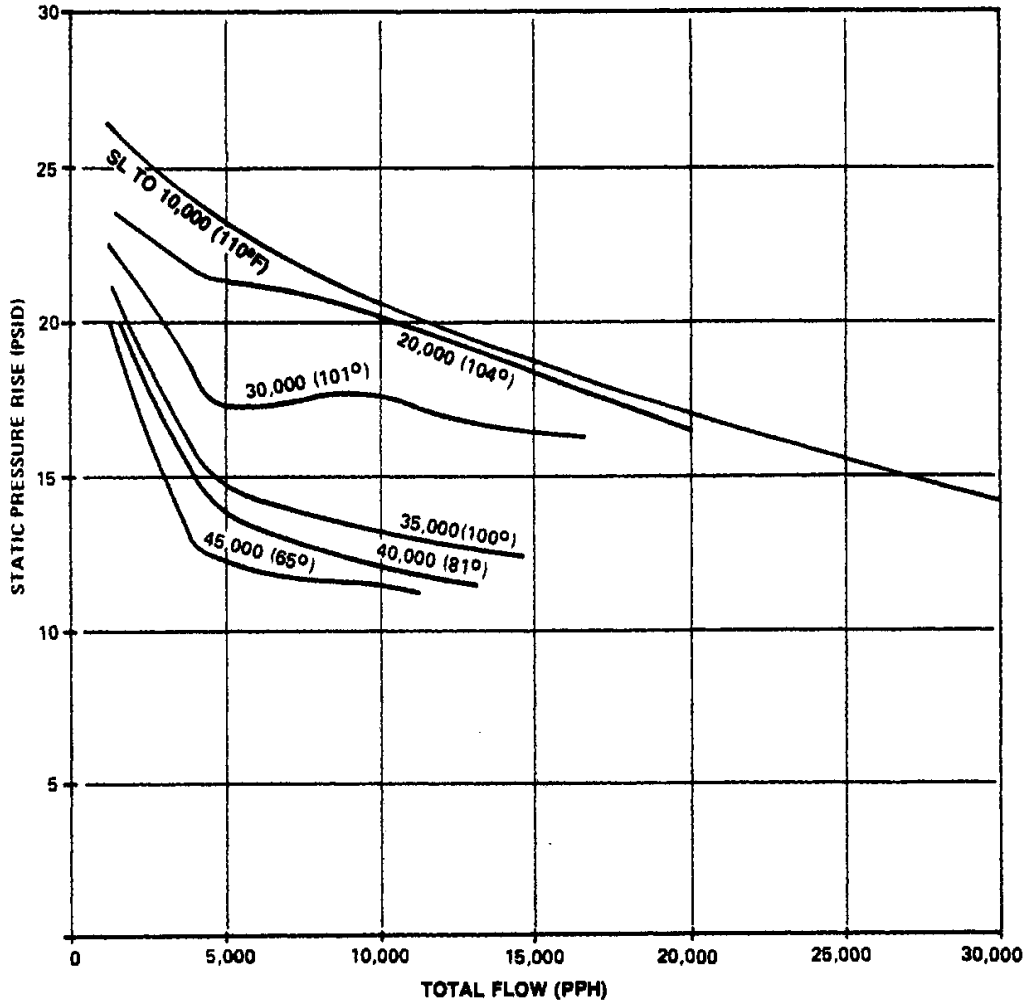


FIGURE 14 Main boost pump altitude performance (Altitudes are shown on the curves in ft; m = ft × 0.3048; kg/h = PPH × 0.4536; bar = psi × 0.06895; °C = (°F − 32) × 0.556)

SAFETY REQUIREMENTS FOR AIRFRAME FUEL PUMPS

Because they are one of the components which consume the most energy inside the fuel tank, airframe motor-driven fuel pumps have been closely scrutinized since the accident of TWA Flight 800 in 1996. Any possible ignition source in the fuel tank needs to be carefully examined and eliminated. Ignition may be caused by failure of either mechanical or electrical components within the fuel tank. The common failure modes of the airframe motor-driven fuel pumps in the aviation industry have been carefully documented in Reference 2, which requires that both the pump designers and users follow the guidelines for checking the pump safety features so as to avoid any malfunction, failure, or degradation.

Special attention in this regard is paid to the pump motor. All openings that communicate the pump electric motor cavity with the interior of the fuel tank must incorporate flame-arresting features. This includes communication through the pumping elements to the pump inlet. Also nonresettable thermal fuses or fusible links are incorporated in the motor end turns to ensure that the motor is disconnected before it can reach the minimum autoignition temperature of jet fuel (approximately 390°F/199°C) through various failure modes. The pump designer must follow the design and test guidelines in Reference 3, which includes attention not only to normal operation but also to the abnormal conditions such as dry run and locked rotor.

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GENERAL DISCUSSION OF ENGINE FUEL PUMPS

The primary fuel pump used for providing the boost and high-pressure fuel pumping function for an engine is called the "main fuel pump." It is engine gearbox-mounted and driven. In its most common form, it includes a positive displacement external spur gear high-pressure stage and an integral centrifugal boost stage. This type of pump is exclusively used on all modern commercial airliner engines and supplies all of the fuel flow requirements of the engine combustors. On commercial airliner engines, the main fuel pump is the sole engine fuel pump and therefore is a prime reliability piece of equipment. All engines, commercial and military, have a main fuel pump.

Positive displacement pumps have certain operating characteristics that make them very adaptable to aircraft gas turbine engine fuel systems. These characteristics include the ability to reprime from a completely dry condition and the ability to deliver useful fuel pressure over a wide speed range. Engine control systems utilize a portion of the main fuel pump discharge flow to power the actuation systems for fuel burn flow metering and engine variable geometry control for all operating conditions. Corner point operating conditions, such as altitude relight and ground starting, occur at 8 to 10% of rated operating speed. The pressure required for actuator muscle and response characteristics at these conditions is generally a 250 lb/in^2 (17.2 bar) pressure rise. At the maximum power takeoff condition at 100% speed, the pressure rise requirement ranges up to 1500 lb/in^2 (103 bar). Positive displacement pumps have proven capable of meeting these needs and the external spur gear pump type has received the widest acceptance from the industry.

The key attribute of the external spur gear main fuel pumps relates to the safety of flight and prime reliability requirements. In over 50 years of experience, this type of pump has demonstrated a graceful type of failure mode. It does not fail in a catastrophic manner in which it suddenly ceases to function. It slowly, over an extended period of time, degrades in performance until it no longer meets minimum requirements, but continues to function. In commercial service, these pumps routinely operate for over 10,000 flight hours without service or repair.

Table 1 presents the pertinent characteristics of representative external spur gear main fuel pumps.

TABLE 1 Main fuel pump parameter chart

Engine Application and Engine Power	Gear Stage Displacement in^3/rev (cm^3/rev)	Rated Input Speed RPM	Rated Output Flow GPM (m^3/h)	Rated Pressure Rise lb/in^2 (bar)	Unit Weight Pounds (kg)
• Commercial Airliner • 68,000 Pounds (302 kN) Of Thrust	3.487 (57.142)	6270	86 (19.53)	1250 (86.2)	34 (15.42)
• Fighter Aircraft • 28,000 Pounds (124 kN) Of Thrust	2.341 (38.36)	6075	55.2 (12.54)	1390 (95.8)	24.5 (11.11)
• Executive Jet • 5,000 Pounds (22 kN) Of Thrust	.631 (10.34)	6300	15.5 (3.52)	1200 (82.7)	12 (5.44)
• Helicopter • 650 H.P. (485 kW)	.233 (3.818)	4205	4 (.91)	1000 (69)	4.6 (2.09)

Other types of positive displacement pumps, such as axial piston and sliding vane pumps, have been applied to engine main fuel pumps. These have not achieved the level of acceptance of external tooth gear pumps.

On highly specialized engines, such as military fighter aircraft engines, additional high-pressure fuel pump functions may be required. These include fuel pumps for supplying the thrust augmentation systems (afterburners) and, in some applications, fuel pressure for the actuation system for the engine's exhaust nozzle. Because augmentation fuel flows are very high and the systems are operational for only a small percentage of the mission time, high-speed centrifugal pumps are used in these situations. Centrifugal pumps offer a low unit weight and the ability to be run dry when the system is not operating, thereby conserving power. These centrifugal pumps operate at speeds up to 25,000 rpm and deliver up to 200 gpm (45.42 m^3/h at 1000 lb/in^2 (69 bar) pressure rise. The fuel actuation function has been provided by variable displacement piston pumps. The fuel boost function for the multiple high-pressure fuel pump engines is usually provided by a single centrifugal boost element either separately mounted on the gearbox or integrated with the main fuel pump.

An alternative drive means has been used for the centrifugal augmentor fuel pump in some applications. An air turbine utilizing engine compressor bleed has been used as a drive source. When the system is nonoperational, the pump may be stopped or idled by throttling the turbine inlet air supply, thereby conserving power.

For large engines, such as those used on commercial airlines, the boost stage is a conventional centrifugal element that generally incorporates an axial inducer. For smaller engines used on general aviation and helicopter applications, various specialized centrifugal elements and jet pumps have been used and have shown the capability of meeting the needs of suction feed fuel systems.

As discussed the engine boost pump must function with the airframe boost pump in both the operating and failed, or not operating, conditions. The engine fuel inlet pressure with the airframe boost pumps operational is generally specified over a range from 50 lb/in^2 (3.45 bar) to 5 lb/in^2 (.345 bar) plus TVP for all engine operating conditions. There are to be no vapor or air bubbles present in the fuel. For the condition of failed, or nonoperating, air frame boost pumps, the engine boost pump is required to operate with an inlet vapor-to-liquid volume ratio (V/L) of .45 at the maximum fuel tank temperature for takeoff conditions to at least a 10,000 ft (3048m) altitude (to cover high altitude airports) and appropriate V/L and fuel temperature conditions for maximum continuous thrust conditions up to 45,000 ft (13,716m). The V/L condition occurs because of the line pressure drop that occurs between the fuel tank and the engine boost pump inlet. This is depicted in Figure 15. The value of the V/L ratio is defined as the volume of vapor divided by the volume of fuel. A V/L of .45 represents a void volume ratio of approximately 30 percent.

Because the engine mounted boost pump is line-mounted and is external to the airframe fuel tank, the vapor entrained by the fuel cannot be ejected back to the tank as it is with the tank-mounted airframe pumps. All the vapor must be ingested by the boost pump and be compressed and redissolved back into the fuel. If the boost stage becomes overwhelmed by vapor, the vapor will build up at the pump inlet and the system will become vapor locked. This must be absolutely avoided.

The value of .45 for the takeoff V/L ratio has been established over the years as a value that will provide adequate vapor lock margin. See Reference 4 for a history of the V/L parameter. Airframe systems are analyzed to ensure the margin is maintained. The reduced usage of wide cut fuels in the future will somewhat relieve this requirement. Figure 16 presents the relationship between V/L and tank altitude. Note that for a constant value of V/L, as the tank altitude level increases, the pump inlet pressure, and therefore its NPSH, decreases.

In addition to the magnitude of the V/L ratio and the boost pump NPSH, the condition of the multiphase flow is important. To ensure continuous and stable operation of the system, the phases must be well mixed and have equal transport velocities. An operating limit is reached when the phases separate and transport velocities of the vapor phases are less than that of the liquid phase. Under these conditions, vapor will collect at a high point in the system. In time, the system will become unstable if some means of removing this collecting vapor is not provided. A rule of thumb for roughly estimating this transition point is a minimum velocity of 3 ft/sec (1 m/s) for the liquid phase occupying the full diameter of the pipe. For velocities below this level, the probability that phase separation will occur

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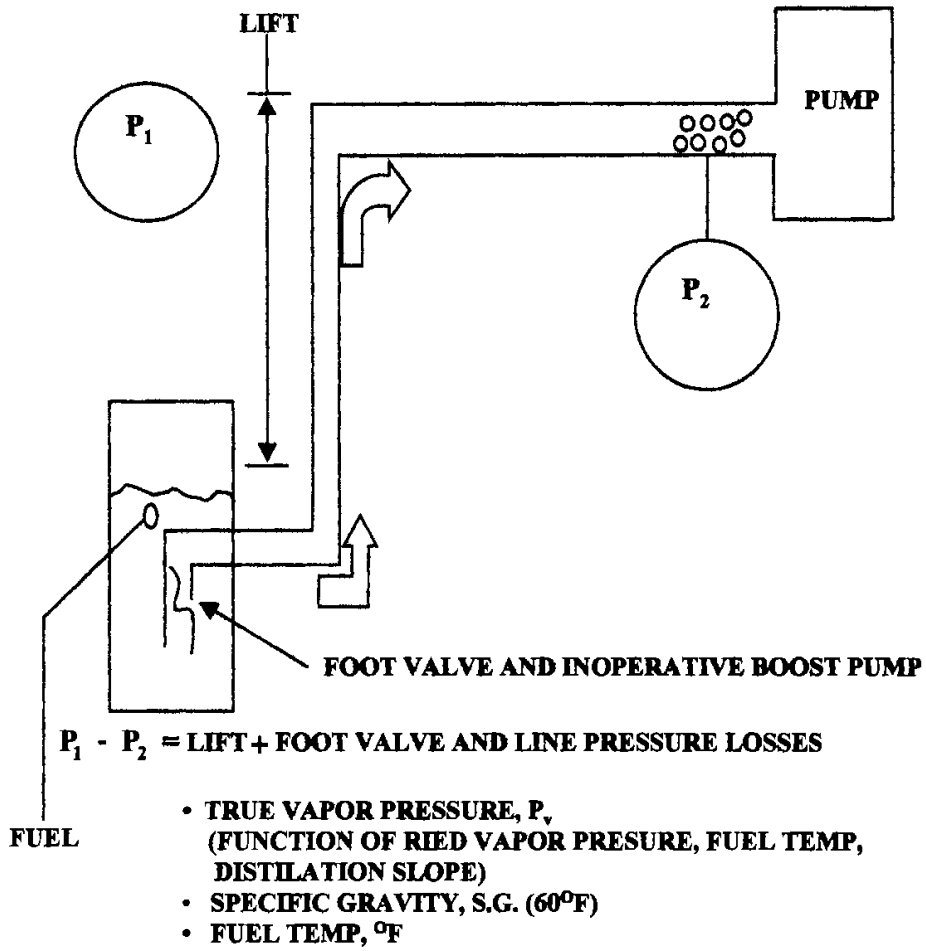


FIGURE 15 Factors that determine V/L ratio. $V/L = O \times (P_1 - P_2)/(P_2 - P_v)$, where the Ostwald coefficient O for typical fuels is in the range 0.1 to 0.2, depending upon these factors. Pressures P are absolute [°C = (°F − 32) × 0.556]

increases. In practice, an adequate mixing of the phases is maintained in pump tests that are run to prove the pump will meet the emergency conditions as previously defined.

For the suction-feed fuel systems employed on some helicopters, the uniform mixing of the vapor and liquid phases may not be maintained for all operating conditions. As previously discussed, these systems operate continuously under fuel-vapor-forming pump inlet conditions for all engine operating conditions; that is, from flight idle to maximum contingency power at all altitudes. Depending upon the engine fuel flow requirements, the fuel temperature and type and the fuel line configuration, separation of the liquid and vapor phases may occur. Because of the significant vertical pipe runs in these systems, this separated flow may exist in the form of alternating "slugs" of vapor and liquid flow in the line. When flow conditions of this type occur at the inlet to the engine fuel boost pump, a reserve volume of fuel must be provided within the pump. It is necessary to maintain a continuous flow of liquid fuel to the engine combustors for the time period when the vapor slug is being ingested by the pump and being compressed and redissolved into the fuel. Figure 17 is a pictorial presentation of the potential vapor and liquid flow conditions at the pump inlet. References 5 and 6 present a detailed review of these flow conditions.

DESCRIPTION OF THE DESIGN OF MAIN FUEL PUMPS

In this section, the design of a main fuel pump utilizing an external spur gear high-pressure element for application on engines powering modern airliners will be discussed.

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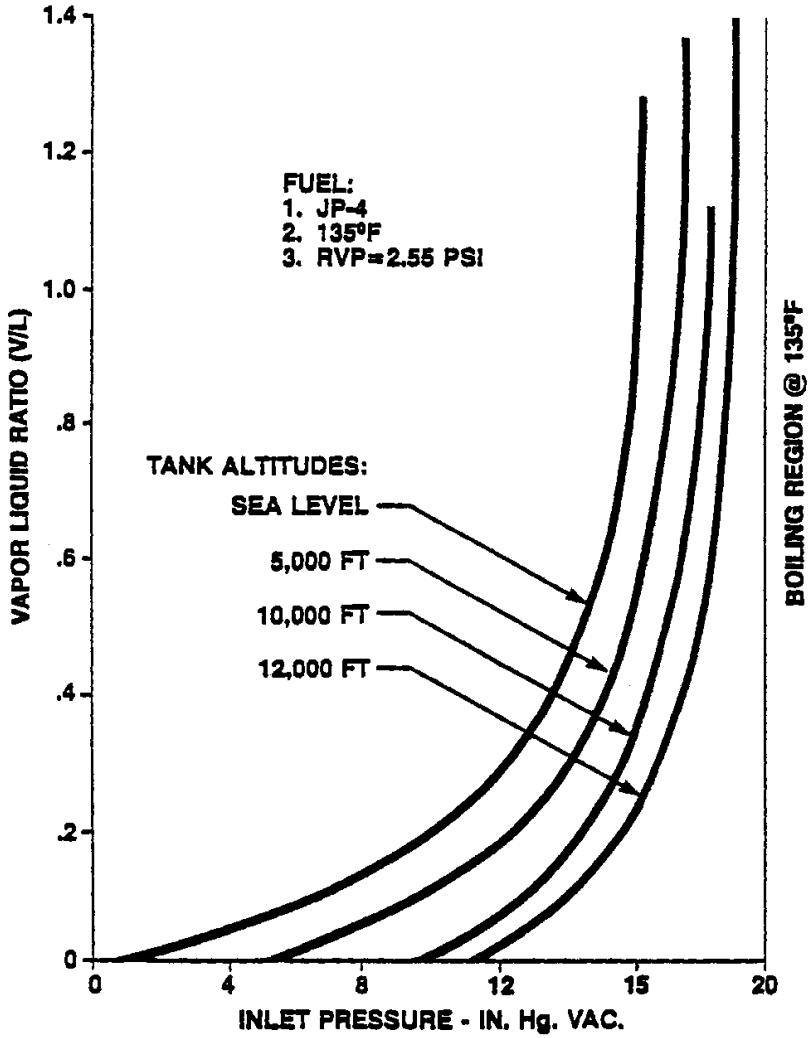
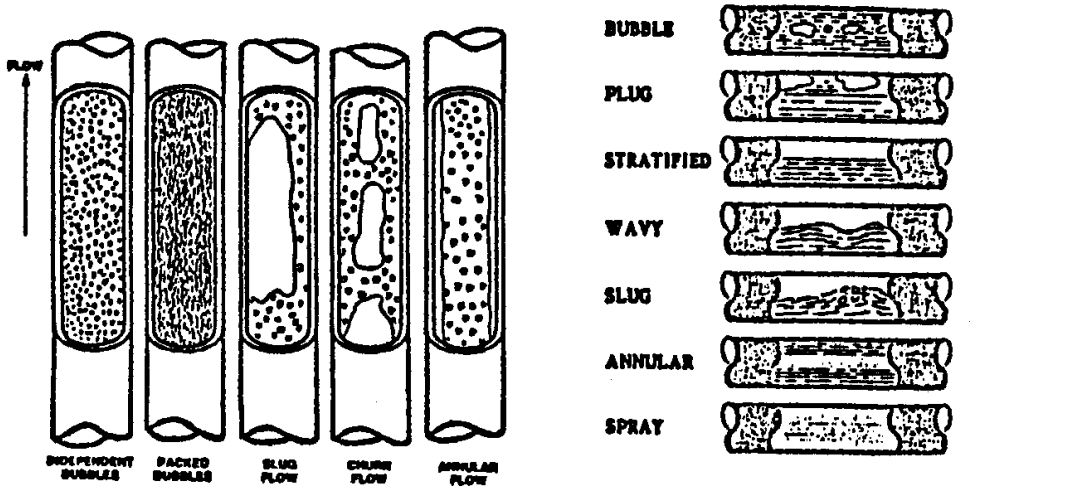


FIGURE 16 Vapor/liquid ratio versus inlet pressure (m = ft × 0.3048; bar = psi × 0.06895; °C = (°F − 32) × 0.556; mm Hg = in Hg × 25.4)



VERTICAL PIPE FLOW CONDITIONS **HORIZONTAL PIPE FLOW CONDITIONS**

FIGURE 17 Possible multiphase flow conditions at pump inlet

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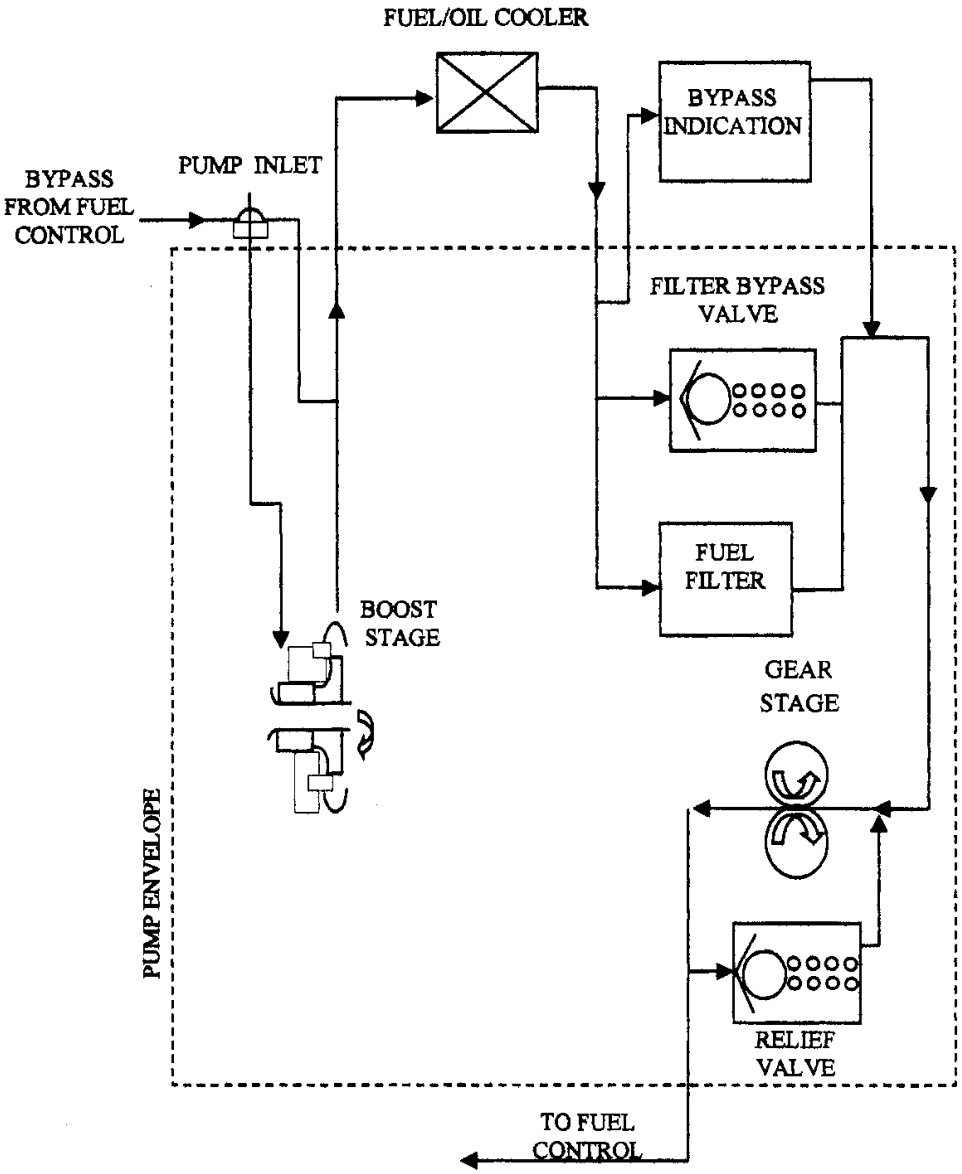


FIGURE 18 Main fuel pump system schematic

The designs of main fuel pumps for engines powering other classes of aircraft such as military fighters, executive aircraft, and helicopters are similar and differ mainly in the details of size and specific operating requirements. The flow and component schematic of a typical commercial engine main fuel pump is presented by Figure 18. The components encompassed by the dotted square are contained within the pump. As shown, the fuel from the airframe system enters the boost stage inlet. It then passes through the engine oil-to-fuel cooler where the fuel absorbs rejected engine heat, thereby fulfilling its function as a heat sink. This positioning of the oil-to-fuel cooler also provides the de-icing function for the next component in the system, the engine fuel filter. Using the engine oil-rejected heat for the de-icing function eliminates the need for fuel heaters using engine compressor bleed air as the heat source. These types of systems were used on previous generations of engines incurring weight and engine efficiency penalties.

Fuel filters are usually rated at 10 microns nominal and 40 microns absolute. The minimum filter surface area is usually determined by the worst icing condition. A filter bypass valve is provided to ensure continuous engine fuel flow in the event the filter becomes blocked. An indicator is provided to indicate impending filter bypass so maintenance actions can be initiated.

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The fuel then enters the high-pressure gear stage element and is delivered to the engine fuel control unit at a pressure determined by the downstream resistance characteristics. A high-pressure relief valve is utilized for burst protection for the high-pressure components of the fuel system in the event of a downstream blockage.

The control of the engine fuel flow is achieved in the fuel control system by bypassing the excess fuel back to the fuel pump upstream of the fuel oil cooler. This returned flow represents waste heat that is proportional to the quantity of the bypassed flow and the pressure differential between the gear stage and boost stage discharge pressures. This waste heat further increases the gear stage fuel inlet temperature.

The volumetric flow characteristics of positive displacement pumps closely match the fuel flow requirements of aircraft gas turbine engines. The ideal volumetric flow rate of a positive displacement pump is directly proportional to its rotational input speed. In actual practice, this is somewhat modified by the internal leakage flows that occur from the high-pressure discharge to the low-pressure inlet. These leakage flows are a combination of flow through the clearances between the pump parts and the necessary bearing lubrication and cooling flows. Also, leakage flows are proportional to the pressure rise across the pump and essentially constant over the operating speed range of the pump. Therefore, the leakage flows are a higher percentage of the ideal flow for lower percentages of rated pump speed. The percentage of the ideal flow that the delivered flow represents is the volumetric efficiency of the pump. Figure 19 depicts the basic relationship between the output characteristics of a positive displacement pump and the engine fuel flow requirements.

As previously discussed, the main fuel pump is required to provide both the engine burn flow requirements and the engine geometry actuation flow requirements. For a given class and type of engine (for example, commercial airline turbo fan engines), the engine geometry actuation flows are usually an essentially constant value for all engine speeds for a given number of actuator servos. Therefore, because of the volumetric characteristics of positive displacement pumps, the displacement sizing point of the high-pressure element for larger engines will tend to be the rated takeoff power high pump rotational speed condition. For smaller engines, it will tend to be the starting low pump rotational speed condition. The example in Figure 19 is sized at the starting condition. The disadvantage of this

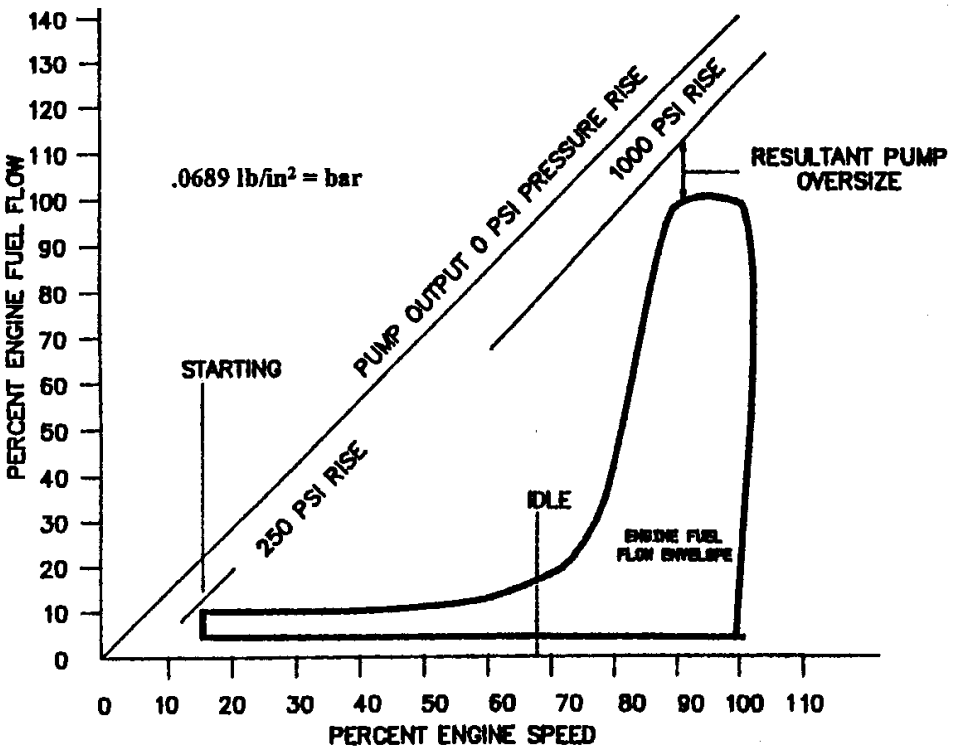


FIGURE 19 Positive displacement pump output characteristic versus engine flow requirements

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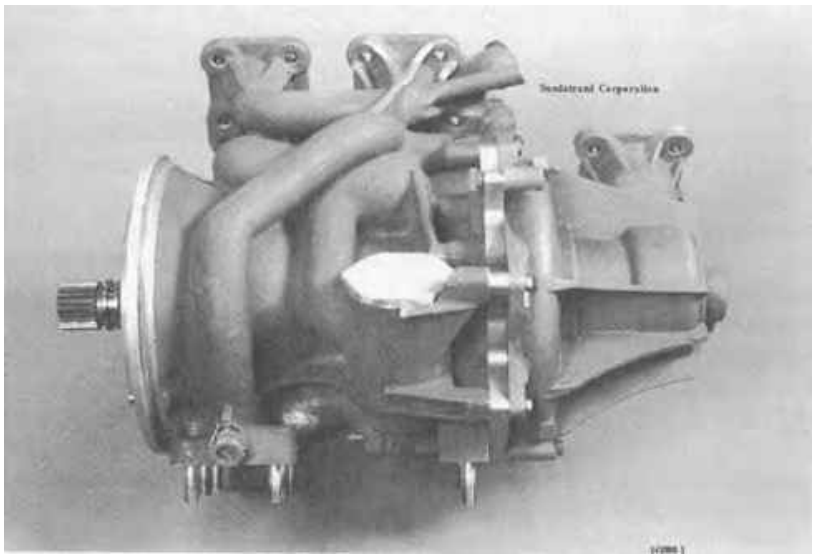


FIGURE 20 Main fuel pump assembly (Courtesy Hamilton Sundstrand)

type of pump concept, clearly shown in Figure 19, is the significant over-capacity of the high-pressure element for low engine power conditions such as the idle condition. This over-capacity represents the quantity of waste heat in the form of throttling loss in the bypass control loop that complicates the engine heat management system. The task is to avoid reaching the fuel thermal stability temperature limit (about 325°F/163°C) in the combustor fuel nozzles. Exceeding this limit will result in clogging the nozzles with fuel "coking" deposits. Significant effort has been expended on alleviating this problem through the application of variable displacement pumps. These efforts have not achieved general acceptance because of reliability, safety, and cost disadvantages.

Figures 20 and 21 present a photograph and a cross-section of a main fuel pump incorporating an external spur gear high-pressure stage and centrifugal boost stage. The pump is mounted directly to the engine gearbox and the rotational input power is transmitted directly to the spur gear stage by a spline coupling. The boost stage is driven by a secondary splined coupling. The drive priority is selected to reflect the power input order of

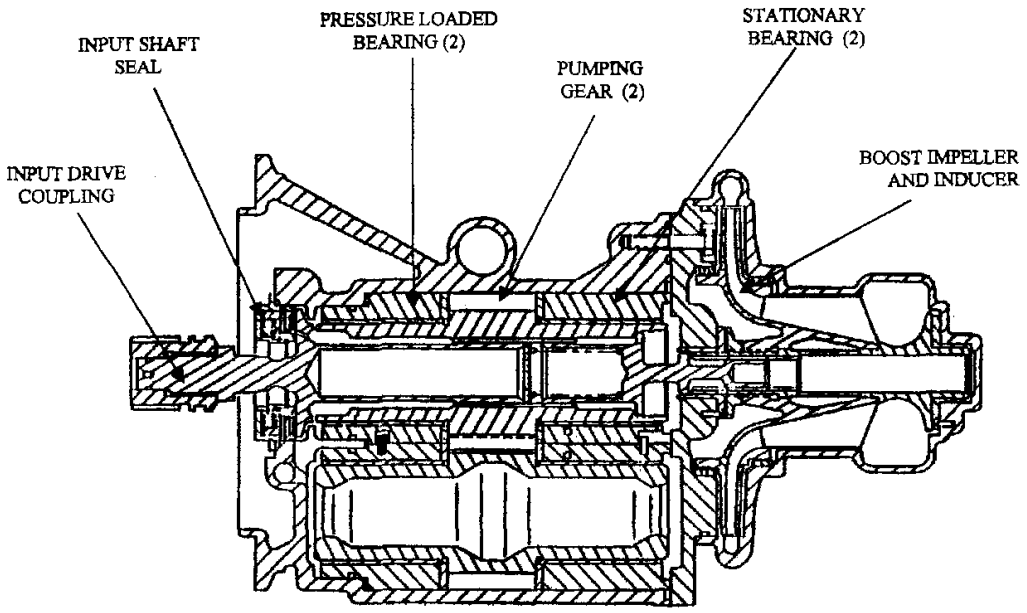


FIGURE 21 Main engine fuel pump cross section (Courtesy Hamilton Sundstrand)

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the elements and that the high-pressure element is the primary pumping element. The housings are aluminum castings that provide the minimum weight and also provide the necessary structural integrity and stiffness for all specified conditions.

A face-type dynamic mechanical shaft seal is provided at the drive end of the pump. A wide range of design configurations have been employed, but all modern pumps utilize the sealing faces as the primary seal and "O" rings for the secondary seal. These seals are required to operate with pressure differentials in both directions. Special design features must be employed to accommodate this requirement. The drive shaft seal is vented to a dry cavity that is connected to the engine overboard drain system. For reasons of safety, all fuel and oil passages separated by seals, static or dynamic, must have double seals with a drain port between them.

The type of external spur gear pump element used in main fuel pumps is the fully pressure-loaded type. Figure 22 depicts the basic details of the configuration used. The six parts shown in Figure 22 are assembled into a figure-eight bore arrangement in the pump housing. The four bearing blocks, which are a slip fit, are pressure-loaded toward the faces of the spur gears, and the assembly as a whole is pressure-loaded toward the inlet side of the housing bores. An initial axial sealing force is provided by springs. The axial pressure blow-off and pressure loading forces are closely balanced. This provides a pressure-balanced seal between the end faces of the spur gears and the bearings. This minimizes leakage over a wide range of delivery pressures on low viscosity jet fuels. The overall arrangement provides the necessary flexibility to maintain the minimum possible leakage clearances between the discharge and inlet over the complete range of operating conditions for many thousands of hours of operating life. The inherent flexibility of the design provides it with the unique capability of automatically compensating for the differing rates of thermal expansion of the materials used in the construction of the pump and the wear that will inevitably occur in usage.

The gears are manufactured from highly alloyed tool steels and surface hardened to resist wear in the harsh environment of low viscosity and low lubricity aircraft gas turbine fuels. The gear profile, lead error, and tooth spacing are held to very low limits. The roundness of the gear journals is also held to a very close limit and a mirror-like finish is applied. These journal dimensional characteristics are required because of the very low film thickness (6 to 30 micro inches or 0.15 to 0.76 microns) encountered. This is attributed to the low viscosity of hot aircraft gas turbine fuel (1 to .5 centipoise). The bearing surfaces, radial and thrust, are a highly leaded bearing bronze alloy generally with a solid film lubricant coating to assist in the initial "bedding" in.

The bearings are designed to provide full film lubrication for all operating conditions with the exception of the starting condition. On some pumps, a hybrid bearing design that incorporates a high-pressure pad to augment the bearings load carrying capacity is used. The bearing thrust faces contain gear trapping relief cuts that control the gear mesh flow dynamics. This includes avoiding any fluid trapping conditions, controlling gear stage inlet, and discharge pressure pulsations. Figure 23 is a photograph of a gear set with two of the bearings.

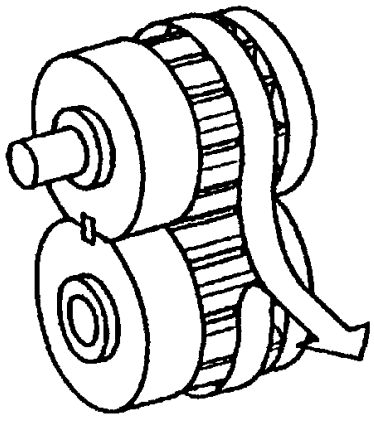


FIGURE 22 Pressure-loaded spur gear pump

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FIGURE 23 Spur gear pump gears and bearings (Courtesy Hamilton Sundstrand)

The design of the centrifugal boost stage is quite similar to that used for the airframe boost elements. The inlet must be sized to handle the total inlet flow rate consisting of the vapor phase and the liquid phase. The use of axial inducers has become common on all sizes of pumps because of the severity of the required inlet conditions. The selected overall pressure rise of the boost stage is primarily dependent upon the temperature and, therefore, vapor pressure of the fuel at the entrance to the high-pressure element and the various pressure losses between the boost stage discharge and the high-pressure element inlet. These pressure losses include filter pressure losses (clean filter and clogged filter), oil-to-fuel cooler pressure losses, and various coring and plumbing pressure losses. A pressure "stack" analysis between the boost stage inlet condition and the gear stage inlet pressure requirement must be made for all operating conditions across the pump input speed range—including the emergency conditions—to determine which operating condition is the critical sizing point for the pressure rise of the pump boost stage. A simplified example of a pump pressure stack is presented by Figure 24. Reference 7 discusses boost impeller sizing.

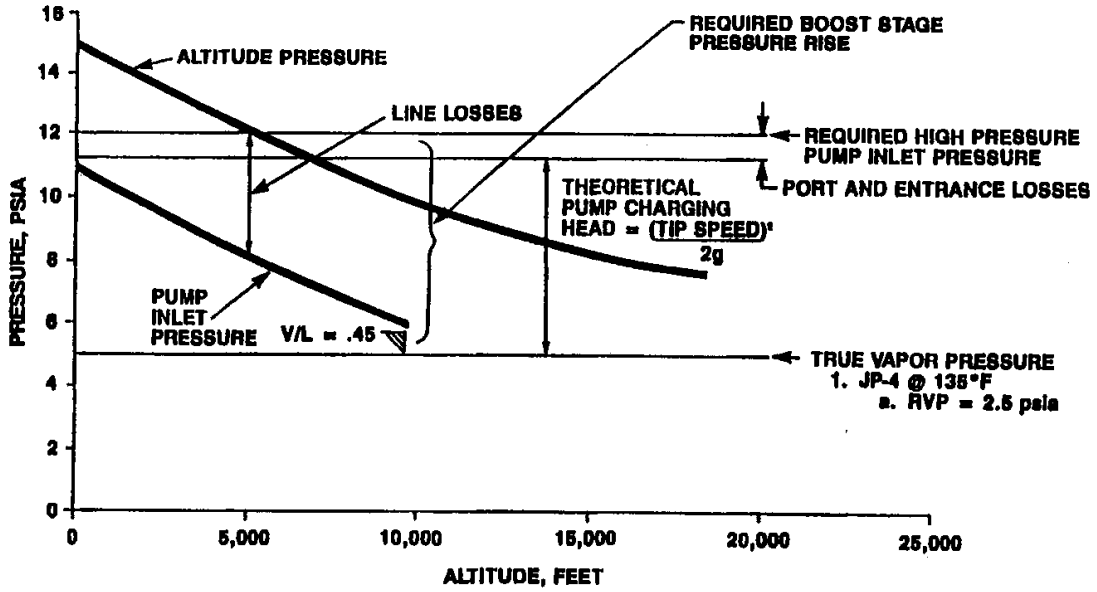


FIGURE 24 Typical pump inlet pressure stack (bar = 0.06895 × psi; °C = (°F − 32) × 0.556; m = ft × 0.3048)

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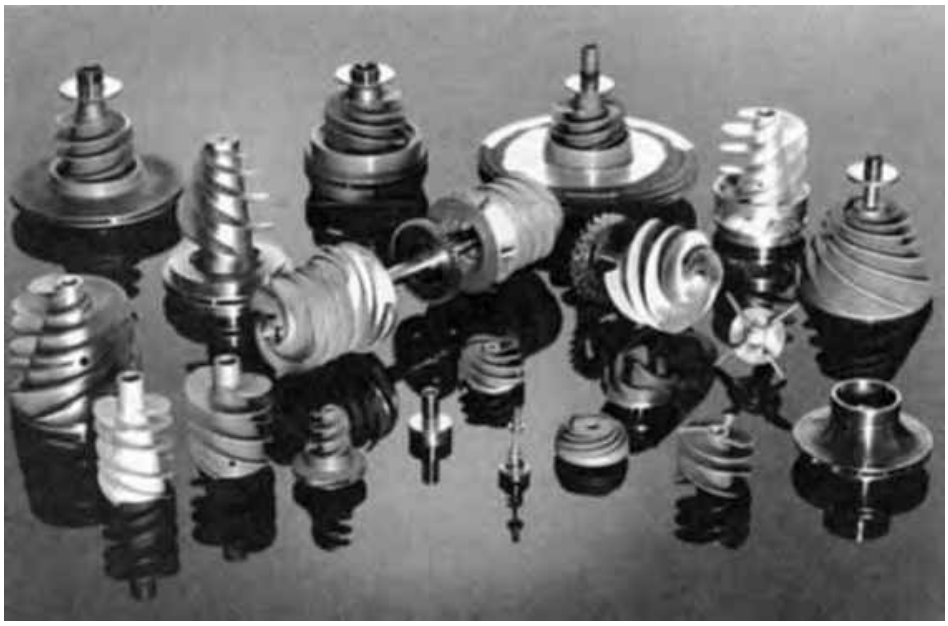


FIGURE 25 Various impellers and inducers (Courtesy Hamilton Sundstrand)

Figure 25 is a photograph of various impellers and inducers that have been used in airframe and engine fuel pumps. The specific speeds in units of feet of fluid, gpm, and rpm range from 500 to 6000 (universal specific speed Ω_s = 0.18 to 2.2) and suction-specific speeds in the same units range up to 40,000 (14.6).

With respect to boost stages, particular attention must be paid to minimize the conduction of heat from the hot high-pressure stage to the relatively cool boost stage. This is required to avoid the possibility of vapor lock through fuel "boiling" in low-pressure regions of the boost stage under conditions of low fuel burn flow rates and, therefore, low boost stage through flow rates. These conditions generally occur at engine idle and descent operating conditions.

The pump-splined drive couplings are key to the reliability and safety criteria of the pump. Similar to the pump gears, the splined couplings are fabricated from high alloy steels with a surface hardening treatment. The design of the involute splines in terms of profile wear must take into account the lubricant used for the splines and the misalignment imposed by the various drive line elements. A spline design that will satisfy the wear conditions at rated speed will generally meet all overload conditions, including the maximum shaft shear torque requirements. Spline lubricants that have been successfully used include engine oil, fuel, and specially blended greases. Specific spline design parameters must be applied for each lubricant.

PERFORMANCE CRITERIA OF ENGINE MAIN FUEL PUMPS

The key performance criterion for the high-pressure positive displacement stage of a main fuel pump is to be capable of delivering the volumetric fuel flow rate required by the engine for all operating conditions. Before it is released for flight usage, the test sequence with which the positive displacement stage must successfully comply is primarily directed at evaluating the durability of the stage under the most extreme conditions it is expected to encounter in service. There are success criteria for these tests. At the completion of each individual test, the stage must meet its specified volumetric flow service limits. Upon completion of all the tests, the unit is subjected to a teardown inspection and the component parts must not exhibit any unusual wear or distress and must be in a condition that is deemed acceptable for continued service.

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The key performance criterion of the boost stage is to provide adequate pressure to the inlet of the gear stage to suppress fuel vaporization and cavitation for all operating conditions; that is, with assistance from the airframe boost pumps; and the emergency conditions without assistance from the airframe boost pumps. The tests that confirm this capability are run in conjunction with the positive displacement stage and are subject to similar success criteria. The performance criteria that specifically determine the design of the boost and high-pressure main fuel pump are presented in the following discussions.

GEAR STAGE VOLUMETRIC PERFORMANCE

The engine fuel flow requirements are specified in mass units. These must be converted to volumetric flow units for the lowest density fuel specified for all operating conditions. For these volumetric flow rates, the displacement required for each operating condition is determined by applying factors for pump volumetric efficiency that encompass the temperature of the fuel, production variance, and the service life flow deterioration. When the critical displacement sizing operating condition has been established, the overall volumetric performance of the pump can be defined for the complete input speed and pressure rise operating range including the required input power. Figure 26 presents the overall performance characteristic of a pressure-loaded external spur gear high-pressure stage of an aircraft engine main fuel pump.

GEAR STAGE CYCLIC DURABILITY

The ability of the pump to accept the cyclic duty imposed by modern engines without distress has been an accurate predictor of the pump's capability of meeting its service life requirement. The test is based upon the real-time pressure, temperature, and speed transients for engine operating conditions such as starting, ground idle, takeoff and climb, cruise, descent, and thrust reverse.

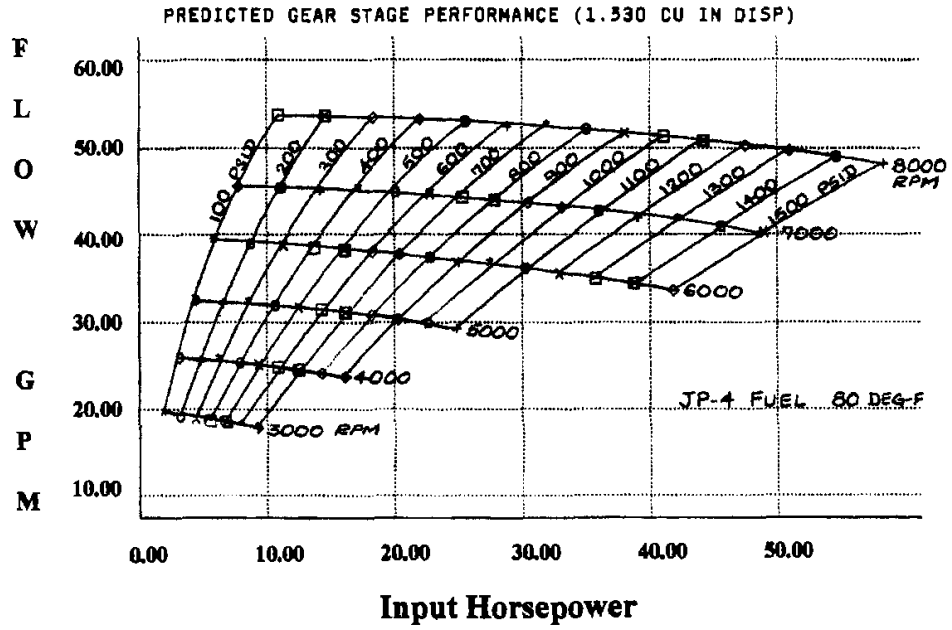


FIGURE 26 Gear stage performance ($m^3/h = 0.227 \times gpm$; $kW = 0.7457 \times hp$; $bar = 0.06895 \times psi$; $liter = 0.0164 \times in^3$; $°C = (°F - 32) \times 0.556$)

The worst cases for the sequencing of the transients are selected to achieve the highest levels of stress in the gear and bearing system. The real transient times are halved and the steady state operating times minimized to accelerate the test. The number of test cycles and overall test time are based upon the expected service life of the pump.

LOW LUBRICITY FUEL

The use of the pumped fuel as the lubricant for the pump and all components within the fuel system is an established and required practice. Unfortunately, there is no requirement for lubricity in the fuel specifications. Therefore, some knowledge of the minimum lubricity to be expected in the field is necessary. The key factor to determine is the lubricity of the fuel under boundary lubricating conditions for evaluating the wear characteristics of the gear tooth profiles and the fuel lubricated spline teeth. An intensive cooperative industry and government evaluation effort has identified the Exxon ball and cylinder wear test machine as an accurate and reliable method of evaluating the lubricity characteristic of aircraft gas turbine fuels. The pump specification will contain the fuel lubricity level with which the pump will be required to demonstrate operation for its intended service. The pump designer must select gear materials and geometries that are compatible with the specified lubricity level and operating conditions. Wear experienced on low-lubricity fuel is a threshold type of surface failure. Therefore, this threshold limit must be avoided for all operating conditions. The test for compliance with this requirement is usually operation at takeoff and cruise conditions on the specified fluid at controlled lubricity conditions. The fluid lubricity level is checked periodically to ensure compliance is met. See Reference 8 for a detailed description of the test procedure.

COLD STARTS

Cold start tests demonstrate the capability of the pump to provide the performance necessary to start the engine under the most severe cold conditions expected and to demonstrate that the fits and clearances in the pump are compatible with the low temperatures. The cold start requirement ranges from −40°F (−40°C) to −65°F (−53.9°C), depending upon the fuel used and service expected.

CONTAMINATED FUEL

The large amount of fuel an engine burns in operation and the widely varying service conditions worldwide virtually ensure that some contaminants will be introduced into the engine's fuel system. In addition to these operational contaminants, there will be built-in contaminants because of the complexity of the airframe and engine fuel system that will be experienced in the initial operation of the pumps. Tests are specified that define operation of the pump at various operating conditions on fuel containing both liquid and solid contaminants. These included salt water, quartz crystals, sand, and iron oxide ranging in particle sizes from 1500 microns to less than 5 microns. These tests are more severe for military applications than commercial applications. Military applications usually require the contamination to be metered into the inlet of the pump in relation to the fuel flow rate and then removed from the system downstream of the pump by a filtration system. Hence, the contaminant is passed through the pump in a continuous single pass manner. In all modern fuel systems, the high-pressure positive displacement stage is protected by the engine-mounted fuel filter element. The low-pressure boost stage is not protected by the engine-mounted filter element. Although the centrifugal boost stage is inherently capable of operating on the specified contaminant, special design features are employed to minimize the abrasive wear effects of swirling contaminated fuel.

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V/L CAPABILITY

The reason for the need of the boost stage of an engine main fuel pump to have a V/L capability and some detail of the design requirements has been presented in previous discussions. The test set-up used for V/L testing is similar to the example shown in Figure 15. The testing is usually accomplished in two parts. Both parts are run with the maximum fuel RVP expected in service. The first part is to evaluate the V/L performance of the pump at all of the specified emergency conditions and prove all conditions can be met. The second part is an endurance test at high V/L conditions to ensure that no excessive cavitation erosion damage occurs that could limit the performance or life of the pump.

MINIMUM INLET PRESSURE

As long as the main fuel pump is being assisted by the airframe boost pumps, it must meet a minimum inlet pressure requirement. This inlet pressure is 5 lb/in^2 (0.345 bar) above the true vapor pressure (TVP) of the fuel. At this condition, the main fuel pump is required to supply the required engine fuel flow for all operating conditions. The design procedure and details for meeting this requirement were previously discussed.

The test set-up is arranged to ensure that vapor-free fuel is provided to the inlet of the fuel pump at 5 lb/in^2 (.345 bar) above TVP for all operating conditions. Heat exchangers are introduced into the system as required to establish the required boost and gear stage inlet temperatures. The full range of hot operating conditions, including the critical altitude idle conditions, are run to ensure vapor lock conditions do not occur. The maximum fuel RVP expected in service is used.

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12.18.2
LIQUID ROCKET PROPELLANT PUMPS

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Liquid-fueled rocket engines have been used on all the major launch vehicles, including the large, first-stage booster rockets as well as the upper stages of those same vehicles, and various smaller vehicles such as the lunar lander. The propellant pumps are major components of the engine. They pump a) the oxidizer, usually liquid oxygen, and b) the fuel, which is usually liquid hydrogen, up to the high pressures needed to feed the combustion processes. In the past, a widely used fuel was kerosene in a rocket propellant formulation specified as "RP1."

Of necessity, the pumps for these engines must have the minimum possible size and weight. They therefore run at extremely high speeds. Most have inducers or are fed by inducer pumps. These inducers have extremely high suction-specific-speed capability, N_{ss} exceeding 35,400 ($\Omega_{ss} = 13$) and being as high as 70,000 ($\Omega_{ss} = 26$) in liquid hydrogen[1]. This is so because the hydrogen is nearer its thermodynamic critical point and so generates less vapor volume when it cavitates than do other propellants. (See the discussion under "NPSH-Effects" in Section 2.1.1. All symbols in this section are defined in the nomenclature of that Section 2.1.1.) The main engine propellant pumps have higher heads per stage than any other centrifugal pumps in existence. As such, they are the world's highest-energy pumps, as indicated in Figure 32 and Table 13 of Section 2.1.1. Whereas the high-energy pump portion of that section is devoted largely to the challenges of designing such machines for long life—say 40,000 hours or more—the life of a rocket engine pump is measured in minutes; or, in the case of reusable rockets, not much more than a few hours. For good materials choices, such a short operating life means that these pumps are unlikely to suffer failure from cavitation erosion or other wear-related phenomena, despite the high inlet tip speeds of the inducers and impellers. Extreme attention to details of the mechanical design is required in order for these pumps to survive at design conditions—near which they generally operate. A few examples of rocket propellant pumps are given in this section, but many others also exist in various configurations[1,2].

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THE SATURN V BOOSTER ROCKET ENGINES

The Saturn V booster rocket was used in the Apollo program of the 1960s and '70s, which landed men on the moon. This rocket had a vertical height of more than 350 ft (197 m) at launch. The first or lowest stage was the largest and was propelled by five F-1 engines, each producing 1.5 million pounds (6.7 million N) of thrust. Propellants were liquid oxygen and RP1. Each engine was topped by a turbopump assembly that included both pumps on the same shaft and driven by the same hot-gas turbine. A cross-section of this assembly is shown in Figure 1, the two pumps being arranged in a back-to-back configuration and each having an inducer. The RP1 pump is the one next to the turbine, there being a considerably lower temperature difference between RP1 and the hot gas flowing through the turbine from the combustor than there would be for liquid oxygen. At the design speed of 5,490 rpm, the two pumps together consumed 52,700 hp (39 MW)—and considerably more at overspeed (approximately 6000 rpm). Table 13 of Section 2.1.1 contains performance figures. Essentially the same data are presented in the following descriptions of the pumps, which are summarized in Table 1. The table has been developed from various sources, including the references cited at the end of this section, as well as information supplied by the Boeing Company.

- *The oxygen pump* took in the liquid axially through a 15.75-in (400-mm) diameter inducer at the opposite end of the assembly from the turbine, as seen in Figure 1. Designed for a suction-specific speed N_{ss} of 35,400 (Ω_{ss} = 13) meant that when ingesting 25,080 gpm (1.58 m^3/s) this machine could operate at an inlet static pressure of 22 lb/in^2 (0.15 MPa) above the vapor pressure p_v of the liquid oxygen. Because this liquid is cryogenic, p_v equals the pressure inside the fuel tank of the first stage of the rocket, which was not much above atmospheric pressure. Actually, the g-force and head of liquid in the tankage above the pump combined to provide additional available NPSH. This pump, which was designed for a specific speed N_s of about 2,100 (Ω_s = 0.77), consumed

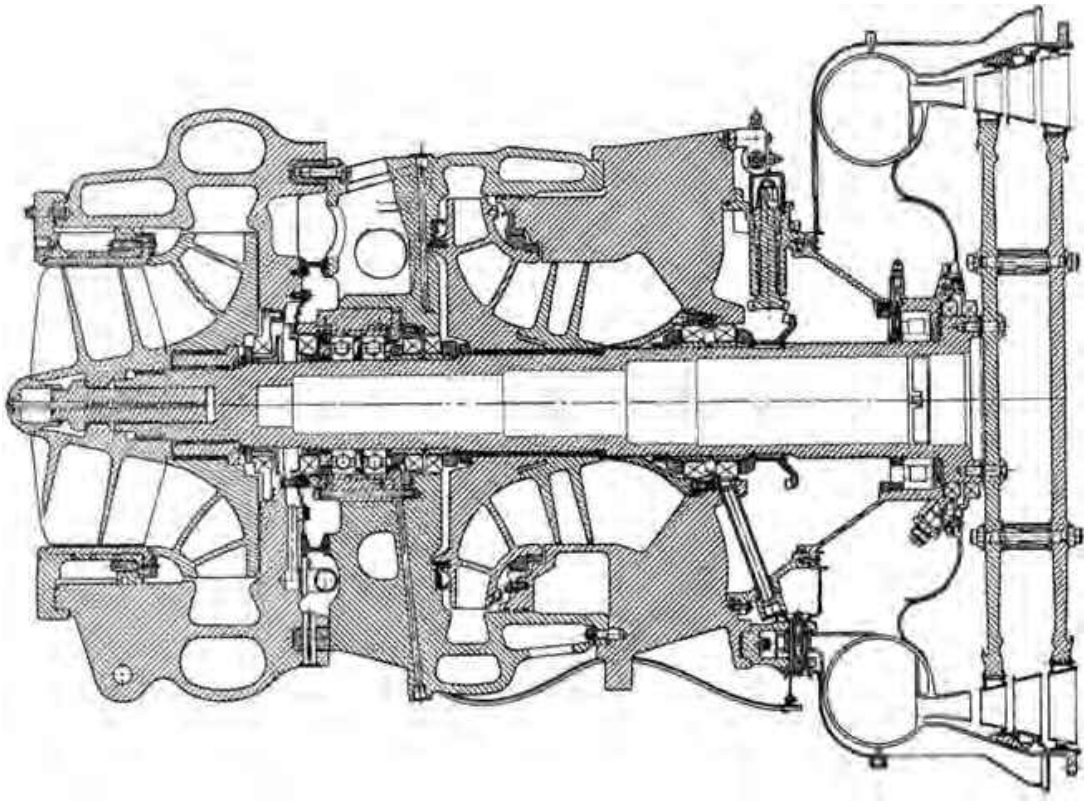


FIGURE 1 F-1 Turbopump assembly, used on Saturn V first stage rocket engine (Courtesy of The Boeing Company)

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TABLE 1 Data on liquid rocket propellant pumps*

Parameter	Units	F1 Engine		Space Shuttle Main Engine			
		Oxygen pump	RP1 pump	Oxygen pumps		Hydrogen pumps	
				LPOTP	HPOTP	LPFTP	HPFTP
Shaft power	hp (MW)	30,000 (22.4)	22,700 (19.9)	1,740 (1.3)	27,770 (20.7)	2,900 (2.2)	77,000 (57.4)
Rotative speed	rpm	5,490	5,490	5,450	31,100	15,700	37,400
Mass flow rate	lb/s (kg/s)	3,977 (1,804)	1,762 (799)	964 (437)	1,148 521	161 (73)	161 (73)
Inlet volume flow rate	gpm (m^3/s)	25,080 (1.58)	15,640 (0.99)	6,080 (0.38)	7,240 (0.46)	16,300 (1.03)	16,300 (1.03)
Pressure rise	Lb/in^2 (MPa)	1,530 (10.5)	1,810 (12.5)	332 (2.3)	4800** (33.1)	222 (1.5)	6,840 (47.2)
Head rise (approx.)	ft (m)	3,100 (945)	5,100 (1,555)	670 (204)	9,700 (2,960)	7,240 (2,200)	200,000 (61,000)
Specific speed (stage)	rpm,gpm,ft (universal)	2,095 (0.77)	1,130 (0.41)	3,230 (1.18)	2,700 (0.99)	2,550 (0.93)	1,150 (0.42)
Inducer diameter	in (mm)	15.75 (400)	16.00 (406)	11.725 (298)	4.7 (119)	12.014 (305)	
Impeller exit diameter	in (mm)	19.50 (495)	22.95 (583)		6.80 (173)		12.00 (305)
Overall diameter x length	in x in (mm x mm)	44 x 60 (1,118 x 1,524)		18 x 18 (457 x 457)	24 x 36 (610 x 914)	18 x 24 (457 x 610)	22 x 44 (559 x 1,118)
Turbopump weight	lb (kg)	3,080 1,397		276 (125)	622 (282)	177 (80)	775 (352)

*Compiled from information supplied by The Boeing Company and References 1–5.

**Main stage. Small preburner stage pumps 117 lb/sec (53 kg/s) to approximately 3300 psi (23 MPa) above main stage discharge.

30,000 hp (22 MW) or 57% of the total turbine shaft power, and it generated over 1,500 lb/in^2 (10 MPa) of pressure rise.

- *The fuel (RP1) pump*, being in the middle of the turbopump assembly, had to take in the 15,640 gpm (0.99 m^3/s) of RP1 through a side inlet piping configuration, which generally results in less N_{ss}-capability than axial inlet piping. Here the inducer is larger and the flow rate lower than for the liquid oxygen pump, creating the potential for higher N_{ss}; yet it ended up being lower, namely $N_{ss} = 34{,}200$ ($\Omega_{ss} = 12.5$). Nonetheless, this enabled the RP1 pump to operate at a static inlet pressure as low as 16 lb/in^2 (0.11 MPa) above the (negligible) vapor pressure of this liquid. With specific speed N_s just over 1,100 (0.40), this pump consumed the remaining turbine shaft power of 22,700 hp (17 MW) and generated a pressure rise of just over 1,800 lb/in^2 (12.4 MPa).
- Throughout the burn, these pumps deviated no more than about 8% from the design value of Q/N; that is, the flow coefficient was essentially constant. Thus, this turbopump did not

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suffer from the off-design low-flow conditions of most commercial, industrial pumps, as described in Section 2.1.1 including those of high energy level, which made it a little easier for this machine to operate at high energy levels.

THE SPACE SHUTTLE MAIN ENGINES

The space shuttle program, which began in the 1970s, dealt with a launch vehicle that was flanked by two large solid-fueled rockets. In the middle at the base were three liquid-fueled "main" engines, each having nearly 500,000 lb (2.2 million N) thrust and propelled by hydrogen fuel and oxygen at a mixture mass ratio of 6:1 (oxygen to hydrogen). Each of these space shuttle main engines (SSME's) has a dedicated pumping system, consisting of a low-pressure single-stage inducer-type pump for each propellant, which in turn feeds a high pressure pump. Three of these four pumps can be seen in the photograph of the engine in Figure 2. Located up above is the low-pressure hydrogen pump, with its large discharge line going off to the right and down to the high-pressure hydrogen pump below. Opposite, on the left, is the high-pressure oxygen pump, with its large inlet line also in view. Engineering of these high-pressure propellant pumps embodied a triple challenge in comparison to the F-1 pumps, because a) this engine was designed to be re-used, so the pumps had to have a life of 7.5 hours with 100 starts; b) the pumps for this engine have higher energy levels than the F-1 pumps, and, c) due to the mission profile, they have to be throttled back further from the design point. Each of these pumps is boosted by a low-pressure inducer-type pump to suppress cavitation enough to maintain performance[3].

The propellant flow system is illustrated in the schematic diagram of Figure 3. (The pump speeds, pressures, and flow rates shown in this figure are somewhat lower than the design conditions of Table 1, which for the SSME pertains to the maximum engine thrust level.) All four of the pumps are driven by turbines and so are called "turbopumps." The system picks itself up by its own bootstraps, so to speak; each low-pressure turbopump boosts the flow to the corresponding high-pressure pump and is driven by the same fluid that it pumps. This driving fluid comes back from each high-pressure pump, the low-pressure oxygen turbopump (LPOTP) being driven by recirculated liquid oxygen, and the low-pressure hydrogen or fuel turbopump (LPFTP) turbine being fed by gaseous hydrogen heated by the thrust chamber, as indicated in Figure 3.

The high-pressure pumps are driven by turbines fed by "preburners," which are combustors that burn hydrogen-rich. Some of the fuel entering these combustors is the gaseous hydrogen coming from the LPFTP turbine exhaust, which cools the turbine housings on the way to the preburners. But most of the fuel supplied to the preburners is the 80% of the liquid hydrogen discharging from the high-pressure fuel turbopump (HPFTP), which first flows through the cooling passages of the nozzle walls. [Eleven percent of the oxygen is also fed to the preburners by way of the preburner boost stage that is a part of the high-pressure oxygen turbopump (HPOTP) package. Finally, the partially burned fuel passes as a hot gas into the main combustion chamber, where more oxygen is added and the pressure is 3,000 lb/in^2 absolute (21 MPa)[3,4].] Most of the remaining 20% of the hydrogen is that which was already described as cooling the main combustion chamber and, along the way, becomes gaseous and powers the drive turbine of the LPFTP. It also cools the hot-gas manifold and injector and pressurizes (in a small amount) the fuel tank. Approximately 75% of the liquid from the high-pressure oxygen turbopump (HPOTP) goes directly to the main combustion chamber, 11% to the preburners (as already stated), about 13% to drive the turbine of the low-pressure oxygen turbopump (LPOTP), and a small amount is sent to pressurize the tank.[5] A brief description of each pump follows:

- *The low-pressure oxygen turbopump (LPOTP)* consumes 1,740 hp (1.30 MW) and runs at 5,450 rpm. It has a single-stage, axial-flow, inducer-type impeller that is 11.725 in (298 mm) in diameter and is driven by a six-stage liquid-oxygen hydraulic turbine. Pump head rise is 670 ft (204 m) so the HPOTP therefore operates without pressure

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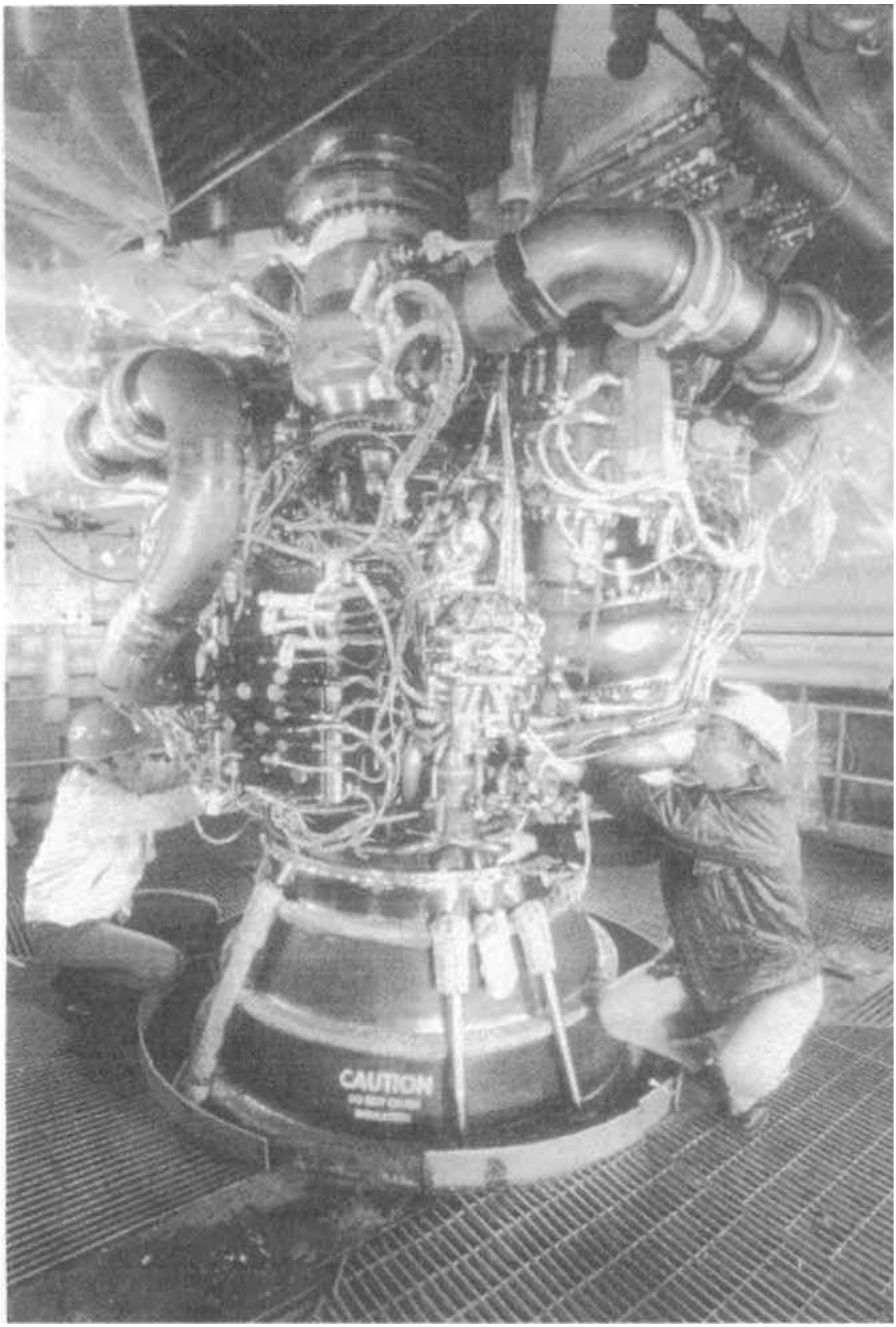


FIGURE 2 The space shuttle main engine (SSME) (National Geographic Magazine, March 1981: Jon Schneeberger/NGS Image Collection)

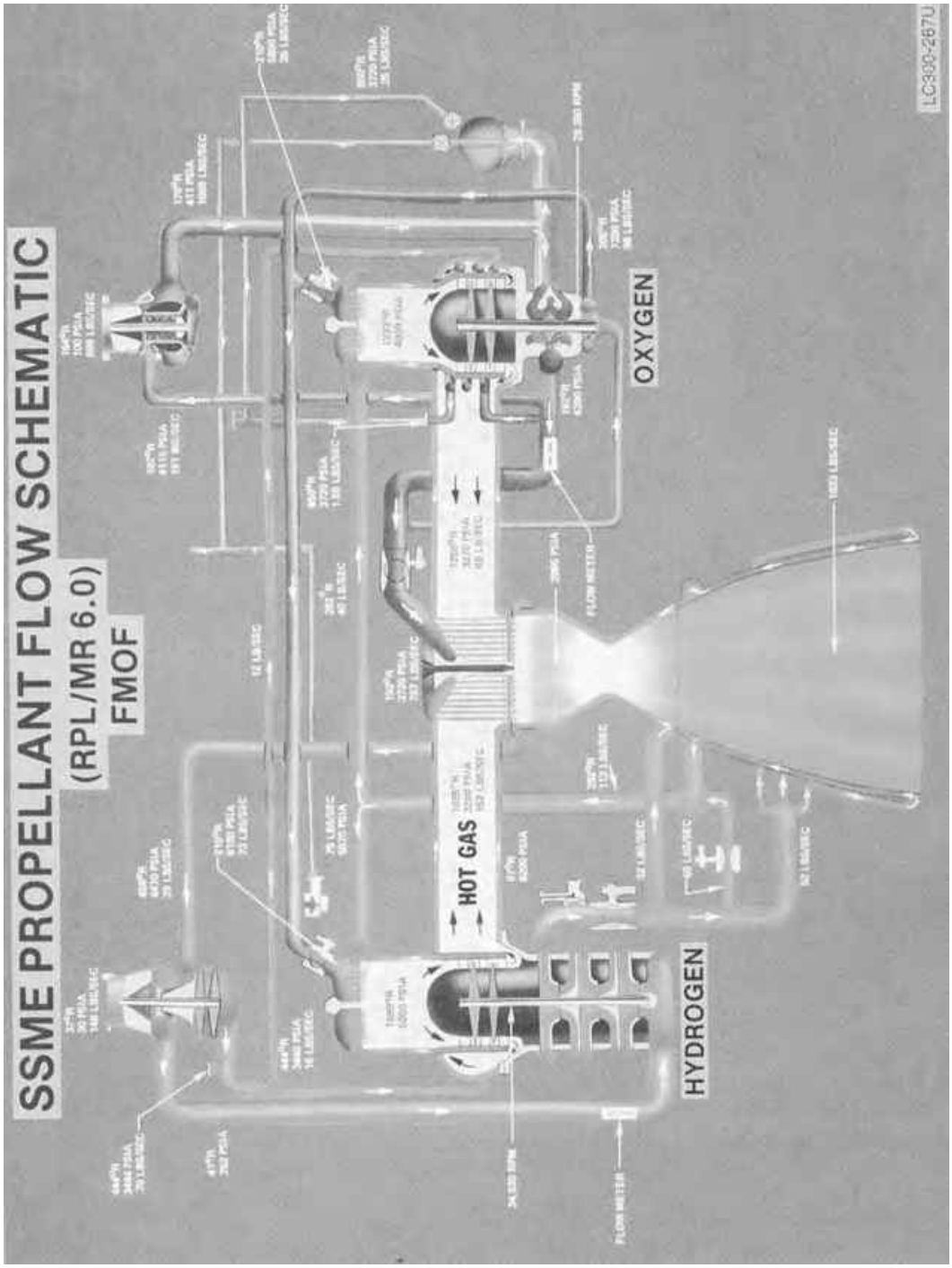


FIGURE 3 SSME liquid oxygen and hydrogen pumping arrangement (Courtesy of The Boeing company)

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loss due to cavitation at all engine conditions. Specific speed N_s = 3,230 (1.18), which is low for an inducer, but this one performs a more substantial pumping task than do inducers running at high speeds on the same shaft as the high-pressure centrifugal stages they feed. The shaft has two ball bearings cooled by liquid oxygen, and the envelope of the package is 18 in (457 mm) in diameter × 18 in (457 mm) long.

- *The low-pressure fuel turbopump* (LPFTP) is similar in concept to the LPOTP. It is a 2,900 hp (2.2 MW) machine that runs at 15,700 rpm. The inducer-type impeller is 12.014 in (305 mm) in diameter and is driven by a two-stage gaseous-hydrogen turbine. It ingests liquid hydrogen to the extent of $\frac{1}{6}$ of the mass flow rate of oxygen. Due to the very low specific gravity of liquid hydrogen (= 0.0708 at atmospheric pressure), the head rise is more than 7,000 ft (2,100 m). N_s = 2,550 (Ω_s = 0.93). The shaft has three ball bearings cooled by liquid hydrogen, and three seals limit leakage between the pump and the turbine before engine start and during operation.
- Both low-pressure pumps exhibited suction-specific speeds N_{ss} in excess of 35,000 (Ω_{ss} = 13)[4]. They are shown schematically at the top of Figure 3, and the high-pressure pumps are below them.
- *The high-pressure oxygen turbopump* (HPOTP) is shown in cross-section in Figure 4 together with the associated preburner unit. Inducers are shown in the detailed cross-sectional drawing of Figure 5 feeding the double-suction main stage that generates nearly 5,000 lb/in² (34 MPa)—making this one of the world's highest-energy pumps (as defined in Section 2.1.1, Figure 32). And more than 3,000 lb/in² (21 MPa) in addition to this pressure is generated in the small booster impeller (on the outer end of the turbopump as seen in Figures 4 and 5) that feeds about 11% of the main flow to the preburners for driving both high-pressure pumps. Both shrouds of the main impeller act as orificed balancing disks to keep all but a small preload off the two angular-contact ball bearings, which are cooled by liquid oxygen. Radial loads are minimized by the use of a vaned diffuser. The compact design and high speed (over 31,000 rpm) of this 28,000 hp (21 MW) machine result in operation between the first and second critical speeds. Several seals are utilized,

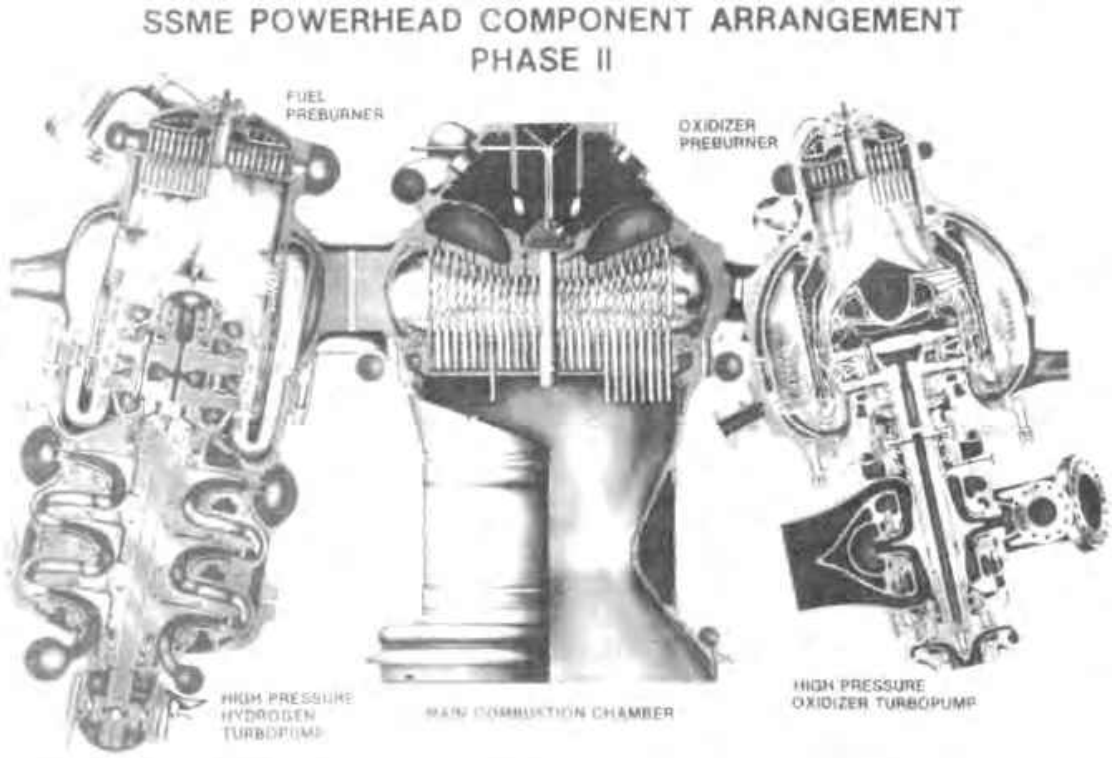


FIGURE 4 SSME high-pressure turbopumps (Courtesy of The Boeing company)

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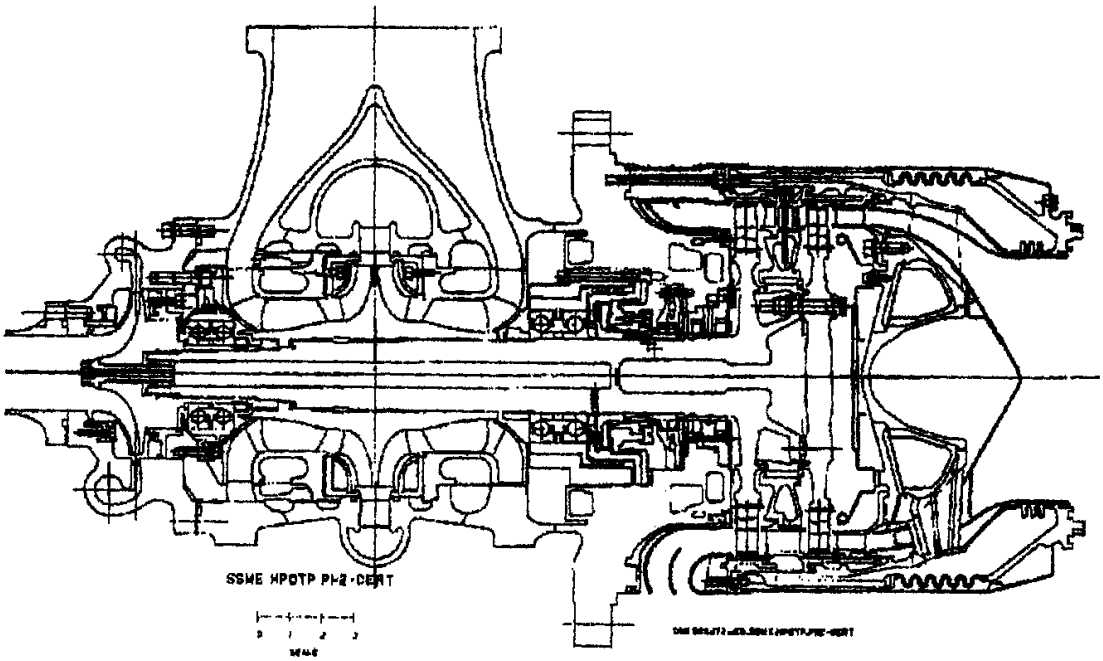


FIGURE 5 Cross-section of SSME high-pressure oxygen turbopump (Courtesy of The Boeing company)

including a helium-purged seal that isolates the hot gas of the turbine from the cold liquid oxygen. Special material combinations are used to prevent sparking, ignition and explosion due to rubbing in the strong oxidizing environment of the liquid oxygen.

- *The high-pressure fuel turbopump* (HPFTP) is also shown in Figure 4. Additional detail is afforded by the cross-sectional drawing of Figure 6. No inducer is needed due to the pressurization of the LPFTP and the more favorable vaporization characteristics of liquid hydrogen. The pump has three identical impellers and diffusers, except that the third-stage diffuser discharges into a scroll or volute, and the third-stage impeller also

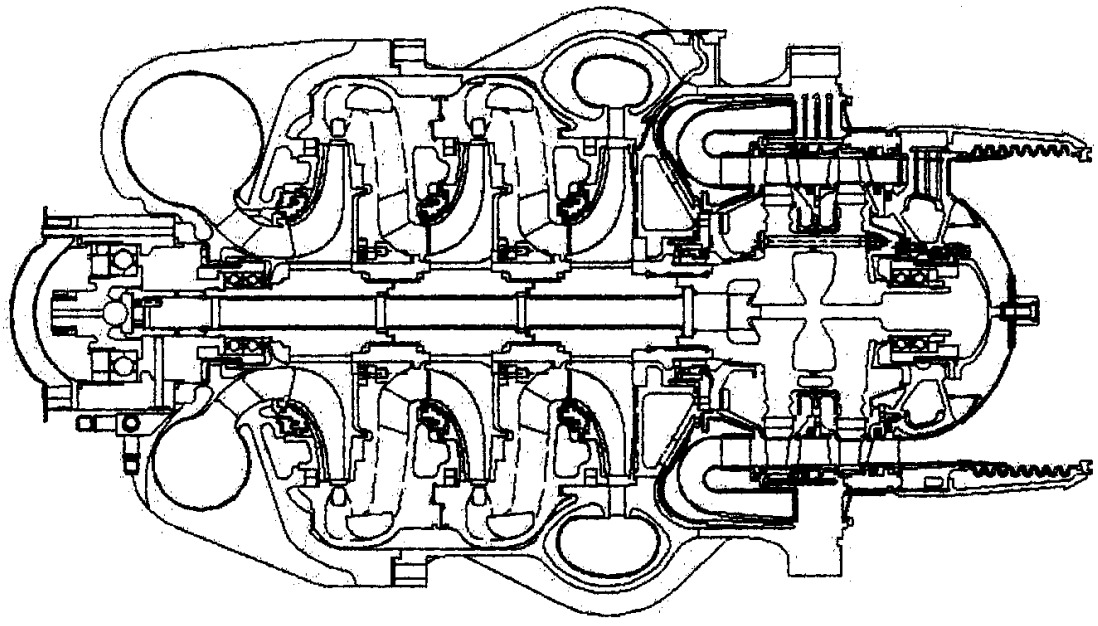


FIGURE 6 Cross-section of SSME high-pressure oxygen turbopump (Courtesy of The Boeing company)

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acts as an orificed balancing disk, which removes all axial thrust (except for a small spring-preload force) from the angular-contact, liquid hydrogen-cooled ball bearings located at each end of the pump-and-turbine package. The head per stage is over 65,000 ft (20,000 m) because of the high pressure rise and low density of liquid hydrogen. This leads to an impeller OD tip speed of nearly 2,000 ft/sec (600 m/s)—probably the highest in existence for a pump and successfully deployed in titanium, which has the benefit of a higher strength-to-weight ratio than the more commonly used steels. The high speed (over 37,000 rpm) keeps this 77,000 hp (57 MW) machine small and light and necessitates operation between the second and third critical speed.

THE RD-170 ENGINE

The RD-170 rocket engine was developed in the Soviet Union and is described in some detail by Sutton[6]. Propellants are liquid oxygen and an RP1 equivalent. Consisting of four nozzles, each of which produces about the same thrust as one SSME, this "four-engines-in-one" configuration is fed by a single propellant pump assembly—similar in concept to the F-1 turbopump. Referring to Table 13 of Section 2.1.1, the oxygen pump produces a pressure rise of about 8,500 lb/in^2 (59 MPa), which is undoubtedly the highest pressure rise of any centrifugal pump stage in the world, making it the highest-energy pump of Figure 32 in Section 2.1.1. Not far behind is the RP1 pump at 7,100 lb/in^2 (49 MPa). Assuming the pump efficiency to be 80% would yield a total shaft power for both pumps of about 230,000 hp (172 MW). At such high energy levels, as discussed in Section 2.1.1, the life of this pump must indeed be quite short. However, the design appears to have been successful. The only drawback is the environmental issue concerning the burning of a large quantity of hydrocarbon fuel, which is also an inhibition to further use of the F-1 engine.

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C • H • A • P • T • E • R • 13

INTAKES AND SUCTION PIPING

SECTION 13.1
INTAKES, SUCTION PIPING, AND STRAINERS

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The suction approach, or inlet, is the most critical part of a system involving pumps, whether in the form of piping or open pit. Improper design of the suction intake or piping can influence pump performance through efficiency and head loss, vibration, and cavitation. A centrifugal pump that lacks proper pressure or flow patterns at its inlet will not respond properly or perform to its maximum capability. The negative impact of improper flow at the pump inlet generally increases with the specific speed of the pump.

Flow control and the uniformity of flow to the point of pumped fluid contact with the impeller inlet vanes are the most important. Part of this may be controlled by proper pump design, but changes in flow direction, obstacles and changes in the geometry in the suction pit or suction piping are often main causes of improper flow quality at the pump inlet. In this respect the pit designer and suction piping designer have also definite responsibilities to achieve satisfactory pump operation.

In open suction pit (wet-well) designs, the fluid flow must be as uniform as possible right up to contact with the pump suction bell or suction pipe, preferably without a change in direction or velocity. Examples of dry-pit and wet-pit centrifugal pumps connected to open suction pits are shown in Figure 1.

In dry-pit pumping, the pipe leading to the pump suction flange should not include elbows close to the pump in any plane. Also, any other fittings that change flow direction and velocity and which may impart a spinning affect to the flow should be avoided within five to eight pipe diameters of the pump suction flange. Centrifugal pumps not designed for prerotation of the fluid entering the impeller, either dry or wet pit, may suffer loss of efficiency and an increase in noise if a spinning inlet flow occurs. Fluid rotation with the direction of impeller rotation can result in a decrease in pump-developed head. Fluid rotation against the direction of impeller rotation can result in an increase in pump-developed head and required power, possibly overloading the driver as well as drastically affecting the pump curve shape and performance in the system. If the total system is to operate efficiently and with minimum maintenance, close attention to the suction environment of the pumps is required.

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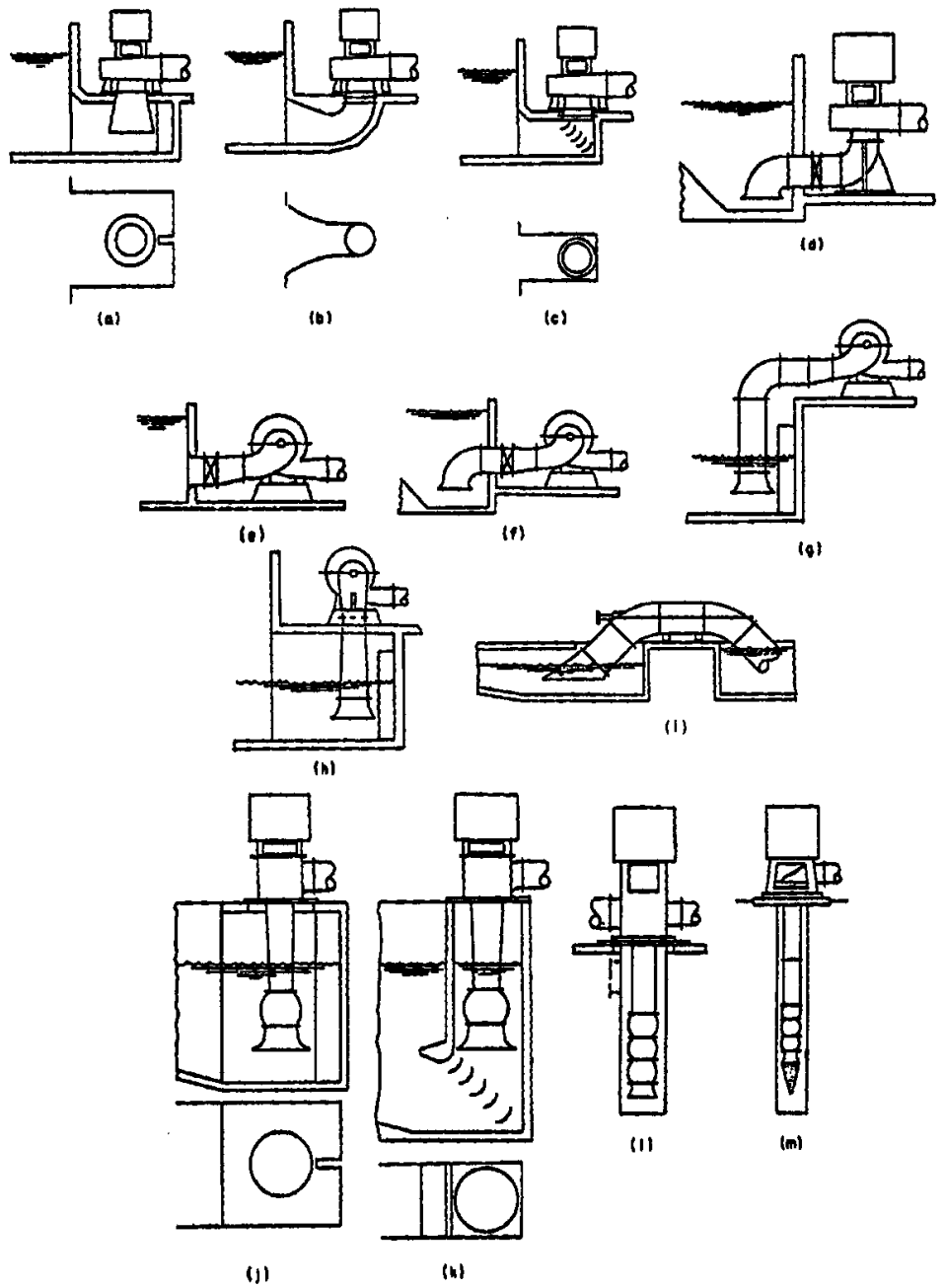


FIGURE 1 Pump suction connections: (a–d) vertical dry-pit pumps. (a) Suction bell diameter and distance from bottom, side, and back walls designed to provide radial inflow, equal velocity distribution, and minimum entrance loss. Proper spacing of bell and vertical baffle required to prevent underwater eddies and rotation at inlet. (b) To minimize excavation under pump, decreasing-type suction elbow, or formed suction intake can be formed in the foundation. (c) Width of suction inlet under pump minimized by use of turning vanes. (d) Smaller pumps provided with integral suction elbows and can be connected to wet well with commercial piping. Flared suction elbow is turned downward to obtain maximum submergence at inlet. Wet well sloped to suction inlet to prevent deposition of solids. (e–i) Horizontal dry-pit pumps. (e) Pump suction connected to larger pipe with eccentric reducer (double-suction pumps only).Wet-well connection should be bell mouth of a size to provide minimum required velocity. (f) Flared suction elbow turned downward for maximum submergence over inlet. Wet-well floor may be sloped toward inlet if solids are present. (g) Vertical suction pipe with bell inlet adequately submerged to prevent vortexing. Bell sized and spaced as in Figures 1a, h, and j. (h) Vertical suction pipe connected to bottom-suction pump. Bell sized and shaped as in Figures 1a, g, and j. (i) Design used for drainage over levees. Suction pipe designed to require minimum submergence and excavation. (j–m) Vertical wet-pit pumps. (j) Distance from bottom, side, and back walls designed to provide equal velocity distribution, with splitter to prevent vortexing. (k) Width of inlet minimized by use of turning vanes. (l) Canned suction pump with either above- or below-ground suction connection. Can length to suit NPSH required. Vertical installation minimizes excavation area. (m) Deep-well pump takes suction from underground water source.

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INTAKE STRUCTURES

Intake structures are used when the liquid to the pump comes directly from fluid bodies with free liquid surfaces. Their function is to provide a good flow quality, in terms of pressure and flow patterns, to the suction bell inlet of the pump.

The quality of the flow at the impeller inlet determines to which extend the maximum performance of the pump can be obtained. Improper design influences pump performance through efficiency and head loss, vibrations and/or the occurrence of cavitation. These effects can be caused by phenomena such as formation of subvortices, nonuniform velocity profiles at impeller inlet, and recirculation-and flow separation zones in the suction intake. In order to guarantee smooth pumping operation standard performance criteria have been defined to which a suction intake has to comply.

The inlet part of the impeller vanes are normally designed assuming a uniform velocity field that is free of rotation. In a well-designed suction intake the fluid enters the impeller inlet with variations of the velocity, that is, the flow distribution, less than 5% with respect to the averaged velocity. In this case also the time fluctuations of the velocities have to be limited.

The presence of swirl at any point in the flow just before the impeller inlet indicates that the fluid velocity consists of an axial component u and a tangential component u_t. Swirl angle α is defined as the angle between the resultant velocity and the axial direction. The amount of swirl at the impeller inlet should be limited, with a required swirl angle less than 4 degrees. The swirl angle can be measured with a neutrally pitched four bladed swirl- or vortimeter (Figure 6, Section 13.2). The allowable maximum number of revolutions per minute of this meter should be less than 1.7 Q/D_1^3 (see Reference 1, Hydraulic Institute Standards, Section 9.8.5.5).

Free-surface vortices (Figure 5, Section 13.2) are caused by improper intake design and/or insufficient submergence. Together with subsurface vortices, which may emanate from the floor, the ceiling and the back and side walls, they are only allowed when of low magnitude (see Section 13.2).

Insufficient submergence can also lead to failure at start up (when the impeller inlet is not filled with liquid), inadmissible amount of cavitation or the entrainment of air or gas bubbles in the pump. A pump may have adequate submergence from a pressure standpoint and still be lacking in sufficient depth of cover above the suction inlet to prevent surface air from being drawn in. Any wet-pit pump must have its suction inlet submerged at all times, and for continuous pumping every pump will have a fixed minimum submergence requirement. Because this relates to velocity, there are two basic parameters: the suction inlet diameter and the depth of water above the inlet lip. As pump size (and flow) increase, the inlet velocity may stay constant as the bell diameter increases, but at the same time, the impeller distance above the suction inlet becomes larger, so a fixed submergence value would lead to increased surface velocity, peripheral drawdown, and an increase in air intake.

Basically, submergence must of necessity increase with pump size. For the final determination, some balance must be struck between submergence and pit width to satisfy an average flow velocity of 1 to 1.25 ft/s (0.3 to 0.4 m/s) and maintain reasonable economic balance between excavation costs, concrete costs, and screen costs without neglecting ecological requirements and still fulfilling the primary need for circulating water in adequate quantities.

In order to obtain a maximum flow rate for a given difference in static head while reducing the available NPSH as little as possible, the head losses in the suction intake have to be confined to a minimum. The loss coefficient C_L of the suction intake is defined as:

$$C_L = h_L \cdot 2g/u^2$$

giving the relation between the head loss h_L and the dynamic head at the impeller entrance. Values of the loss coefficient of the suction intake alone (without trash racks and/or screens), may be expected to have values from 0.05 to 0.2.

The nonuniform velocity profile and/or swirl at the suction intake's outlet, and/or a high resistance factor of the (formed) suction intake can be caused from high velocities,

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improper flow guidance and/or high local decelerations of the fluid. These last two phenomena cause flow separation and the occurrence of secondary flows and/or dead corners.

Types of Intake Intake structures can be categorized as being for clear liquids or solids-bearing liquids. For clear liquids, intakes are further classified into rectangular, formed, circular, and trench-types, as well as suction tanks and cans. For solids-bearing liquids, trench-type and circular wet wells are usually considered. The different kind of intake structures will be briefly illustrated here; they are covered in detail in Reference 1.

RECTANGULAR INTAKE Probably the most common type of intake structure for vertical wet-pit pumps (or for dry-pit pumps having vertical suction pipe with a bell-mouth entrance) is the rectangular-shaped intake. For the design of such an intake, the recommendations found Reference 1 should be followed. The basic layout of the pumps and intake structure are shown in Figures 2 and 3.

The geometry is generally defined in terms of the pump inlet bell diameter, as shown. Once the number and size of pumps required is determined, a pump inlet bell diameter can be estimated. At this point, the bell diameter can be derived from the impeller inlet diameter or estimated based on an impeller axial inlet velocity of between 3 and 8 ft/s (0.9 and 2.4 m/s). The resulting impeller inlet diameter can then be converted to a corresponding bell diameter, approximately 1.5 to 2.0 times the impeller inlet diameter. With the bell diameter selected, the proportions of the inlet structure can be estimated from Figures 2 and 3. Table 1 gives recommended values for the dimensions. The minimum submergence S over the suction bell diameter D for vertical wet-pit pumps is a function of the Froude number F_D. Hydraulic Institute recommends using Hecker's formula:

$$S/D = 1.0 + 2.3\,F_D,$$

where F_D = Froude number = $V/(gD)^{0.5}$

D = suction bell diameter, ft (m)

S = submergence, ft (m)

V = suction bell inlet fitting velocity, ft/s (m/s)

g = acceleration due to gravity, ft/s^2 (m/s^2)

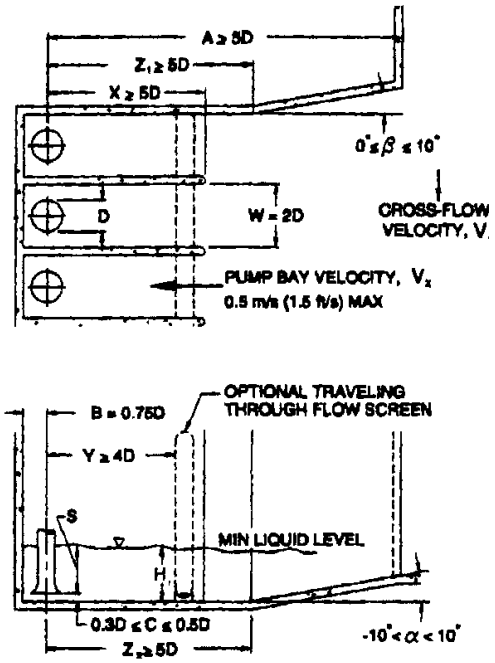


FIGURE 2 Recommended intake structure layout (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

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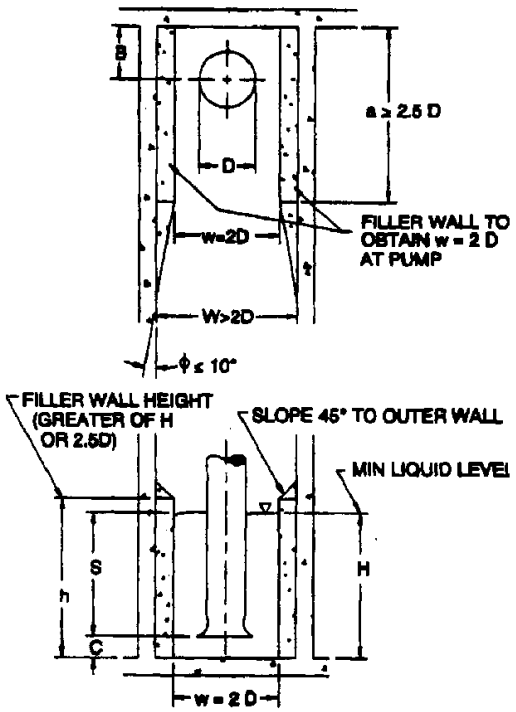


FIGURE 3 Filler wall details for proper bay width (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

For establishing velocities, the minimum submergence over the suction bell for vertical wet-pit pumps can be estimated from Figure 4, based on maximum expected flow rate. The graph is valid for suction pits that follow the standard dimensions indicated in Figure 2. Bottom clearance, back wall clearance, pit width, presence of antirotation baffles, and the velocity profile at the inlet of the suction intake are of critical importance in determining minimum required submergence.

The intake design process is intended to have an intake cross-sectional area average velocity of 1.5 ft/s (0.5 m/s) or less. It further assumed that the fluid approaching the intake with cross-flow velocities not exceeding 50% of the pump bay entrance velocity.

Any necessary change in inlet channel dimensions should be made gradually. Tapering walls should not diverge at more than a 14° included angle. If it is necessary to slope the floor, a maximum of 7° is recommended and, if possible, the floor should level off before reaching the pump area, as far back as possible. No sharp drops (waterfall effect) should be permitted.

If the inlet channel is a closed pipe with full-wetted perimeter, pipe velocities can be used up to a distance from the pump chamber where the tapered wall rule can be applied. This means that the pipe size would be increased in accordance with the rules for maximum taper rate to ensure that the velocity as it discharges into the pump chamber is not over 1.5 ft/s (0.46 m/s). Unless the inlet velocity into the pump chamber is kept below 1.5 ft/s (0.46 m/s), extensive baffling and additional space will be necessary, and this can only be effectively determined by analyzing the design modification with CFD and subsequently model testing of the pit.

TRENCH-TYPE INTAKES An effective way of decreasing the fluid velocity of a concentrated flow through a channel or pipe before the pump inlet is obtained with a trench-type intake. The trench-type intakes use a sudden transition in the vertical plane to guard the pump intake from the flow characteristics of the concentrated flow (when these changes would be in the horizontal plane often stilling chambers are used). The vertical transitions can be either in the wet-well intake structure or in front of the inlet of a suction intake. The maximum velocity in the approaching is limited depending on the direction of this flow relative to the direction of the well: 4 ft/s (1.2 m/s) when coaxial and 2 ft/s (0.6 m/s) when

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TABLE 1 Recommended dimensions for Figures 2 and 3 (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

Dimension Variable	Description	Recommended Value
A	Distance from the pump inlet bell centerline to the intake structure entrance	A = 5D minimum, assuming no significant cross-flow[a] at the entrance of the intake structure
a	Length of constricted bay section near the pump inlet	a = 2.5D minimum
B	Distance from the back wall to the pump inlet bell centerline	B = 0.75D
C	Distance between the inlet bell and floor	C = 0.3D to 0.5 D
D	Inlet bell design outside diameter	(see text)
H	Minimum liquid depth	H = S + C
h	Minimum height of constricted bay section near the pump inlet bell	h = (greater of H or 2.5D)
S	Minimum pump inlet bell submergence	$S = D(1.0 + 2.3 F_D)$
W	Pump inlet bay entrance width	W = 2D minimum
w	Constricted bay width near the pump inlet bell	w = 2D
X	Pump inlet bay length	X = 5D minimum, assuming no significant cross-flow at the entrance to the intake structure
Y	Distance from pump inlet bell centerline to the through-flow traveling screen	Y = 4D minimum. Dual-flow screens require a model study
Z_1	Distance from pump inlet bell centerline to diverging walls	Z_1 = 5D minimum, assuming no significant cross-flow[a] at the entrance to the intake structure
Z_2	Distance from inlet bell centerline to sloping floor	Z_2 = 5D minimum
α	Angle of floor scope	$\alpha = -10$ to $+10$ degrees
β	Angle of wall convergence	β = 0 to +10 degrees (Negative values of β, if used, require flow distribution devices developed through a physical model study)
ϕ	Angle of convergence from constricted area to bay walls	ϕ = 10 degree maximum

[a]Cross-flow is considered significant when $V_C > 0.5\ V_X$ average

normal to each other. The width of the sump above the trench should be designed so that the velocity of the fluid above the trench has a maximum of 1 ft/s (0.3 m/s). The height of the trench should be twice or more than twice the bell diameters above the suction bell or twice the height of the suction intake to which it is connected.

For larger capacities of the individual pump (more than 40,000 gpm (2.5 m^3/s)) total intake structure (more than 100,000 gpm (6.3 m^3/s)) it is advised to conduct a model test of the intake structure.

FORMED SUCTION INTAKE For large circulation pumps it can be advantageous to use a dry-pit pump (or even wet-pit pump) with formed suction intake instead of a wet-pit pump with intake structure. From a fluid dynamics point of view the formed suction intake has certain advantages over the open pit intake. In cases where high demands are required to the pump material, as for instance with tropical seawater, it is often cheaper to switch

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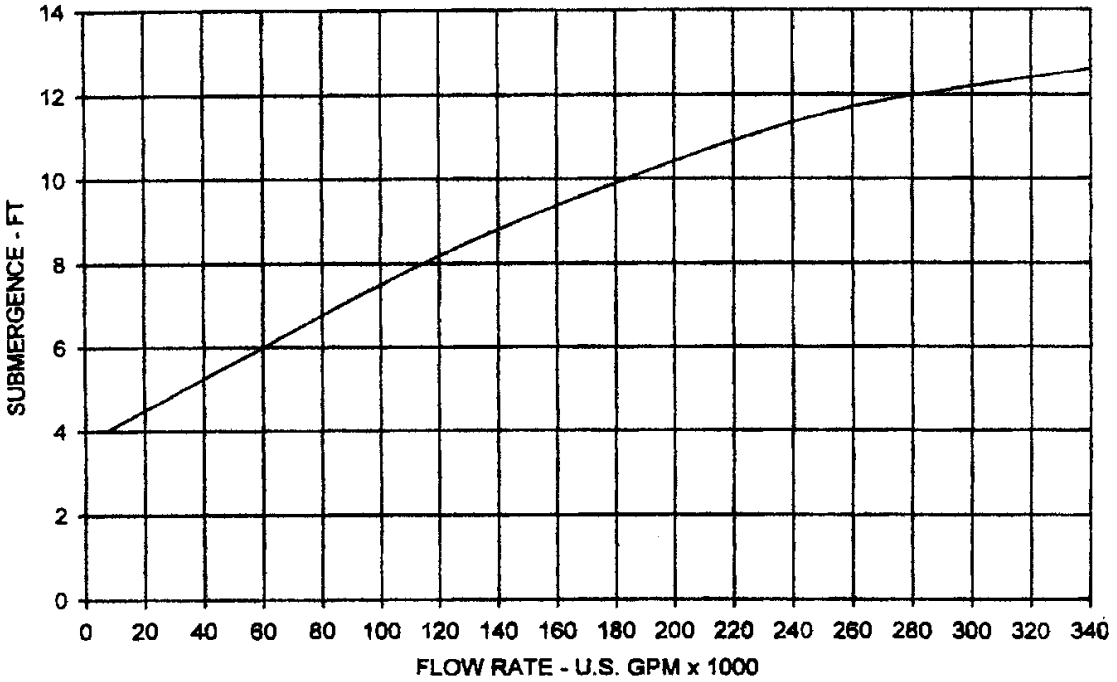


FIGURE 4 Flow rate versus submergence over suction bell for vertical wet-pit pumps (U.S. gpm × 0.227 = m^3/h, ft × 0.305 = m).

to dry-pit pumps instead of wet-pit pumps. The advantage is than that part of the high alloy metal in wet-pit pumps than be replaced by cheaper concrete (in the discharge piping and/or volute). Depending on the situation and demands of the pumping station, a choice will be made between an open-pit suction intake and a formed suction intake.

With respect to their function the formed suction intake has the advantage of the absence of dead water areas in which sand or silt can deposit. Further its performance is less sensitive to the direction and distribution of the flow at its inlet and it will not produce subsurface vortices. While during operation the minimum submergence required is often below the entrance level of the impeller, to get the pump priming at start-up, a water level that fills the inlet of the impeller is sufficient.

The most famous formed suction intake design is the TYPE 10 design developed by the U.S. Army Corps of Engineers of which the geometry is accurately described in Reference 1. Many of the major pump companies use their own developed formed suction intake designs. With the help of CFD it is nowadays very well possible to optimize the formed suction intake designs for certain design parameters, as there are minimum width, minimum length, minimum excavation, simplicity of shape, minimum hydraulic losses, or minimum submergence, without having penalties on their performance. As for all suction intake designs it is recommended to validate the performance of the designs by model testing.

An extreme case of optimization of a formed suction intake is described for a pumping station where the pumps and formed suction intake are situated in concrete columns that are part of a dam in a river. For the subject case a formed suction intake needed to be developed that is insensitive to debris and fluid mud and low water levels at the entrance. The spatial limitations were bounded by the dimensions of the column. As a solution a double entry formed suction intake has been designed. In Figure 5 a cross-sectional and three dimensional impression of the design are shown. Besides the fact that the design is very compact it also shows that it is not fully symmetrical, due to the fixed position of the pump in the column. The intake has been designed and validated by means of CFD and model tested in air as well on full scale on site, both to fully satisfaction. A minimum submergence of S/D=0.25 has been met.

A wet-pit intake style that closely approximates a suction pipe arrangement and uses what is essentially a dry-pit pump has been expanded by the U.S. Department of the Interior, Bureau of Reclamation to an elbow-type suction tube design. This incorporates a

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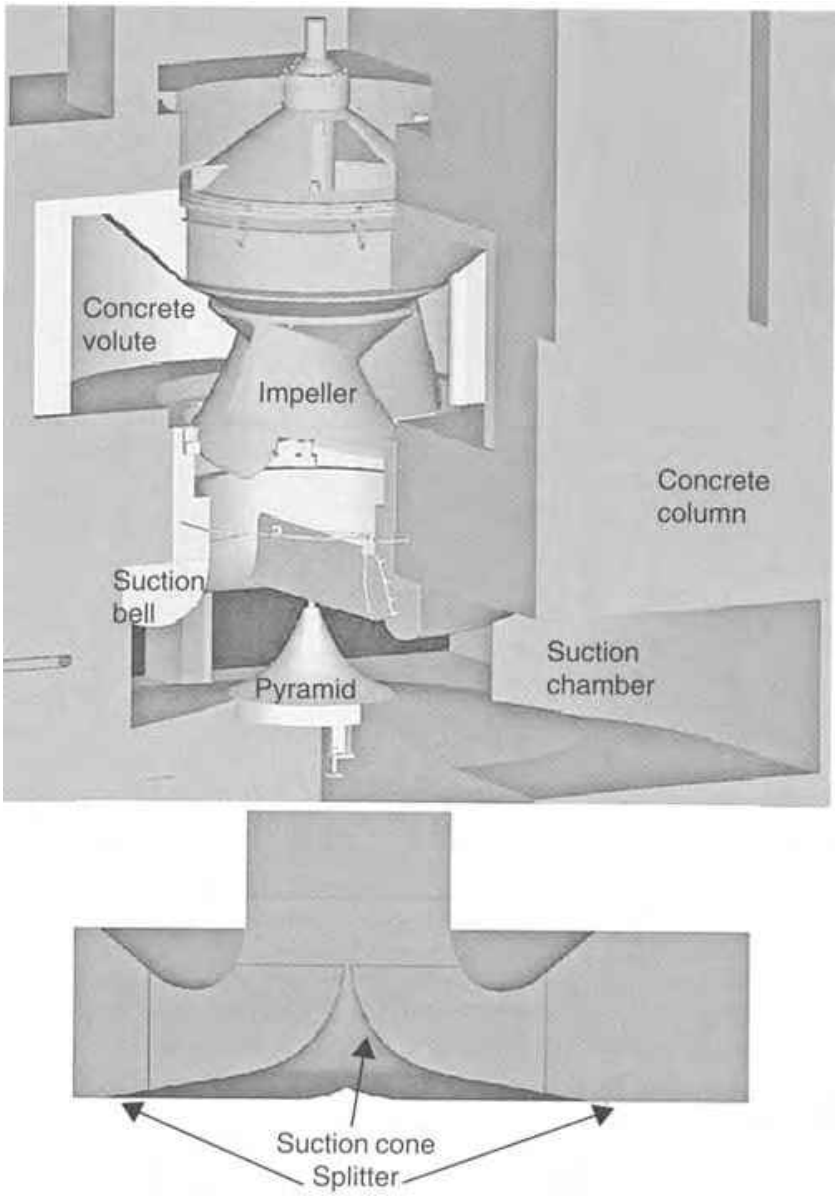


FIGURE 5 Design of an asymmetric double entry formed suction intake for pump inside concrete column which is situated in a river dam (Flowserve Corporation)

formed concrete suction inlet with a swing of 135° in the vertical plane and a gradual decrease in area to the suction eye of the pump (Figure 6). The resulting design saves considerable excavation in the wet-well area, reduces losses in the pump, and allows a smaller, higher-speed pump to be used. With higher velocities, debris dropout is reduced so silt buildup does not occur as readily and a smaller trash rack area remains effective longer. These inlets are independently self-sufficient and may be grouped into multiples as long as the forebay is designed to adhere to the basic design rules for wet-pit approach channels. Additional guidelines on the proportions of formed suction intakes are included in Reference 1.

SUCTION TANKS There are a variety of shapes and configurations for suction tanks. They may be vertical or horizontal, cylindrical, or rectangular. Outlets from the tanks may be vertically downward, horizontal, from either the side or bottom of the tank, or in the case of a pressurized tank, vertically upward. The outlet fitting of the discharge pipe of the suction tank can be straight, cone or bell shaped, either flush with or protruding through the

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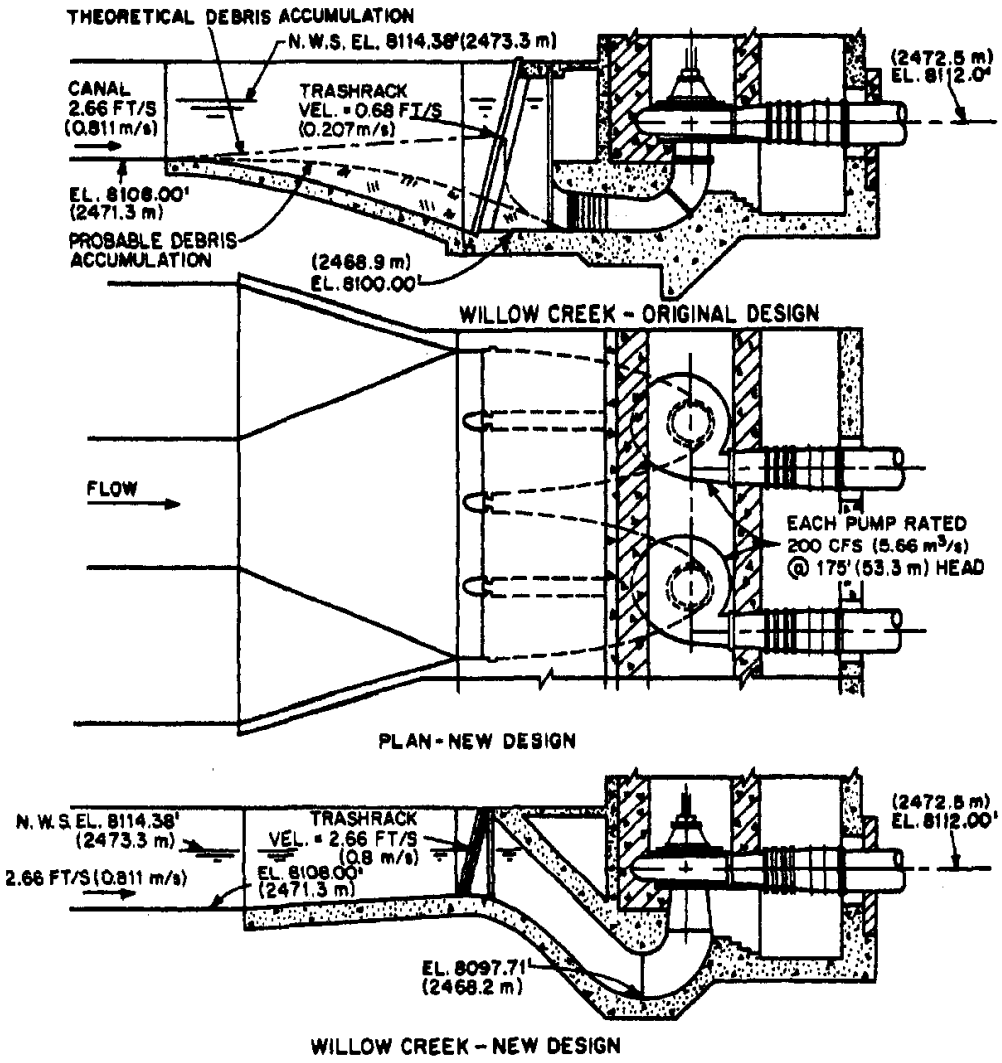


FIGURE 6 Improved 135° design of Bureau of Reclamation elbow suction tube inlet (U.S. Department of the Interior, Bureau of Reclamation)

tank interior surface. The formation of vortices in the tank can cause air or gas entrapment even when the tank outlet is fully submerged. This can cause pulsating pump flow, noisy operation, or a reduction in pump performance.

To determine minimum submergence of the outlet pipe in the tank, the Hydraulic Institute again recommends the following relationship:

$$S/D = 1.0 + 2.3\,F_D,$$

where F_D = Froude number = $V/(gD)^{0.5}$

D = outlet fitting diameter, ft (m)

S = submergence, ft (m)

V = outlet fitting velocity, ft/s (m/s)

g = acceleration of gravity, ft/s^2 (m/s^2)

The submergence S is the difference between the surface level and the centerline of a horizontal outlet or the level of the very beginning of a vertical oriented outlet pipe. The diameter D and the fluid velocity V are to be considered at the entrance plane of the straight, cone or bell-shaped outlet. If the tank has multiple inlets or outlets, the design should be such that flow interaction is not detrimental to overall performance.

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UNCONFINED INTAKE In case of unconfined intakes there is no structure around the wet-pit pump inlet to guide the flow. This is mostly seen with pumps placed on platforms in open water. Unconfined intakes give the best performance when unidirectional cross flow occurs at velocities less than 25% of the bell velocity. Obstructions to the pump suction that cause wakes should be located more than five times the cross sectional of the obstruction. The suction bell is best located 0.3 to 0.5 suction bell diameters above the bottom situated. In case of bottom debris this value can be increased.

CIRCULAR INTAKE The simplicity of shape, ease of construction, and compactness of a circular shape intake structure make this type of intake one of the most popular in the industry for both clear and solids-bearing liquids. Standard layouts of circular intakes are available for duplex and triplex sumps with pumps centerline or offset, dry or wet pit (see Reference 1 for further guidance). Intake designs with four or more pumps and/or flows exceeding 315 L/s (5000 gpm) require a model test.

A special type of circular intake is the Ranney well (Figure 7): a concrete silo 13 ft (4 m) in diameter, which becomes the collecting basin and pump well. It may be situated in or near a river and can be partly below and partly above water level with settings up to 100 ft (30 m). Small perforated pipes radiate horizontally from the base of the well and tap into porous strata, bringing small flows of water into the main well. This water can then be pumped from the well with a deep-well, wet-pit vertical pump. In the Ranney well, the

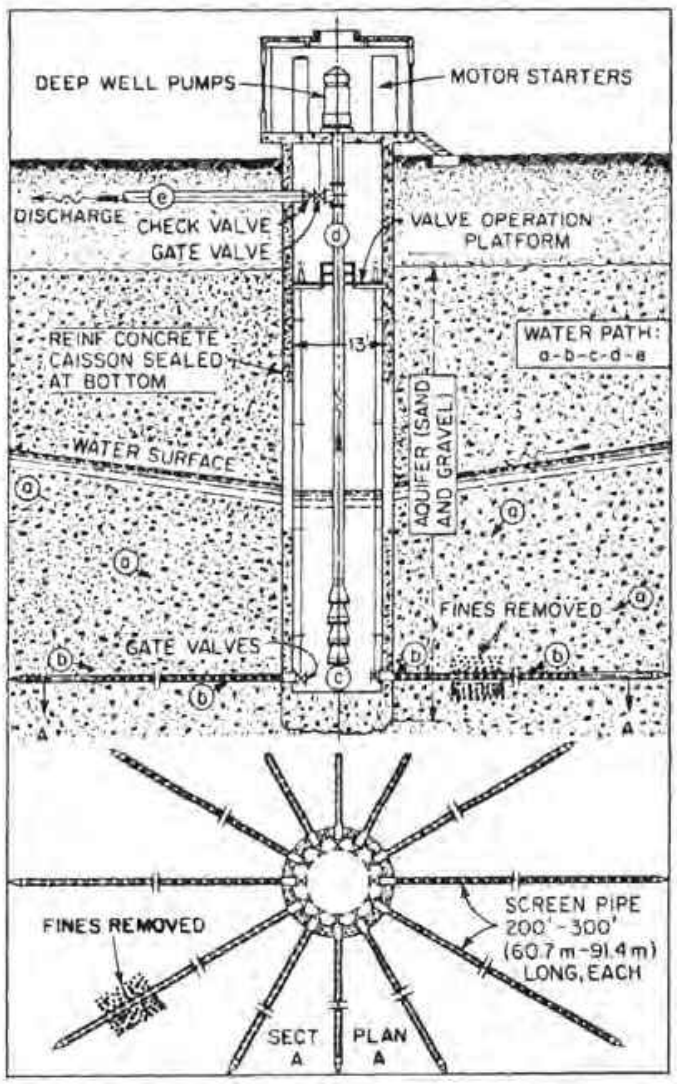


FIGURE 7 Ranney well collects into a pool from underground strata (Ranney Method Division, Pentron Industries)

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ratio of pump size (and flow) to pit size (and capacity) is such that very low velocity exists, as in a lake inlet. Water comes into the Ranney well all around the periphery. These conditions of direct flow and low velocity prevent vortexing.

Intakes for Solids-bearing Liquids Where most pump types must be prevented from entering debris and solids into the pump, there are also pump types that are made for pumping solids-bearing liquids like wastewater, polluted drainage water or discharges produced by industry. In case of wet-well intakes, care must be taken that the solids do not get too much accumulated, floating or settled, and that they can easily be removed.

In order to minimize the accumulation of solids in the intake structure the main idea is to minimize the presence of horizontal surfaces in the wet well. By positioning these horizontal surfaces only in the direct surroundings of the pump inlet, it is granted that the highest amount of solids will be sucked into the pump. The transition walls around the pump inlet area shall have steep angles (depending on the roughness of the wall material, larger than 45 to 60 degrees) to direct all solids in the direction of the pump inlet. Two types of intakes that perform well with solids-bearing liquids are trench-type and circular intakes, both modified with minimized horizontal surfaces at the pump inlet and sloping walls around.

Cooling Water Pump Intakes Cooling water circulating systems must have either a continuously renewable source, such as an ocean, lake, or river, or they must recirculate the same water from cooling ponds or cooling towers. Regardless of the type of pump selected (wet or dry pit), the suction water will come from an open pit of some sort or from a pressurized pipeline. Cooling water intake designs are categorized depending on the intake source: cooling pond, cooling tower, or ocean/lake/river source.

ONCE-THROUGH: OCEAN, LAKE, OR RIVER SOURCE—WET PIT Once-through intake structures are usually constructed of concrete and are arranged to gather the water into a localized area for pickup and to support the pumps. The optimum design will bring relatively clear water directly into the pump suction area at a low velocity.

For further guidance, Reference 1 provides intake design recommendations for both suction pipes and all types of wet pits that are a result of the combined efforts of sump designers, hydraulic researchers, pump manufacturers, and end users. It is intended to provide designers, owners, and users of pumping facilities a foundation upon which to develop functional and economical pumping facility designs. The choice of providing a trash rack or screen, or both, is based on the type and amount of debris likely to be encountered at the inlet.

An ocean inlet has additional requirements because of tidal action and variable direction currents that may exist. For these situations, it may be necessary to create a forebay inlet basin. Such a basin would be independent of the pump pit and fed by a submerged inlet tube extending out into the ocean for some distance (Figure 8). This tube may utilize normal pipe velocities, but the inlet must turn upward and the opening should be protected by a horizontal cap to allow the water to travel horizontally at a velocity high enough to scare fish away. This will prevent fish from being subject to a vertical velocity, which they are unable to notice, and will allow them to maintain a horizontal velocity that will direct them away from the inlet. The discharge into the forebay can be pipe velocity because the turbulence will be dissipated by lower outlet velocities [2 ft/s (0.6 m/s)] through trash racks and traveling screens. From this point onto the pump chamber, the velocity should be low and constant.

RECIRCULATING SYSTEM—COOLING TOWER For cooling tower systems, the pump pit is normally adjacent to the tower basin. Cooling tower basins are shallow, usually not more than 6 ft (1.8 m) water depth. Because pumps will require more submergence than this, a sloping ramp or sharp drop-off will be required if pumps are to be placed close to the basin. This will save initial costs for excavation and concrete work, but the resulting poor flow patterns conditions may result in endless pump problems.

Some successful small installations have been built with downslopes of 15° or more, but tower basin outlet velocity has been no more than 2 ft/s (0.6 m/s) and the

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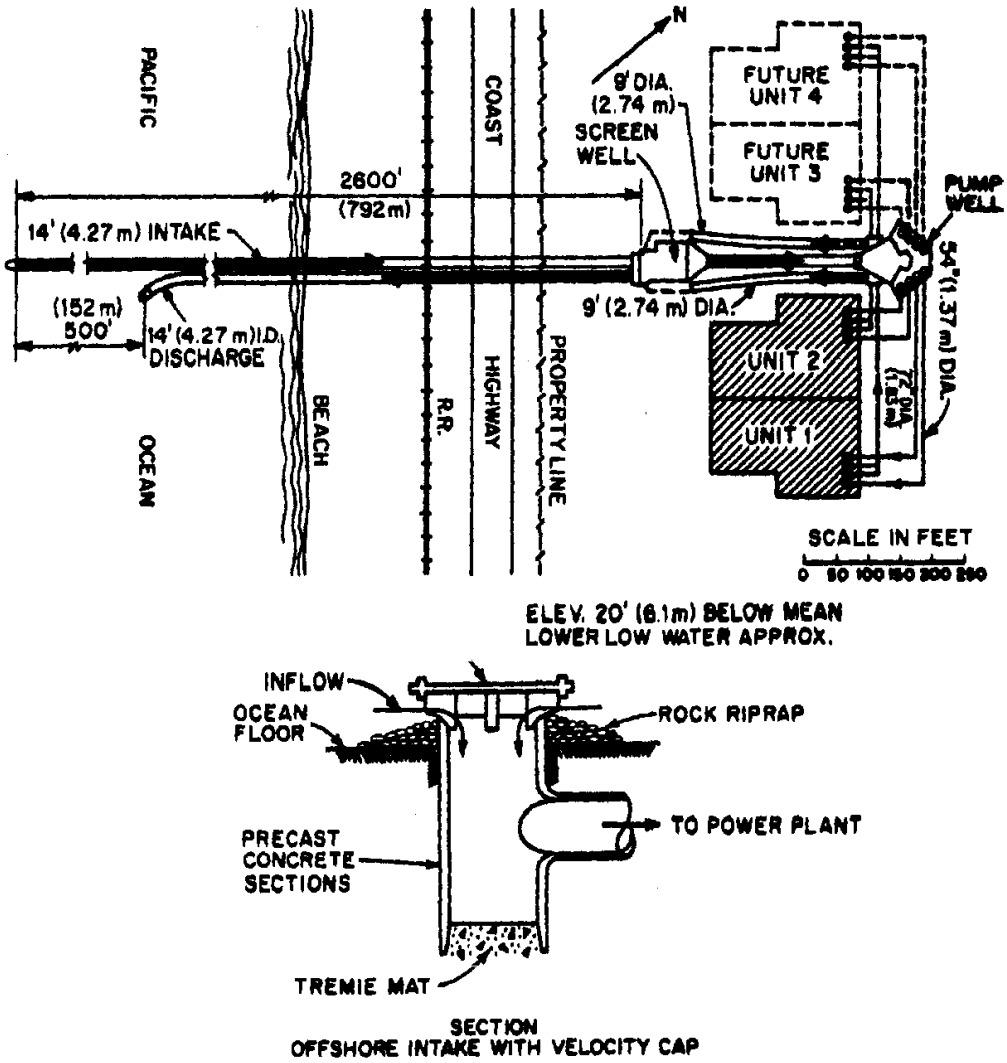


FIGURE 8 Ocean intake with velocity cap to minimize fish pickup

downslope widened as it approaches the pump pit. For low flows, low velocities, and fortunate arrangements of tower piers (lack of recirculation piping, and so on), such installations may be operated with reasonable success. Others of similar design, however, have proven to have disastrous effects on pump performance. Impellers and bearings may suffer rapid deterioration under such conditions.

The best solution, but obviously the most expensive solution, is to build the pump pit far enough from the tower basin to obtain a flat channel bottom. The channel should have an average channel velocity of 1 ft/s (0.3 m/s) or less for a distance at least equal to A in Figure 2, whether trash racks and screens are available. The slope from the basin floor to the channel bottom should not be more than 10°, preferably less than 7 degrees.

The basin exit width should be such as to make the exit velocity less than 2.5 ft/s (0.75 m/s). If several tower piers are in the exit path, this velocity limit should be reduced to 2 ft/s (0.6 m/s) or less. Upstream sides of the piers should be rounded off and the downstream sides should be tapered. Make the sidewalls of the downslope diverging so the velocity at the bottom is no more than 1 ft/s (0.3 m/s).

Velocities should be based on the runout flow of one pump. All dimensions and velocities mentioned are limits. If they are used based on maximum required flow rate, and the pump is selected for reasonable velocities, operation should be satisfactory. Obviously, if a pump with high velocities (suction bell inlet, impeller eye, rotational speed) is selected, it

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will require more submergence than a more conservatively designed machine. This will increase the cost by requiring a deeper pump pit located further away from the tower basin in order to perform satisfactorily.

A model test of the inlet structure is highly recommended if the best solution seems too expensive, or if topography will not allow the ideal arrangement. A model with a variable slope, variable sidewalls, and true representation of piers, pipes, and other obstructions should enable a satisfactory design to be achieved with a possible net reduction in overall cost. Actual pumps and valves should be used to create the flow in the pump pit, and extremes of flow and submergence should be included in the test program. Sometimes the cooling towers are on a hill, so a substantial drop in elevation provides pressure available nearer the flow area. If dry-pit pumps are used, this pressure can be utilized to reduce pump requirements. When this is done, the pump suction becomes a pipe, and the latter part of this section should be consulted.

RECIRCULATING SYSTEM—POND WET PIT The intake structure in a cooling pond should be located as far as possible from the inlet pipe to the pond to generate the maximum cooling effect. If spray surface equipment is used, it should be so arranged in relation to the intake building that a minimum surface disturbance is encountered. Prevailing winds should be considered, and the building should be located on the lee (upwind) side of the pond. If side and bottom areas might be easily disturbed to include silt in the flow, riprap—a wall or covering of stones thrown together randomly—should be applied to approach slopes as well as to bottom mats well beyond the inlet wing walls.

The handling of silt is usually not desirable in a pumping system. A high velocity through the pump will accelerate wear. At low velocity points in the system, silt will settle out and produce higher velocities, and more wear will occur as a result of the area blockage. If space is available, a silt-settling basin can be constructed ahead of the inlet basin. Cross baffles should be provided to slow the inlet flow to a velocity less than 1 ft/s (0.3 m/s). Most stream debris will settle out at this velocity, and the flow into the pump suction pit will be relatively clean, preventing the deposit of additional silt around the pump suction bell. If space is not available for silt beds in large-capacity installations, the main channel can be furnished with a weir across the flow path. The height of the weir should be selected to give an overflow depth above the weir of not more than one-third to one-fourth the water depth just preceding the weir. The velocity over the crest should not exceed the intake channel velocity. Weirs of this type are particularly effective when the intake channel is at right angles to the supply mainstream.

Multiple Pumps Some pump requirements are easily met with 100% capacity single pumps, but more reliability usually requires two 50% or three 33% pumps, or, if the service is sufficiently critical, three 50% or four 33% pumps, and so on. There are practical limits of size for various pump types, so large flow demands will undoubtedly call for a multiple pump arrangement.

The problem arising from multiple pumps arranged in a common pit is from the probable nonuse of some of the pumps while others are operating. This can cause variables in flow patterns that may lead to eddying and vortexing (Figure 9). Installation of separating walls in the common pit may introduce additional problems because the ends of the separating walls can create eddy currents in the corners at the unused pumps. A back vent in the dividing wall will relieve this situation, provided it vents at the water surface (Figure 10b left). If walls are extended past the screens and trash racks to a forebay, this problem will not occur, but the design has then become that of a single-pump basin.

The same velocity rules apply to multiple arrangements as to single-pump basins. Odd arrangements should be avoided even when they look invitingly symmetric—fan-shaped, round, radial, peripheral—all have directional problems that are not easily overcome. A basic pit design, consisting of a number of equally sized pumps in a common pit basin with flow entering parallel and straight in at 1 ft/s (0.3 m/s) or less, would not need to be model tested to assure reliability (Figure 10a left). If separating walls are required for structural support, and they are properly shaped and vented, no model test is recommended. The pumps should be located at the extreme rear of the pit so the whole approach assumes the characteristics of a suction pipe. Individual pump manufacturers may vary the location of

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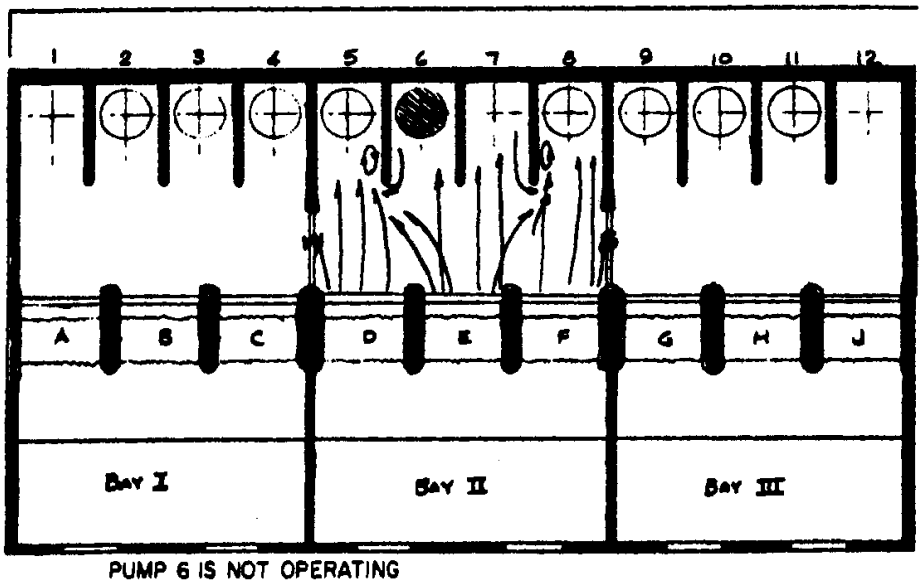
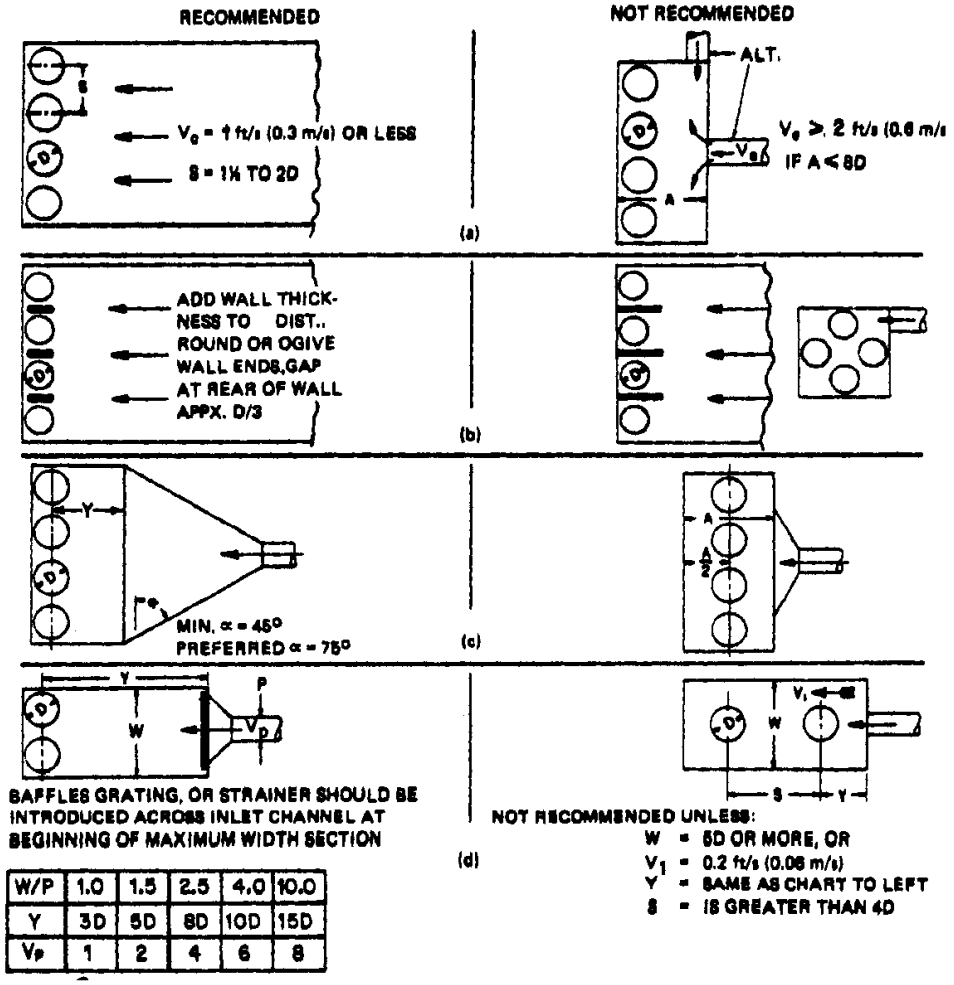


FIGURE 9 Empty spaces on nonoperating pumps cause vortexing at wall ends because of flow reversal



W/P	1.0	1.5	2.5	4.0	10.0
Y	3D	5D	8D	10D	15D
V_p	1	2	4	6	8

FIGURE 10 Using basics of good pit design precludes the need for model testing (Hydraulic Institute Standards, 13th Edition, 1975—out of print)

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the pump relative to the pit bottom, velocity of inlet spacing, and so on. It has been found that some of these variations require additional splitters or baffles below the pump, up the back wall behind the pump, or centered in the flow ahead of the pump. If so, a model test should be run and the additional pit cost weighed against other alternatives (changing the pit shape, pump location, pump size, and pump speed).

A dry-pit pump installation will have the pumps located either in a dry well at or below wet-well water level (Figures 1e and 1f) or directly above the wet well and using a suction lift (Figures 1g and 1h), which calls for priming equipment. The additional cost of priming equipment (vacuum pumps, and so on) may be partially offset by the additional space and valve requirement of the first option. In either case, the suction piping in the wet well should be treated in the same fashion as the pump suction bells in wet-pit installations as far as spacing, direction, and velocity of flow are concerned. When the pump is installed at an elevation that may be below the water level in the suction pit, a valve must be installed at the suction of the pump. The temptation to reduce the inlet size to use a smaller valve should be avoided. Pipe size at a pump inlet may decrease into the pump down to the pump suction size, but it would not be reduced below that size, and then have to increase again as it enters the pump. The pipe in the wet well should preferably have a bell end and project downward. The minimum water level above the lip of the bell can be estimated from Figure 4 for a recommended entrance velocity of 5.5 ft/s (1.7 m/s). The bell mouth should protect downward to assure uniform inlet flow and to attain maximum submergence.

Considerations for either single- or multiple-pump pit design relate primarily to even flow and low velocity into the pump. At the leading edge of the pump impeller (suction) vanes, this velocity will be increased and the direction of flow violently changed. To make the transition from pit flow to pump flow is the work of the pump designer. Some pump designs include ribs in the suction bell; others do not. It is obvious that the transition from parallel flow at 1 ft/s (0.3 m/s) to right-angle rotating flow at 15 to 20 ft/s (4.6 to 6.1 m/s) requires a high degree of skill in matching the suction bell to the impeller. If the pump is not designed to handle a wet-pit installation as described in previous paragraphs, a turning vane pit may be required.

Suction Pit Turning Vanes Figure 1c illustrates how turning vanes are used to guide the flow of water into the inlet of a vertical volute dry-pit pump. Figure 1k shows the same for the suction bell of a vertical diffuser wet-pit pump. If these pumps were connected to an inlet sump without turning vanes, as illustrated in Figures 1a and 1j, a wider and longer channel would be required to feed the suction bell from all directions. The pumps with turning vanes a narrower inlet channels may be desirable as multiple pumps can be spaced closer together providing screen width requirements do not dictate spacing.

Comparing depth of pit bottom below pumps with and without turning vanes, the following is to be noted. The excavation beneath a vertical volute dry-pit pump with turning vanes can be slightly less than the excavation beneath the same pump with no vanes but with a suction bell. The velocity approach into the closed portion of the channel beneath the pump can be as high as 3 ft/s (0.9 m/s) if turning vanes are used at the design flow; but it should be limited to 1.5 ft/s (0.46 m/s) with no turning vanes. Although the suction bell design requires a wider channel than a vaned inlet (approximately two-bell diameters), this is not wide enough and the channel must be made deeper to meet the lower velocity requirement for this type of inlet. When turning vanes are used with a vertical diffuser wet-pit pump, the pit must be excavated deeper than would be required if no vanes were used. This additional depth is required to form an elbow in the narrower channel and provide equal flow distribution to the impeller. Design velocity at the inlet vanes is 3 ft/s (0.9 m/s).

As an alternative to the use of turning vanes under a vertical volute dry-pit pump, the long-radius suction elbow inlet illustrated in Figure 1b offers some advantage in reducing the width and depth of excavation under the pump. The inlet velocity to the long-radius elbow, which is usually formed in concrete, is preferably no greater than 3 ft/s (0.9 m/s) at the design flow rate.

A decision to use turning vanes should not be based on a guarantee that there will be an increase in pump efficiency. The design of the vanes—their number and spacing—is still an art more than a science, and it is difficult to prove pump performance in the field.

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Turning vanes can be effective in eliminating underwater vortices, a problem sometimes associated with suction bells without turning vanes. It has been observed during model testing that the suction bell, as illustrated in Figure 1a, must be placed closer to the back wall than normally recommended for open channel inlets (similar to that in Figure 1j), to prevent underwater vortexing. The flow of water into a closed channel from an open pit containing water of considerably greater depth creates an unequal flow pattern in which the maximum velocity is along the floor. Also, there is little, if any, flow to the backside of the suction bell down from the top of the channel. Unless the bell lip is close to the back wall, flow along the floor and from the front only will overshoot the inlet and roll over, back, and up into the bell, forming an underwater vortex. Although turning vanes can prevent this, they may not prevent uneven flow distribution up into the pump impeller unless they are properly designed. They may even cause hydraulic and mechanical unbalance, which could result in noise, vibration, and accelerated wear of the pump bearings. For this reason, model testing of the turning vanes is recommended.

Environmental Considerations A pumping station has impact on different environmental parameters as it may:

- Change the water quality
  - Increase in temperature
  - Decreases the oxygen content
  - Change the clarity
  - Pollution by greases and/or lubrication oil
- Kill inhabitants living in the water
- Produce noise
- Change the hydrology and hydraulics

Off all these parameters, the suction pit design mainly influences the amount of inhabitants that travel though the pumping station with the chance of experiencing undue disturbance and destruction. The suction pit should be designed in such a way that it prevents as much fish as possible from entering the intake and the pumping system.

The most logical choice is to select a site for an intake not near feeding areas for large schools of fish (kelp beds, coral reefs, and similar attractive spots). Sheltered spots most suitable for intake flows may also be most attractive to fish. If the source is a river, the angle of the intake structure relative to the direction of flow is important in modifying the impact of these design requirements. Placing the screen house at a low angle to the river flow (instead of the usual 90° inlet) allows the river current to provide a swim-by attitude for fish while a low velocity screen approach is maintained. If the pump house is in line with the screen house, the pumps will feel minimal disturbance. A tendency for eddies to form (and create vortices) can be minimized by placing wing walls upstream and downstream to control velocity.

As an alternative to keep the fish away from the pump intake, remote intakes might be considered as an ocean outfall with velocity cap (Figure 8) or a Ranney well (Figure 7).

A design to accommodate fish limitations was mentioned briefly in a previous paragraph. Fish react to a horizontal velocity but are not aware of a pull in a vertical direction. Thus, to keep them from entering the inlet, a horizontal flow must be established at a velocity low enough to permit fish to escape. Inlet screen areas should be in small sections rather than one long face so a fish will not be trapped in the center and find it too far to swim to safety after it realizes its predicament. The maximum deterrent flow is about 10 ft/s (3 m/s), but this may be too much disturbance for the flow to the pumps. Smaller areas allow short-term limits for enticing fish away from the inlet, and survival will be much higher.

Fish congregating at an inlet or in a forebay pool can be crowded or herded to an outlet point by the use of vertical nets or horizontal screens. In a direct channel, horizontal moving screens can route fish past a sloping (relative to stream flow, say 35 degrees) moving screen. This results in directing fish to a narrow outlet at one side, leading to an outlet channel away from the main inlet.

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The most difficult problem to overcome is related to small fish. Screen openings must be held to a minimum, and under velocity conditions, small fish have much less swimming-sustaining ability than larger fish, both in speed and in duration time. In a given steam flow (such as is generated by pumps with inlet water going through screens), a fish must have the ability to sustain a given speed against this flow for a certain length of time. When it weakens, it will fall into the current flow and will be impaled against the screen and destroyed. If the fish senses the velocity early enough, and has an alternate route, it can use darting speed to escape. Or it can follow another attraction (cross velocity flow into a separate chamber or a light attraction to the chamber) and be removed on an elevator or pumped out to a safety channel (Figure 11). Migrating fish need a continuation channel to restore their interrupted journey.

In designing an intake, it is necessary to keep the velocity below 0.5 ft/s (0.15 m/s) through the screen to avoid drawing fish into the screen. Alternatively, a cross flow can

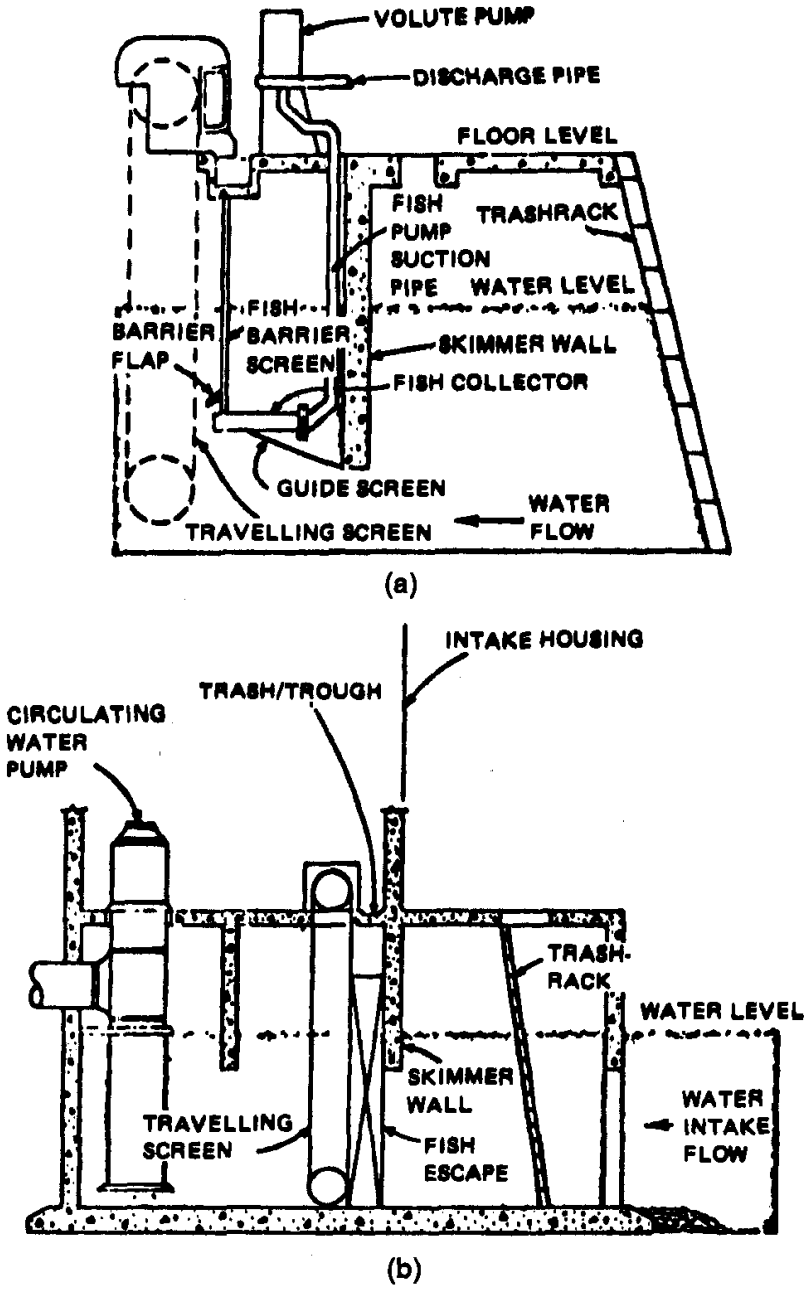


FIGURE 11 (a) Fish pump prevents entrapment on traveling screen (Detroit-Edison). (b) Fish escape allows fish to bypass screen area.

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be created that will propel or attract the fish to one side of the inlet area. From there, they can be directed into a bypass pool and lifted back to their own living area, or they can be sent around the plant to a downstream location. If they are anatropous, they can be sent to an upstream rendezvous. Piers and screens should be kept flush across their inlet face to prevent attractive pockets where fish can hide and be drawn into the screens when they weaken.

Intakes that take their flow directly from a river may have a high velocity that would trap fish. Even if the velocities are lowered to reasonable screen levels—2 ft/s (0.6 m/s)—fish may still be drawn into the screen area and carried up to trash disposal. Obviously, trash and fish must be handled in separate areas, and screen wash pressures must be lowered to prevent harm to the fish.

Small slot-width wedgewire screens (Figure 12) are now being furnished in larger sizes, and, with their very low inlet velocity and backflushing capability, offer one good approach to reducing entrainment. Sand, chemical, and cloth filters may suit some situations, but backflushing and cleaning problems make them less attractive costwise.

Entrainment of marine organisms too small to be restrained by normal screens causes further problems in areas where such organisms normally develop. If the intake structure must be located in such an area, extensive information must be gathered at the site. Analyzing as much data as are available, in conjunction with plant flow and location requirements, may give the designer some idea of which equipment to select.

CFD Analysis and Model Testing of Intake Design When the basic rules for good pump suction pit design are adhered to, no model test will be required to ensure proper operation of the pumps and pumping system. The substance of these rules is to keep a straight-in approach at a constant low velocity from the water source to the pump chamber. The ANSI/Hydraulic Institute Standard dimensions and charts satisfy these criteria for the average pump in general application.

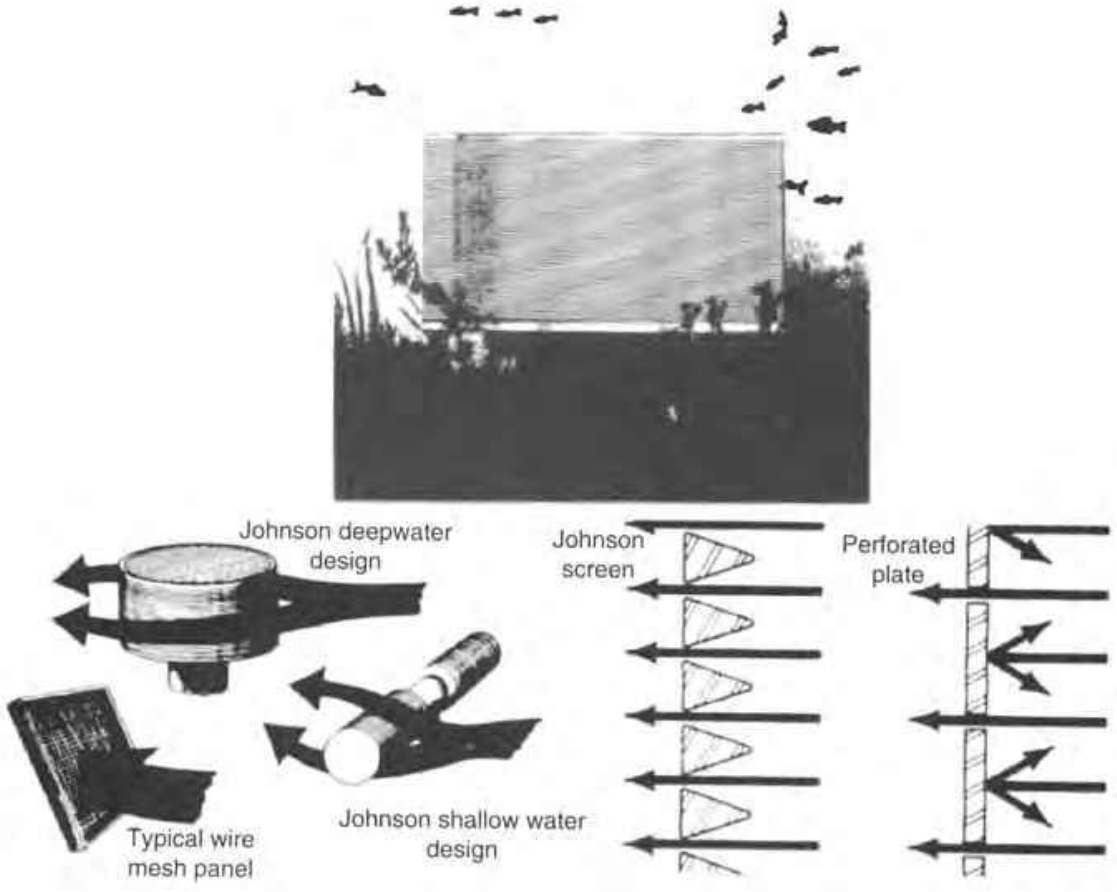


FIGURE 12 V-shaped slotted screen provides velocity control to avoid attracting fish and good air or water backflushing for debris cleanup (Johnson Division, UOP)

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Site layout problems may make the ideal solution impossible. Structural and environmental requirements may outweigh hydraulic requirements in some instances. When the ideal pit may not be possible or economically feasible, CFD analysis is very convenient to optimize the design within the given limitations. It is recommended that the outcome of such study be verified with a physical model test. It should be noted that the ideal dimensions are a composite covering not only a range of specific speeds but also a complex melding of pump design philosophies. Some variation from ideal dimensions should be expected from individual pump manufacturers.

Pump manufacturers are not in a position to guarantee the pump pit design. Differences of opinion between the structural and hydraulic pit design engineers and the pump design engineers may best be resolved by performing CFD analysis and/or a model pit test (Figure 13). Refer also to Section 13.2.

Vortexing The real problems resulting from improper pit design occur largely on the water surface in the form of vortices, or cones, produced by localized eddies on the surface of the water. If this disturbance continues, the flow of water will carry the underwater part of the vortex down toward the pump suction bell and ultimately into the pump (Figure 14). This introduces air into the impeller and will affect the mechanical radial balance of the impeller by interrupting the normal solid-liquid flow pattern. This type of disturbance will produce hydraulic pulsations in the pump flow and mechanical overloading of bearings and impeller guides.

Underwater vortexing sometimes occurs in round pits or in pits where the pump suction bell is at some distance from the rear wall. Flow past the suction bell strikes the rear wall and rolls back toward the bell, wall, and rolls back toward the bell, forming an eddy current that disturbs the normal flow into the pump. In a round pit, a cross baffle below the pump bell may reduce this effect. Where the pump is some distance from the back wall, a wall can be installed near the pump, or a horizontal baffle at suction bell level behind the pump will also reduce the disturbance. In all cases, the distance between the suction bell and the bottom of the pit should not be more than one-half the suction bell diameter, and one-third the diameter is preferable.

The use of the suction bell diameter as a basis for spacing should be carefully evaluated. It will be seen that there is nothing magical in this relationship, especially when several pump manufacturers all use different bell diameters. The real criterion is the allowable velocity at the suction bell. It has been found that very-low-head pumps are



FIGURE 13 A typical suction bay intake model test (University of Kaiserslautern)

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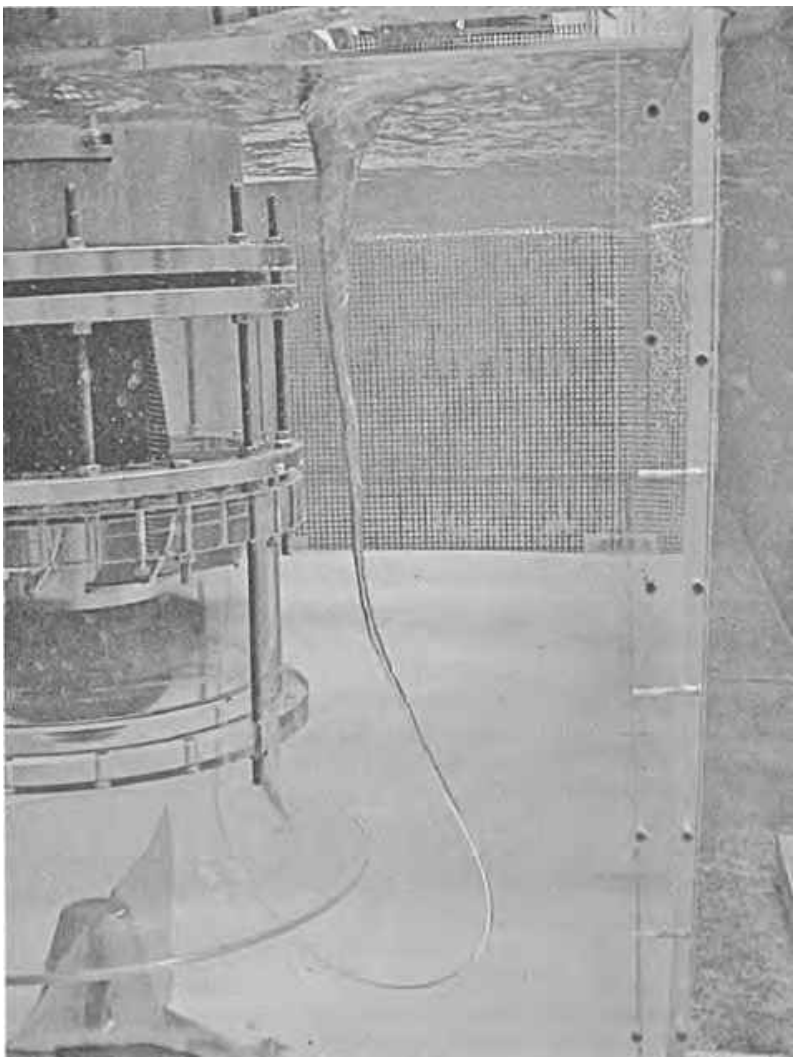


FIGURE 14 Surface vortex drawing air into pump suction bell (University of Kaiserslautern)

much more sensitive to bell velocity over 6 ft/s (1.8 m/s). For example, the velocity head loss at the bell inlet with a high velocity may be such a large percentage of total pump head that efficiency could drop as much as 10%. A good design rule for safe operation can be related to pump head. For pumps having up to a 15-ft (4.6-m) head, the suction bell velocity should be held to 2.5 ft/s (0.76 m/s); up to 50-ft (15-m) head, 4 ft/s (1.2 m/s); and above 50-ft (15-m) head, 5.5 ft/s (1.7 m/s). These values should be used for any substantial amount of pumping, but for occasional short-term pumping, they can be exceeded without destroying the pump.

Vortices may be broken up and effectually nullified by arrangements of baffles and vanes, or they may be prevented from occurring initially by a proper pit design. To determine what type of baffling should be used, CFD can be of help as it can predict the origin of vortices. To fully predict the effectiveness of the proposed baffling, model testing is needed. Methods of eliminating vortexing are discussed in Section 13.2.

Vortices are usually generated when the flow direction of the liquid to be pumped changes or when there is high velocity past an obstruction, such as a gate inlet corner, screen pier, or dividing wall. In combination, these two causes invariably generate vortexes. For this reason, the pump suction pit should be immediately preceded by a straight channel in which the velocity does not exceed 1.25 ft/s (0.4 m/s). Satisfactorily operating pump pits with higher velocities are rare and should not be put into operation without the assurance of a model study.

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An additional condition likely to generate vortexes is a multiple pump pit with individual cells in which only a portion of the pumps will operate simultaneously. The dead space behind the nonoperating pump will have flowing water tending to reverse direction and form eddies. Eddies and vortices can be avoided by eliminating or venting the walls at the rear of the pit. It also helps to position the pumps at the extreme rear of the pit. Alternatively, expensive modifications to the pit, such as splitter walls and baffles, may be required (Figure 9).

Round pits tend to generate vortices, especially when the pump is centered in the pit. These vortices usually occur around the pump column because of eccentric inlet flow. Special cases of the round pit are tolerable either when a Ranney well (Figure 7) is used or when there are booster pumps in the pipelines (Figure 15). In the Ranney well, the ratio of pump size (and flow) to pit size (and capacity) is such that very low velocity exists, as in a lake inlet. Water comes into the Ranney well all around the periphery. These conditions of direct flow and low velocity prevent vortexing. Booster pumps installed in a circular can (suction tank) must be centered in the can, and all inlet velocities to the can and flow in the can and into the pump must be uniform and high enough to provide fluid control. This velocity will vary from 4 to 6.5 ft/s (0.76 to 1.98 m/s).

Vortices are not generated by a pump or pump impeller and so do not fall into clockwise or counterclockwise rotation because of the pump rotation. Also, in a pump pit, vortices do not have a directional rotation induced by the rotation of the earth and therefore are not of opposite rotations above and below the equator, as is the case with "bathtub vortexing," which occurs in tanks being drained without pumps.



FIGURE 15 Booster pump suspended in a steel well (or can) that requires a minimum space for suction pit (Flowserve Corporation)

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CAN PUMP INTAKES

A can pump is defined as one that has a can, or barrel, surrounding the pumping unit. This can acts as a "sump" or intake structure for the pump suction impeller. The can may be either closed bottom, and contain the pump suction nozzle, or open bottom and connect directly to a piping header. The can design must provide uniform, stable flow distribution to the suction impeller inlet.

Vertical turbine pumps (Figure 15) require uniform inflow to the suction bell to avoid swirling and submerged vortices that may result in cavitation, vibration, and accelerated pump wear. When a pump with an open bottom can design is connected to a horizontal header (Figure 16a), the velocity in the header should be no more than 6 ft/s (1.8 m/s) to allow the liquid to turn and flow upward to the pump suction bell. The velocity in the can rising to the pump should not exceed 5 ft/s (1.5 m/s), and the suction bell inlet should be located at least three diameters above the top of the horizontal header. For velocities approaching the maximum recommended levels, vortex-suppressing vanes may be added to the suction bell area to break up swirling and nonsymmetrical flow patterns as they approach the impeller inlet.

If a 90° turning vane elbow on the can assembly surrounding the pumping element is used (Figure 16b), the velocity in the horizontal header can be as high as 8 ft/s (2.4 m/s). The turning vane elbow should be sized for a maximum velocity of 5 ft/s (1.5 m/s).

Most can pumps are of the closed bottom design (Figure 17). In this arrangement, the pump suction nozzle is located either in the can or in the pump nozzle head that contains both the suction and discharge nozzles. The pumping element (assembly) must be centered in the can to avoid nonuniform flow to the suction impeller. Flow straightening vanes are suggested for all can intakes. Because of the limited volume of liquid in the can, surging of the liquid level within the can—or barrel—cannot be tolerated; the can should be arranged so it is always full. The velocity of the flow between the pumping element and the inside of the can should not exceed 5 ft/s (1.5 m/s). For a suction nozzle velocity of 4 ft/s (1.2 m/s) maximum, the centerline of the nozzle should be at least two diameters above the suction bell inlet. To ensure uniform flow distribution into the can, the suction nozzle should be connected to at least five pipe diameters of straight pipe before elbows or other flow-disturbing fittings are installed.

Pumps with submersible well-type motors require flow around the motor for cooling. A shroud is typically used to direct flow across the motor as it goes into the pump. The top of such a cooling shroud is covered to restrict downward fluid flow and still allow venting

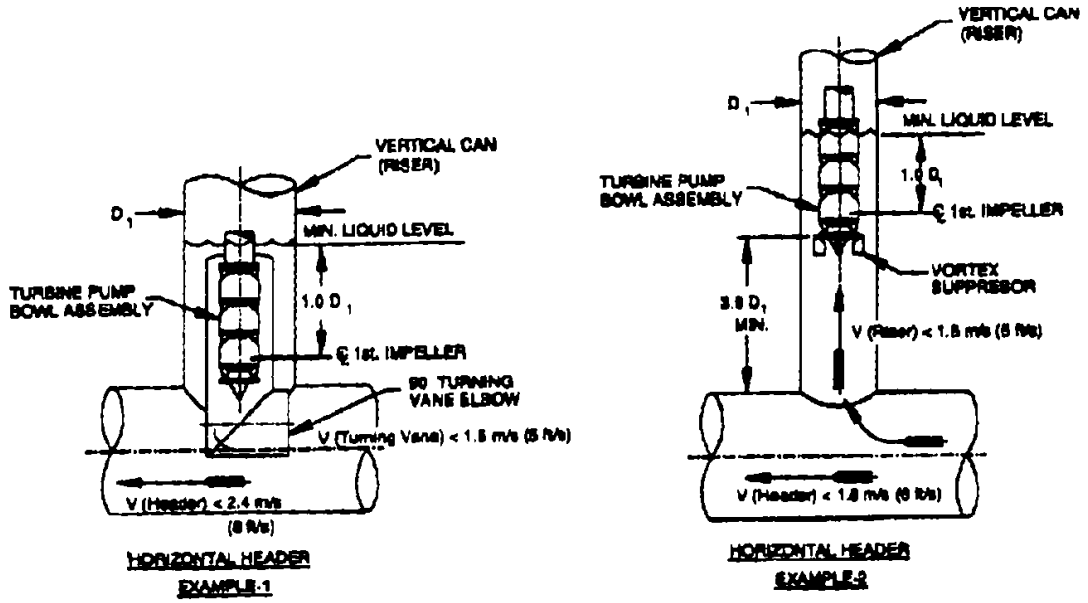


FIGURE 16 Open bottom can intakes for pumps less than 5000 gpm (315 L/s) (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

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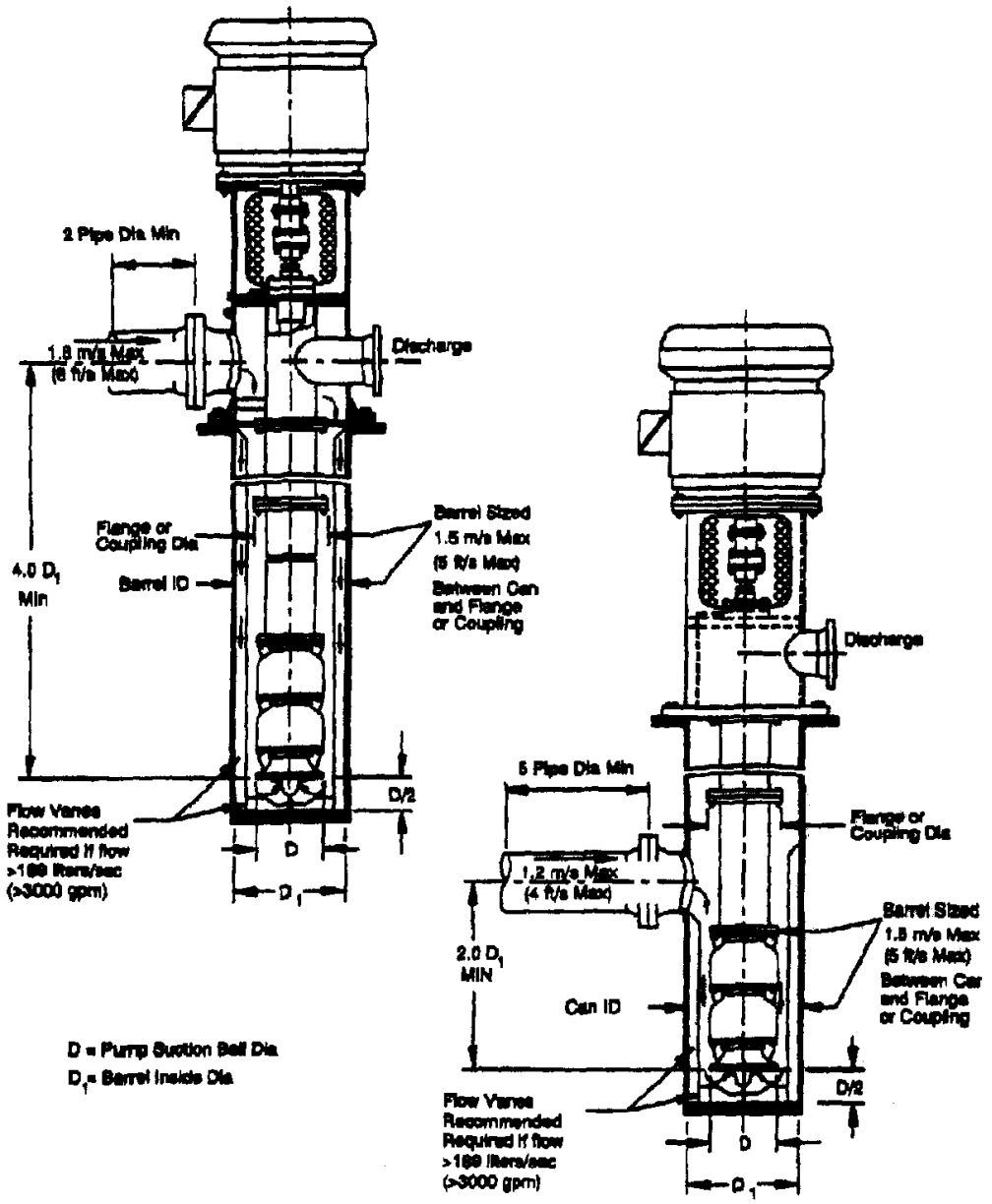


FIGURE 17 Closed bottom can (American National Standard for Pump Intake Design, ANSI/HI 9.8-1998, Reference 1)

of air from the shroud. This arrangement is described in detail in Reference 1. It is recommended that the inlet piping be sized to limit draw-down of the liquid below the minimum required level during the start-up to a period of less than 3 s.

SUCTION PIPING

Single Pumps Piping to the suction of a dry-pit centrifugal pump (Figure 18) must be carefully planned to provide uniform, straight-line flow to the impeller, and adequate pressure and sealing against leakage, in or out. Air pockets just prior to entrance to the pump should be avoided, as well as down flow lines subject to sudden pressure changes. Air pockets can be prevented by proper elevations (Figure 18a). Pressure surges can be controlled by surge tanks, air tubes, and so on, which may require a system pulsation study to determine possible need and solution.

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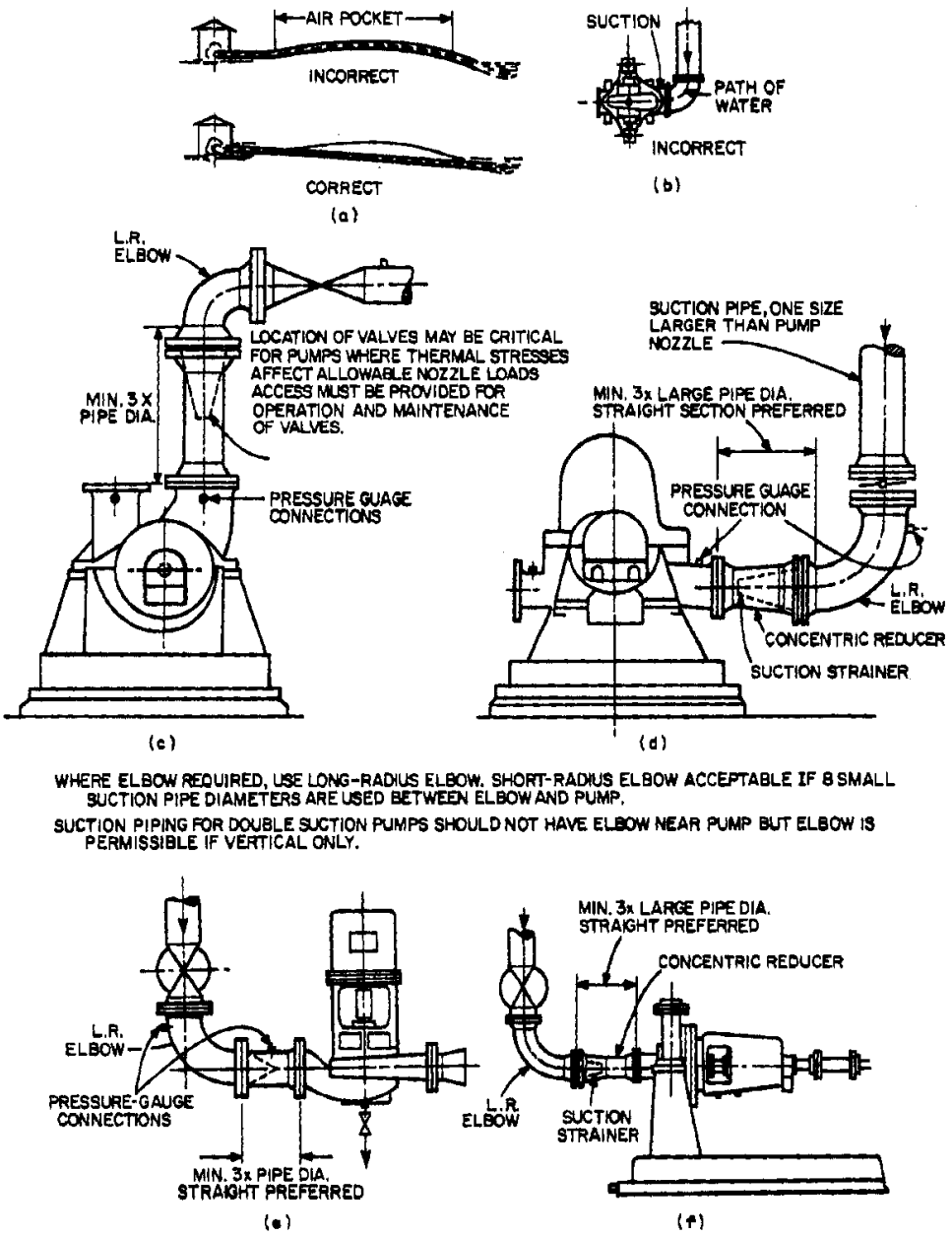


FIGURE 18 Suction piping faults for dry-pit centrifugal pump. (a) Air pockets should be avoided. (b) Suction elbow should not be in a plane parallel to pump shaft. (c) Valve location may be critical to pump nozzle loading. (d), (e), Suction elbow should be one size larger than pump's, long radius and three large pipe diameters distant. (d), (f) Concentric reducer should be installed distant from pump.

To provide an optimum flow pattern to avoid impeller disturbance, it may be necessary to have a straight run of pipe of as much as eight pipe diameters immediately prior to the pump suction (e.g., following a short radius elbow or tee). Following a long radius elbow or a concentric reducer, a straight run of at least three pipe diameters is recommended (Figure 18c to f). Eccentric reducers should not be used next to the pump suction nozzle. Although installing eccentric reducers with the flat side on top will eliminate a potential air pocket, large changes in diameter could result in a disturbed flow pattern to the impeller and cause vibration and rapid wear. Pipe venting, in conjunction with a concentric reducer, may be preferable to the use of an eccentric reducer.

The velocity of the fluid in the suction piping should only be exceeding 8 ft/s (2.4 m/s) when a uniform velocity profile and sufficient pressure at the pump inlet flange can be granted. The velocity may increase in the direction of the pump through the use of reducers.

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Ideally, a suction pipe should approach a double suction pump perpendicular to the shaft centerline. If there is an elbow in the suction piping upstream of the suction flange, it should be bringing flow from either overhead or below, not from the side of the pump. If there is a short radius elbow or other flow-disturbing device in the suction piping upstream of the suction flange, there should be at least five pipe diameters between the device and the suction flange. If a short radius elbow is in the same plane as the impeller shaft, there should be at least eight pipe diameters between the elbow and the suction flange. An incorrect installation could result in an uneven flow to both sides of the double suction impeller. This could cause a reduction in capacity and efficiency, an increase in thrust on the bearing, vibration, noise, and possible cavitation damage to the impeller. A dry-pit pump (Figure 1g) may operate with a suction lift and therefore will be located above the liquid source. All losses in piping and fittings will reduce the available suction pressure. Suction piping should be kept as simple and straightforward as possible. Any pipe flange joint or threaded connection on the suction line should be gasketed or sealed to prevent air in-leakage, which would upset the vacuum and keep the pump from operating properly (already at air rates of 1% and more).

If expansion joints are required at the suction of a pump, an anchor should be interposed between the pump nozzle and the expansion joint to prevent additional forces from being transmitted to the pump case and disturbing rotating clearances. The same requirement applies to a sleeve coupling used to facilitate installation alignment.

Reciprocating pumps must have additional consideration because of the pulsating nature of their flow. Suction piping should be as short as possible and have as few turns as possible. Elbows should be long-radius. Pipe should be large enough to keep the velocity between 1 and 2 ft/s (0.3 and 0.6 m/s). This will generally result in pipe one to two sizes larger than the pump nozzle. High points that may collect vapor are to be avoided or, if necessary, properly vented. A pulsation dampener or suction bottle should be installed next to the pump inlet. Available NPSH should be sufficient to cover not only reciprocating pump requirements and frictional losses but also acceleration head (see "Surge and Vibration").

Manifold Systems All comments relative to single pumps apply to manifold-pump systems, as well as some additional points.

In a suction manifold, the main-line flow should not be more than 8 ft/s (2.4 m/s). Branch outlets should be at 30 to 45° relative to main-line flow rather than 90° but never positioned opposite toward each other. The diameter of the branch outlet piping should be smaller than in the manifold piping (Figure 19). Uniformity of the velocity profile in the branch outlet is further increased through the use of cone- or bell-shaped fittings. Branch outlets can be spaced 2 header diameters apart from each other for larger ratios between branch outlet diameter and header diameter ($\geq$0.3), while 3 branch outlet diameters is sufficient for diameter ratios less than 0.3. Also, if the angled manifold outlet is used, pumps can be set as close to the manifold as the elbow, valve, and tapered reducer will allow.

Manifold sections beyond each branch takeoff should be reduced to such a size that the velocity remains constant. One exception to this scheme is tunnel suction (Figure 20). The flow through the tunnel may operate independent of the pumps, which are suspended in boreholes drilled into the roof of the tunnel. Boreholes at least one-third of the tunnel diameter should be horizontally spaced at least 12 borehole diameters apart. Smaller ratios can be closer to a minimum of six diameters. Pump suction bells should be at least two borehole diameters above the tunnel roof. Velocities in the tunnel should be kept below 8 ft/s (2.4 m/s) for best pump performance.

NPSH The net positive suction head so essential to trouble-free pump operation is always reduced by losses in suction piping. An economic balance must be obtained between pump size and speed, required NPSH, pipe size, and suction velocity. If the suction source is a tank, such as a deaerator in a power plant, it is quite expensive to elevate the tank. Therefore, the available NPSH is low. The pipe size from the tank to the pump should be large for low velocity and the length should be short for minimum losses.

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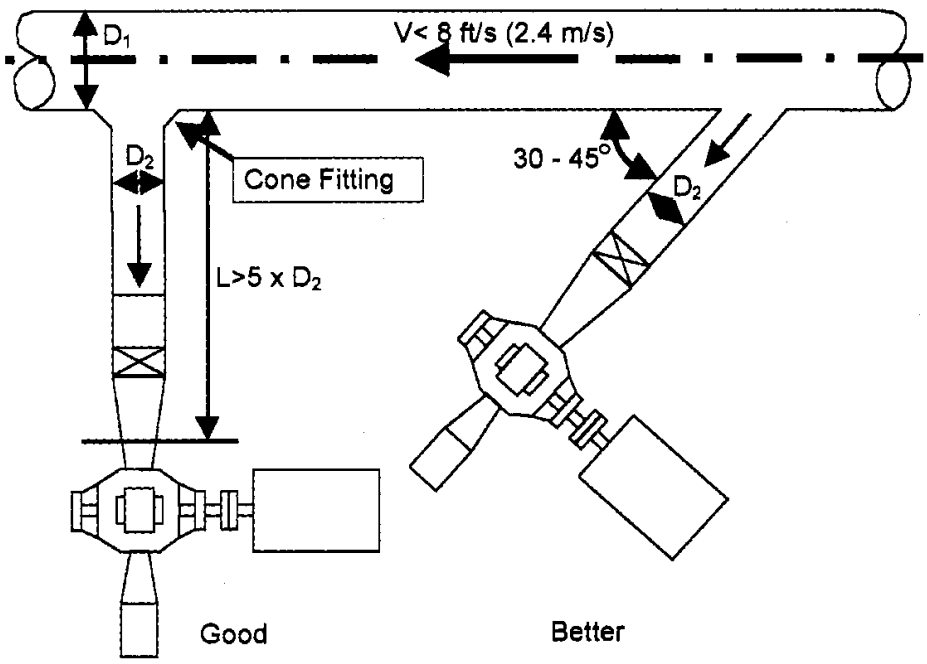


FIGURE 19 Suction pipe header recommendations for dry-pit centrifugal pump

A cooling tower on a hill supplying water to a pump station below could have a much higher pipe velocity and longer length without forcing an extremely low NPSH requirement on the pump.

A dry-pit pump should be as close to the suction source as possible. When the NPSH required indicates a suction lift is possible, the most advantageous solution is to reduce the suction losses by increasing the pipe size, flaring the inlet bell, and keeping the pump suction eye of the first-stage impeller close to the minimum water level.

High-Pressure Inlets Pumps in series build up pressure, so the second and following pumps will have a high-pressure suction piping connection. This emphasizes the need for tight joints and flanges and careful welding. Expansion joints should not be used because the hydraulic forces on the pump would be large, difficult to restrain, and perhaps impossible for the pump to handle without distortion. As NPSH will not be a problem for pumps downstream of the first pump, the pipe size between pumps can be kept small to minimize design problems and valve costs.

Effect on Pump Efficiency Other than mechanical operation, the greatest effect of flow disturbance at the pump suction is on pump efficiency. The higher the pumping head, the less this effect becomes. For very-high-head pumps, a high velocity may be ignored completely unless there is an extremely large power evaluation factor. Greater attention

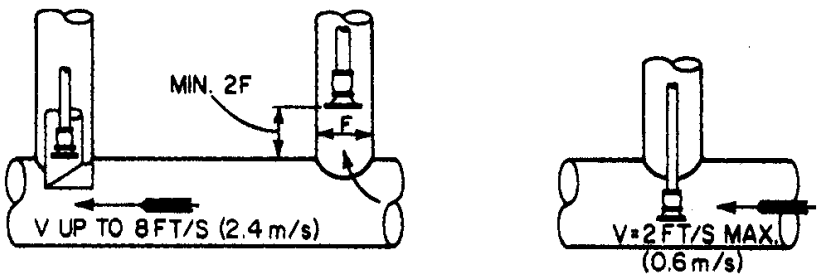


FIGURE 20 Higher tunnel velocities require isolation of pump from direct flow to prevent distortion of close clearance parts and shaft (Hydraulic Institute Standards, 13th Edition, 1975—out of print)

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should be given to the suction pipe design for pumps producing 100 ft (30 m) of head or less than those with high head, as efficiency may be worth many dollars in power costs over the life of the plant and equipment.

Pump efficiency relates to the efficiency of the whole system, and so it may be well worthwhile to invest more money in a large suction pipe.

Surge and Vibration One of the possibilities arising from a power failure at a pump station is the reversal of a centrifugal pump if a valve fails to close, and its subsequent operation as a turbine. Underrated head, a pump will run from 20 to 60% above rated speed in the reverse direction. In its transition to that phase, the forward motion of the water is interrupted and gradually reversed. At the time of the power failure, the flow velocity in the suction pipe may not decelerate slowly enough to prevent a surge in the direction of the pump. The suction piping should be designed to withstand the resulting pressure rise because absolute integrity of valving and power supply will be too costly as a design parameter (see Chap. 11 for additional information on this subject).

Rotating machinery must have a design vibration frequency that is sufficiently removed from a system frequency to avoid sympathetic activity. Any combination of vibration frequencies near enough to each other to react will do so when a prime source, such as a pump, excites them. When a piping system has been designed to allow only very low stresses to be transmitted to a pump nozzle, it is quite vulnerable to vibration. It is usually good practice to analyze a suction system made up of pipe, valves, hangers, restraints, pump nozzle loads, pump speed and impeller configuration, foundation, and anchors to be sure the system is not "in tune." At the design stage, it is relatively easy to change a valve or elbow location or to add a surge suppresser. It is usually much more costly to change a pump.

Reciprocating pumps are surge-producing machines. In particular, they require sufficient energy at suction to overcome pump required NPSH and pipe friction and a form of energy called *acceleration head*. The pump energy must overcome the acceleration-deceleration pulsation flow in the suction end, which could lead to liquid flashing with pump noise and vibration. Surges large enough to rupture the pump cylinders may also be produced.

The total NPSH required by a reciprocating pump must include the NPSH required by the pump plus frictional loss from the suction pipe plus acceleration head. Acceleration head in feet (meters) is

$$Ha = \frac{LVNC}{gK}$$

where L = actual (not equivalent) length of suction pipe, ft (m)

V = velocity of flow in the suction line, ft/s (m/s)

N = pump speed, rpm

C = pump constant, decreasing with number of cylinders from 0.2 to 0.04

g = acceleration of gravity, 32.17 ft/s^2 (9.807 m/s^2)

K = liquid constant: 2.5 for compressible fluids, 1.5 for water

The acceleration head can be reduced by installing a pulsation dampener in the suction line near the pump.

TRASHRACKS, SCREENS, AND STRAINERS

Except for special designs for handling slurries and other solids, most pumps are limited in their ability to handle dirt, trash, and solids of any size down to microparticles. Pump passageways (volutes, diffusers, impellers) handle generally spherical objects up to a "sticking size," but a severe problem may be caused by dust-size particles that lodge in sleeve bearings, bushings, and wear rings and cause either rapid wear or rotation seizure.

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Wet-pit sources with water intake structures are very likely to be polluted by different kinds of debris. Depending on the probable source of debris, such as a river subject to flooding with considerable flotsam in the runoff and with a very loose bottom, or a lake at constant level without disturbing inlet flows near the structure and with a solid bottom, the protection needed may include only a bar trash rack plus rotating, flushed, fine-mesh screens. If sediment deposit is likely, a settling basin may be required.

The designer must note that, if this equipment is useful, it will pick up debris and gradually increase the velocity through the openings as the net area decreases with blockage. When this occurs at the trash rack, the water level differential will build up, causing a waterfall with increased velocity and turbulence on the pump side of the rack. In addition, the increase in velocity may pull more debris through the bars than can be tolerated. It is best to rake these racks (Figure 21) frequently enough to keep the differential head across the rack below 6 in (0.15 m). The spacing of the bars should be such that objects that cannot be pumped would be excluded from passing through. This, in

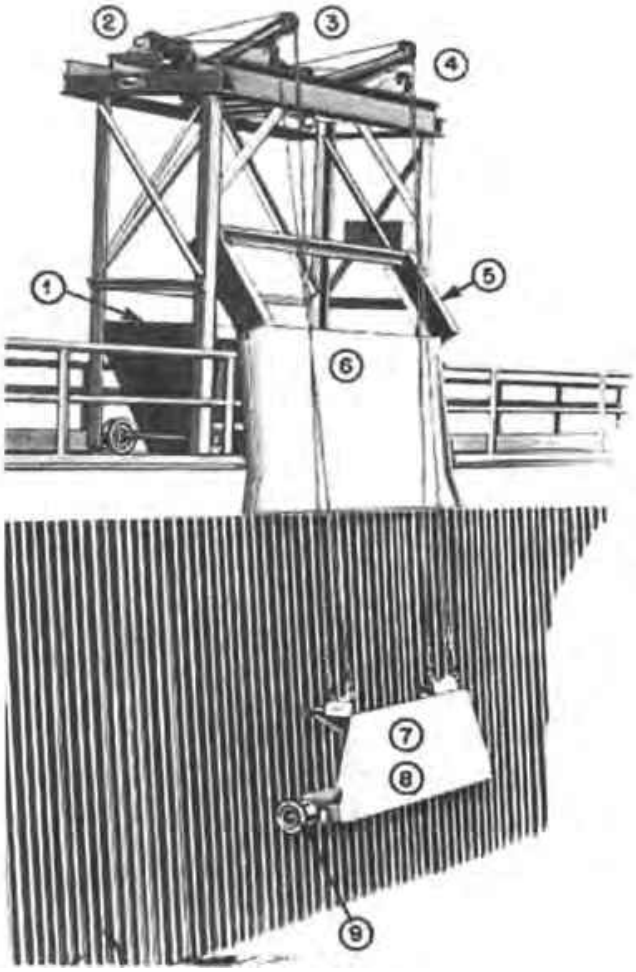


FIGURE 21 Trash rack with raking mechanism: (1) Large enclosed trash hopper contains debris discharged by rake. Hinged door in end of hopper opens wide for debris removal. (2) Heavy-duty, single-drum hoist, push-button controlled with two separate cables—one for carriage, one for rake teeth. (3) Walking beam actuated by hydraulic cylinder controls position of rake teeth. (4) Fixed sheave. Cable operating over stationary sheave raises and lowers rake carriage. (5) Discharge guide flanges assure positive positioning of rake over trash chute prior to dumping. (6) Dead plate, or apron, integral with superstructure guides rake to discharge point—prevents trash from falling off rake prematurely. Design permits operation over 3.5-ft. (1.1-m) high hand rail. (7) Self-centering rack-guided carriages allows rake to ride over obstructions in water during lowering cycle. Debris of all types picked up on lifting cycle rather than forced to bottom of channel. (8) Rake mechanism assures positive removal of debris with maximum carrying capacity. Hydraulic relief valve provides automatic overload relief. Teeth automatically open if overload occurs, permitting load to drop off rake. No cable failures due to overload. (9) Wide, flanged rollers ride on at least two rack bars (Envirex, Inc., a Rexnord Company)

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general, will call for the bar spacing to be in proportion to the size of the pump. A pump manufacturer can determine the maximum size sphere a pump will handle, and the bar spacing should be limited to 50% of that value. The size of the bar, the lateral distance between supports, and the pier spacing will influence the rate of debris accumulation and the allowable design differential head.

Rotating screens (Figure 22) will remove trash of a much smaller size because the accumulation is continuously removed and the open area is kept uniform. Finer screening than that required by the pump may be necessary in installations where the liquid pumped must pass through small openings in equipment serviced, such as condenser tubes or spray nozzles. Screens are usually installed in conjunction with trash racks so large, heavy pieces will not have to be handled by the screens. Because velocity through the screen is limited the pit cross section may be determined by screen requirements. The velocity through the screen can have maximum values up to 2 ft/s (0.6 m/s) if the amount of trash is limited. High amount of trash and/or environmental considerations require lower velocities. If flow is such that a maximum-width screen available would be too long (deep) for practical or economic reasons, two screens may be employed with a center pier. In this case, the distance to the pump should be increased 50% over single-screen distance. Piers should be rounded (radiused) on the upstream side and ogived (tapered to a small radius) on the downstream side. Any corners at the sidewalls should be faired at small angles to the opening and wall to prevent pockets where eddies can form.

The trash collected by the screens must be disposed of. Traveling screens carry trash up into a hood above the operating floor, at which point a series of spray nozzles wash the trash into a trough leading into a disposal area (Figure 23). The nozzles are supplied by pumps sized for 200 to 300 gpm (45 to 68 m^3/h) with pressures of 60 to 100 lb/in^2 (413 to 690 kPa). These pumps are normally deep-well turbine multistage units. They are located

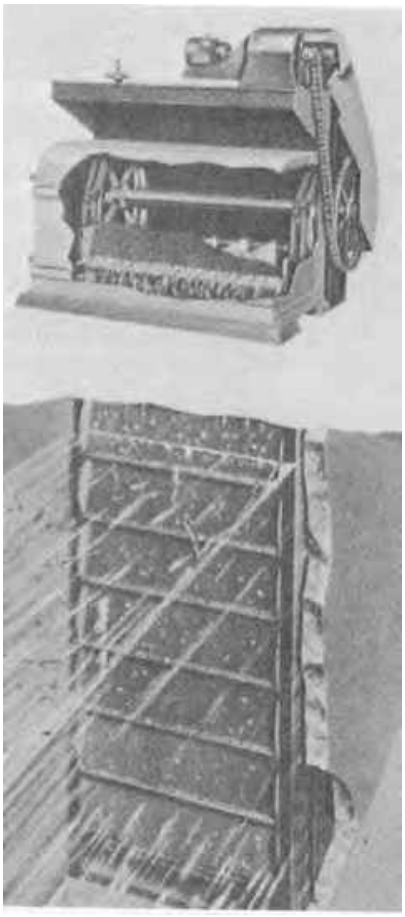


FIGURE 22 Traveling water screen (Envirex Inc., a Rexnord Company)

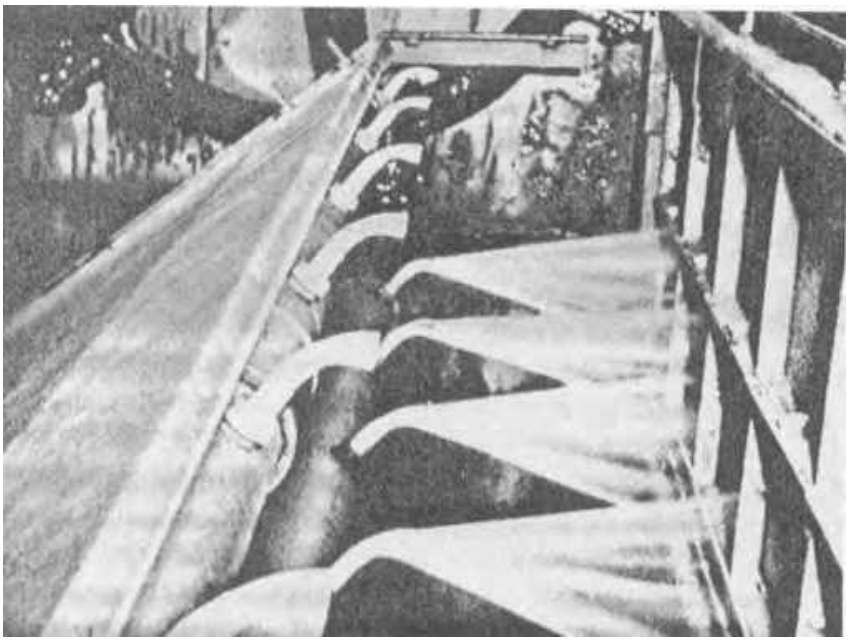


FIGURE 23 Spray nozzle cleaning of baskets on traveling water screens (Envirex Inc., a Rexnord Company)

in the clear well, if possible, close to a wall. If this is not possible, they can be suspended in the circulating water pit, to one side and ahead of the main pumps. Care must be taken that they do not disturb the flow to the larger pumps. Their submergence requirements are usually less than those of the main pumps, and this allows use of the pump setting that gives the least interference with either pump flow.

It is possible to have these pumps also dewater the pit. This will require additional piping and valves and a more careful location of the pump because it will, of necessity, be close to the bottom of the pit. A small chamber off the main pit, located far enough from the main pumps to avoid eddying, will be required.

The direction of the flow from the forebay through the screens and into the pump area should be continuous. Avoid right-angle screens, through which flow must change direction at least once and possibly twice. If screens must be at an angle to the flow into the pumps, increase the screen-to-pump distance by 100%. Environmental considerations may increase the possibility of problems in this area.

A wet-pit source may never be quite free from some degree of silt or sand. After it is accepted by the pump, this liquid goes into a piping system, usually one with other pumps involved. The original pit pump may also be a dry-pit type. In either case, if the pumps of a system are to operate for long periods of time with minimum maintenance, wearing parts must be protected by fine screens or strainers.

In any screening operation, it is essential to remember that if the screen is necessary and does a good job, it will plug up and thus requires cleaning. In a wet pit, therefore, neither the suction bell of a wet-pit pump nor the suction bell going to a dry-pit pump should be basket-screened. Collection of debris will soon increase inlet velocity to unmanageable proportions, and the pump operation will suffer.

It follows that if the basic screening discussed previously does not provide liquid clear enough to be pumped, additional strainers must be put into the system. The pump manufacturer usually specifies the need for such strainers, and the degree of particle reduction. In some instances, the system may require water as pure as it is possible to produce, as in boiler-feed service. For any situation, a variety of strainers are available.

Drum rotating strainers (Figure 24) can handle solids as small as approximately 0.01 in (250 μm). Backflushing is used to renew the strainer area. The addition of woven wire cloth to such strainers may remove materials down to 0.001 in (25 μm). Woven wore drum strainers used in conjunction with sand filters will reduce the amount of backflushing required for the sand filters.

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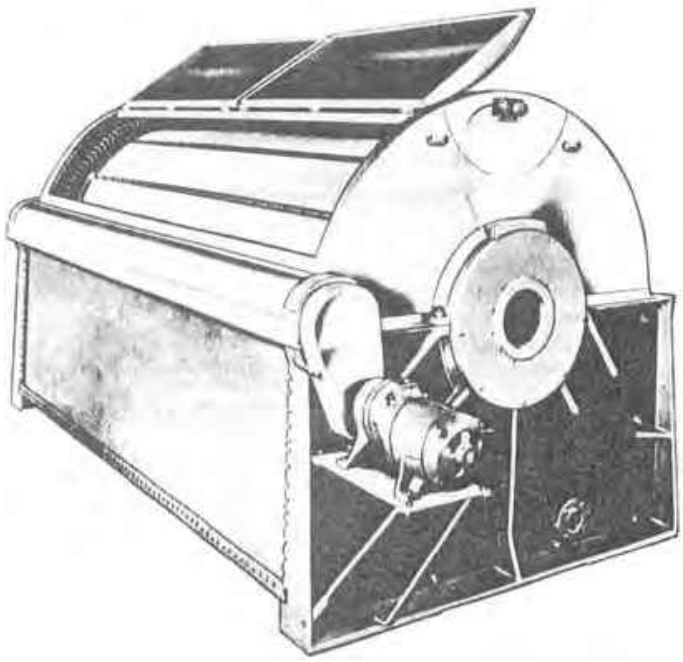


FIGURE 24 Drum filter (Green Bay Foundry and Machine Works)

Where a finer degree of filtration is required, a line filter is used. Single flow filters require system shutdown to remove and clean baskets, but either a motorized automatic type of single flow filter (Figure 25) or twin filters with alternate flow by valve arrangement (Figure 26) permit uninterrupted system operation. When a particularly difficult situation exists, a battery of filters in series may be used.

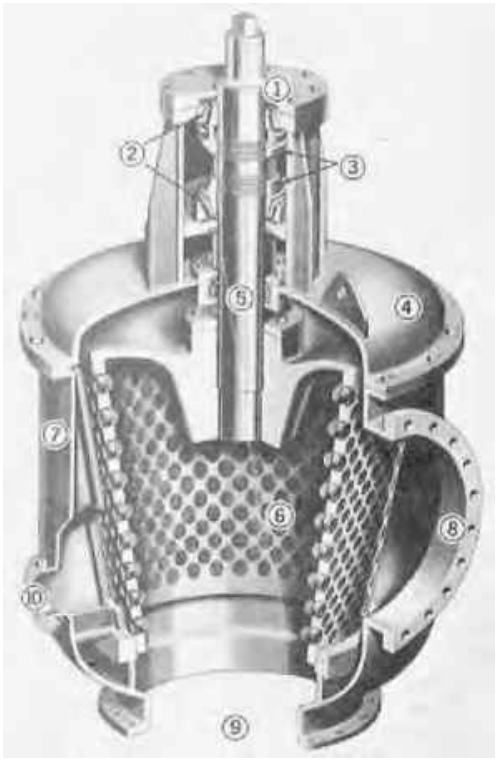


FIGURE 25 Single flow strainer with provision for backwashing without interrupting service: (1) strainer drive, (2) strainer drum support bearings, (3) lock nuts for drum adjustment, (4) cover, (5) shaft, (6) drum—tapered for adjustment, (7) body, (8) inlet, (9) outlet, (10) backwash opening (S. P. Kinney Engineers)

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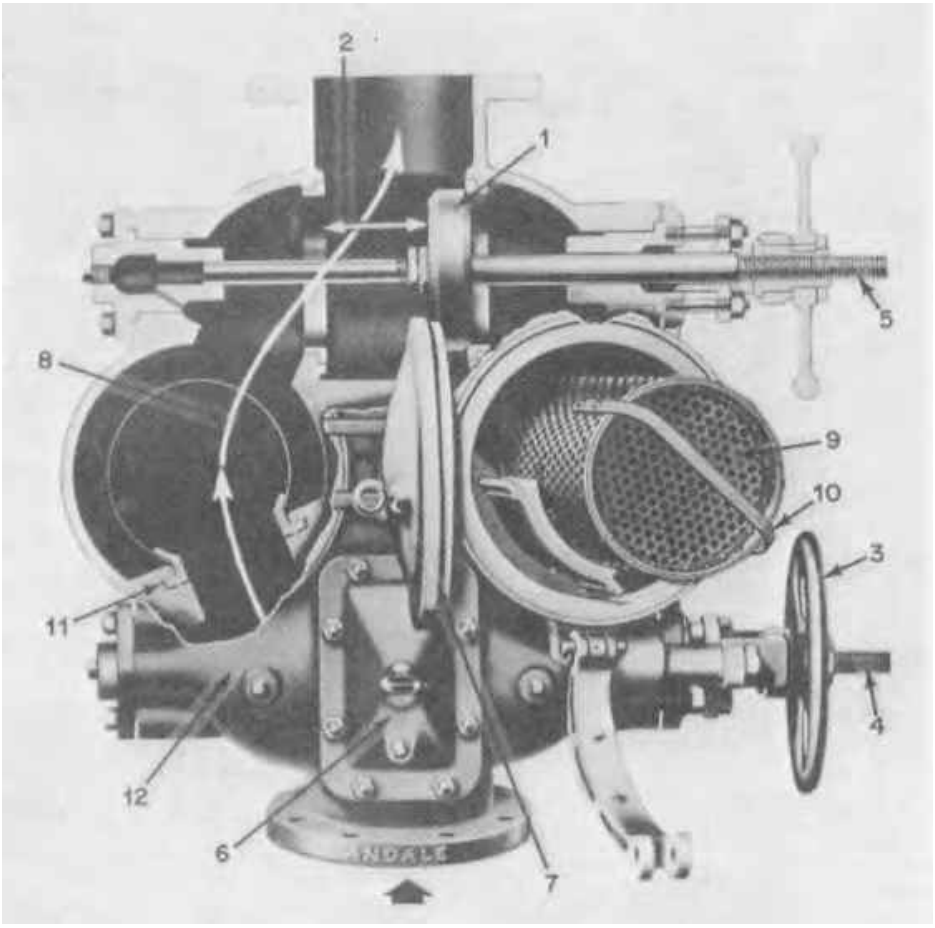
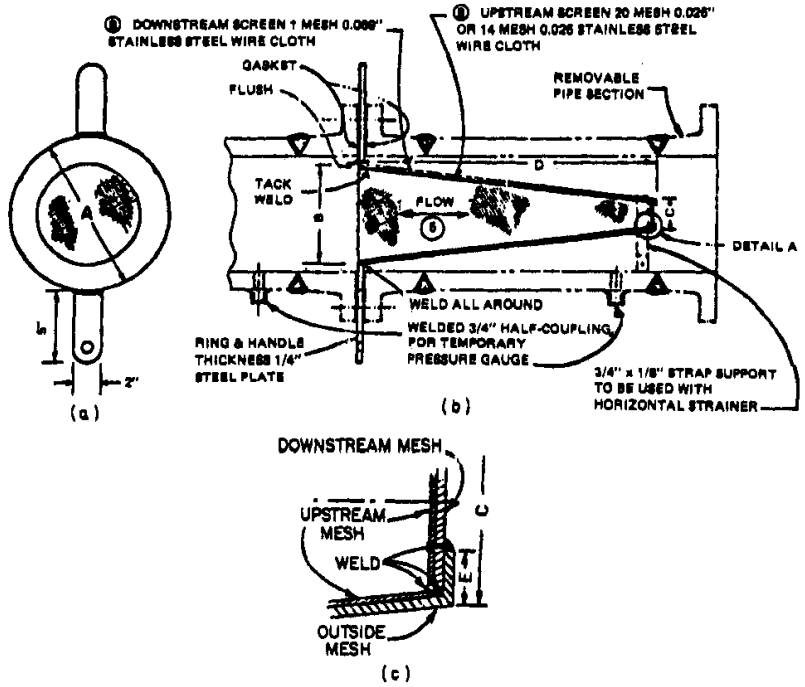


FIGURE 26 Duplex basket strainer. Flow through one side allows cleaning of opposite basket while line is still in service. (1) positive flow diversion, (2) rapid switchover, (3) free-turning handwheels, (4) visual indication of valve position, (5) external acme threads, (6) internal access, (7) cover and clamp(s), (8) streamlined flow, (9) basket, (10) basket handle, (11) basket, (12) one-piece body construction (Andale)

For areas where wear can be extremely critical, such as bearings and stuffing boxes (mechanical seals), additional filtering is necessary, especially if the entrained silt is abrasive. A vortex filter using centrifugal flow action will remove particles down to a few micrometers. These filters have continuous sediment removal and should not require any maintenance (Section 2.2.1, Figure 78).

Some systems, such as boiler feedwater, are "closed" after they are in an operating cycle. Chemical and mechanical cleaning components are designed into the system to obtain a high degree of water purity. Before the system is ready to operate, however, the piping, tanks, valves, and so on will have a residue of weld spatter, metal chips, and other debris from the fabrication process. Temporary strainers are used in the suction lines of boiler feed pumps and condensate pumps to gather up this debris. These strainers are in-line fabricated metal mesh of either a cone or box type. Feed pumps are most often equipped with cone strainers, pointing preferably upstream (Figure 27). The length and diameter are set to conform to the suction pipe size and the mesh so the velocity does not become excessive. These strainers, set between flanges in a pipe section, must be removed and cleaned whenever the differential pressure reaches 2 to 3 lb/in^2 (15 to 20 kPa). Condensate pump strainers are usually box strainers, a square section directly in front of the suction flange with a square box larger than the pipe diameter laced with egg crate metal mesh (Figure 28). This basket strainer can be lifted out of the box for cleaning as necessary. These strainers for collecting fabrication debris may be removed when the need disappears and the system will remain clean.

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Flange or pipe size	A	B	C	D: 20 mesh 0.025-in wire, 0.025-in opening	D: 14 mesh 0.025-in wire, 0.046-in opening	E
3	5	2-¾	1-½	16	11	⅛
4	6-⅝	3-¾	1-½	16	11	¼
6	8-¼	5	2	18	12	⅛
8	10-¾	7-½	2	20	13	⅛
10	12-¾	9-½	3	23	14	⅛
12	15	11	3-½	24	15	⅛

FIGURE 27 Cone strainer for start-up service in boiler-feed pump system. (a) cross-section, (b) lengthwise section, (c) typical detail A (nose pointing downstream). All dimensions are in inches (1 in = 0.0254 m = 2.54 cm). Strainer ring material is carbon steel. When a strainer has served its purpose, the screen portion is removed, the ID of ring is cut to suit ID of pipe, and the ring is used as a spacer. Cone strainers may be installed with nose pointing either upstream or downstream. The fine mesh screen should always be on the upstream surface of the strainer.

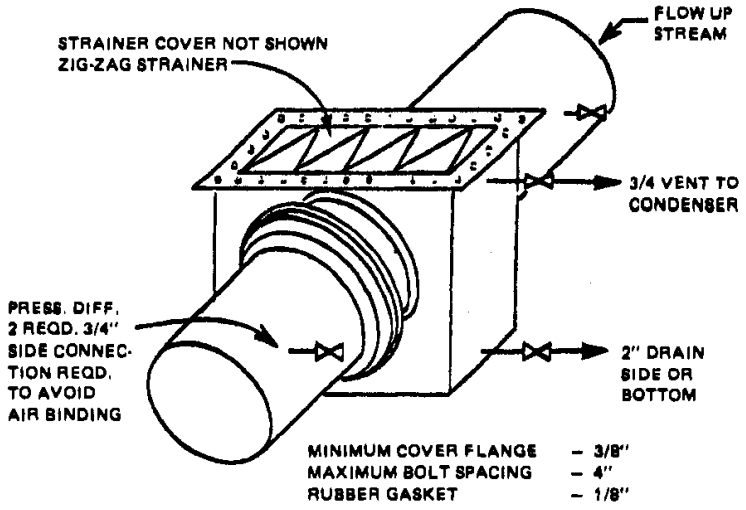


FIGURE 28 Box strainer for temporary start-up service. Pressure differential connections are on top or side only. Upstream mesh is #16 stainless steel with 0.018-in (0.46-mm) wire diameter. Downstream mesh is #2 stainless steel with 0.063-in (1.6-mm) wire diameter (1 in = 25.4 mm).

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SECTION 13.2
INTAKE MODELING

MAHADEVAN PADMANABHAN
JOHANNES LARSEN

PROBLEMS ENCOUNTERED IN PUMP INTAKES

The various hydraulic problems associated with a pump pit include formation of surface and subsurface vortices, prerotation and swirl, and flow separation at or near the suction bell of either wet-pit or dry-pit centrifugal pumps (Figures 1a and 2). Any of these problems can adversely affect pump performance by causing cavitation, vibrations, or loss of efficiency.[1] Usually there is more than a single reason for these problems, and the extent of the combined effects is difficult to predict reliably by mathematical modeling or computational fluid dynamics (CFD). Formation of vortices, for example, even though dependent on suction pipe velocity and submergence, is strongly influenced by added circulation from vorticity sources, such as a nonuniform approach flow resulting from intake and approach channel geometries; rotational wakes shed from obstructions, such as columns or piers; and the velocity gradients resulting from boundary layers at the walls and floor.[2] The circulation contributed by these vorticity sources is unpredictable and strongly dependent on intake design and operating conditions, especially for large pumping units with multiple bays fed by a common approach channel. In these cases, physical modeling is the best way of predicting the behavior of the prototype with a reasonable degree of reliability.

Free Surface Vortices This type of vortex is considered objectionable when it draws air bubbles or an air core into the pump inlet (Figure 1b). Under nominal approach flow conditions, strong air core vortices causing air ingestion with air concentration as high as 10% have been reported.[29] It has been established that an air concentration in the suction pipe of from 3 to 5% can lower pump efficiency.[3] Also, air in the form of large bubbles or slugs can cause the impeller to vibrate. Strong surface vortices that do not draw air are also objectionable if they draw debris into the suction pipes and because their high rotational velocity can reduce the local pressure sufficiently to induce cavitation.

Subsurface Vortices This type of vortex, also known as a submerged vortex (Figures 1c and 2), usually originates from a floor or wall and is induced by vorticity produced in separation zones close to the pump entrance or, in wet-pit pumps, below the bell. The reduced pressure in the vortex core causes fluctuating load on the pump impeller along with associated vibration and noise,[4,5] increased possibility of cavitation, higher inlet losses, and decreased pump efficiency, especially when the core pressure is sufficiently low to release dissolved air or other gases from the fluid.

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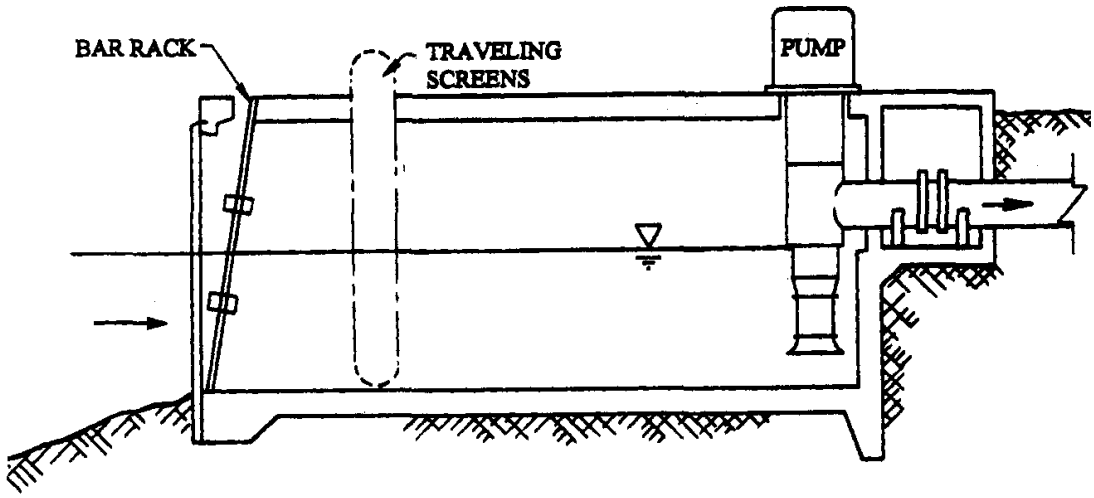


FIGURE 1A Typical vertical wet-pit pump intake

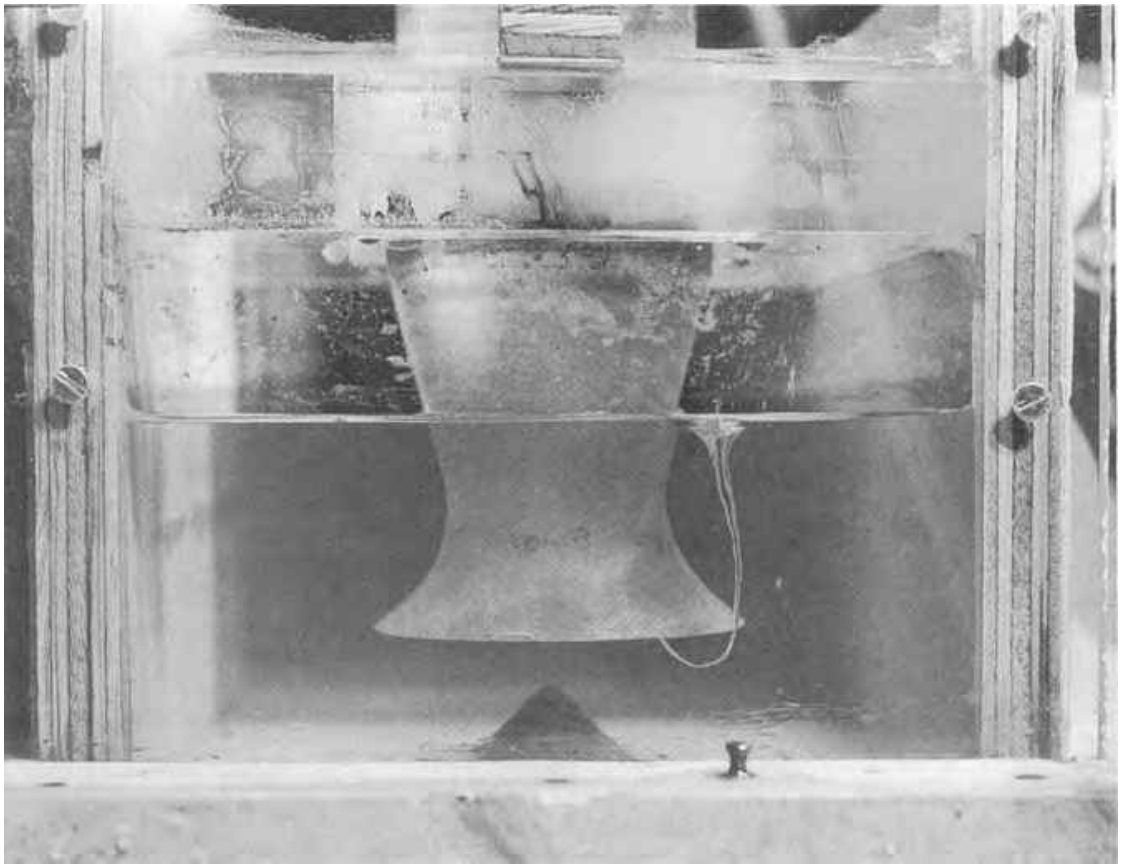


FIGURE 1B Free surface vortex can form at pump suction bell entrance due to combination of low submergence, high suction velocity, nonuniform approach flow, or other vorticity sources (Courtesy of Alden Research Laboratory, Inc.)

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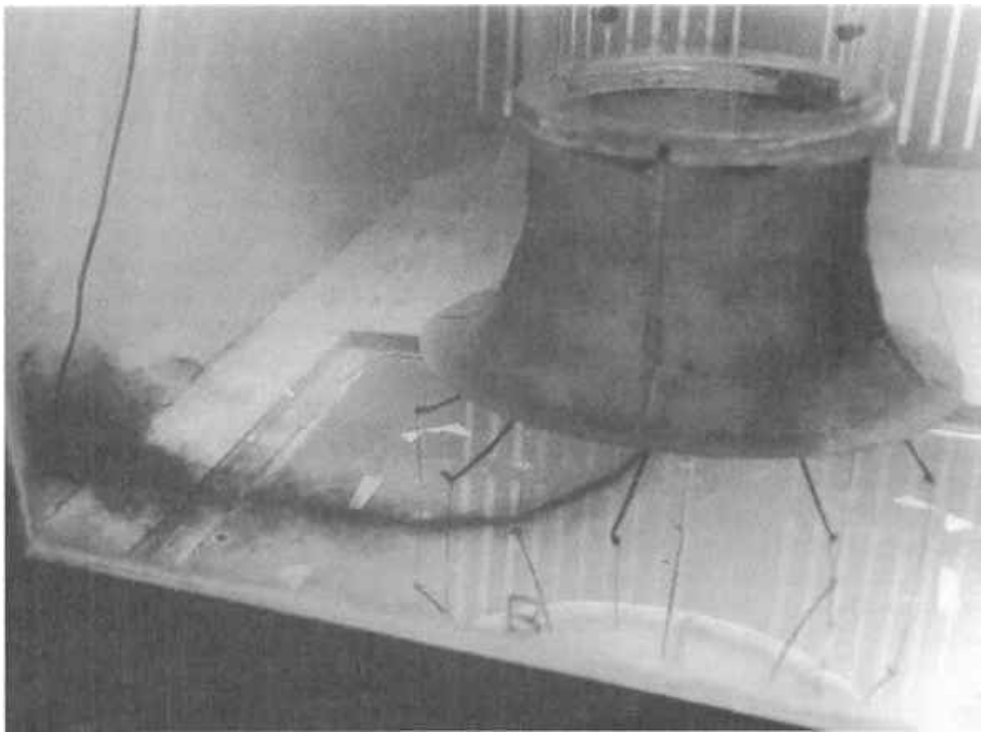


FIGURE 1C Subsurface vortex formed at the side wall due to poor flow guidance to the bell; vortex identified in the model with dye (Courtesy of Alden Resrearch Laboratory, Inc.)

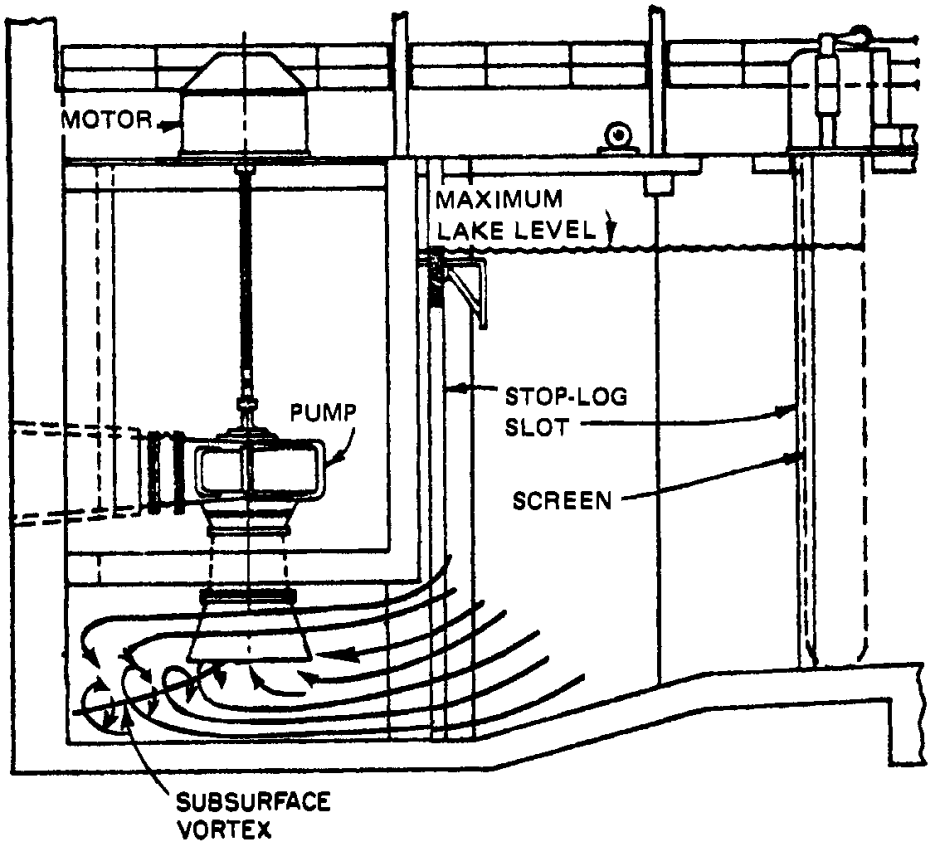


FIGURE 2 Typical vertical dry-pit pump intake. Improper spacing of suction bell relative to floor and back wall caused subsurface vortexing. Model test revealed problem and correction (Reference 4).

Prerotation and Swirl *Swirl* is a general term for any flow condition (due to vortexing or a pipe bend) where there is a tangential velocity component in addition to a usually predominating axial flow component. *Prerotation* is a specific term to denote a cross-sectional average swirl in the suction line of a pump or, in case of a vertical wet-pit pump, upstream of the impeller.

The prerotation angle θ is a measure of the strength of the tangential velocity component u_t relative to that of the axial velocity component u in the flow approaching the pump impeller; that is, $\theta = \tan^{-1}(u_t/u)$. Adverse effects on the pump are decreased capacity and head when the rotation is in the direction of pump rotation and increased capacity and head when the rotation is opposite the pump rotation (antirotation). The increased capacity is associated with an increase in power requirement and may cause motor overheating.

Prerotation will influence pump performance because the flow approaching the impeller already has a rotational flow field that may oppose or add to the impeller rotation, depending on direction. The design of the pump blades (i.e., shape and angle) usually assumes no prerotation, and the existence of prerotation implies flow separation along one side of the impeller blades. The degree of prerotation that should be of concern depends on the type of pump and may not always be known. Prerotation could be quantified in a model by an average cross-sectional swirl angle, determined by detailed velocity measurements, or by readings on a swirl meter. Because swirl decays along a pipe as a result of wall friction, internal fluid shear, and turbulence, the swirl meter in a model suction pipe should be located near the impeller.

Losses Leading to Insufficient NPSH A poorly designed pump intake could result in large inlet losses. Losses caused by screens, poor entrance conditions, vortexing and swirl, and vortex suppression devices may add up to a value so great that the required NPSH of the pump is not satisfied. Increased inlet losses due to swirl have been reported in laboratory studies.[6] In a nuclear reactor residual-heat-removal sump model, inlet losses in a preliminary design wherein air core vortices and a high degree of swirl were present were 20% higher than in a revised design with no strong vortices and swirl, and with similar pipe entrance geometry and flows. Because the degree of vorticity and swirl cannot be predicted and it is therefore not possible to calculate inlet losses reliably; they are usually obtained from model studies. With the experimentally derived values of the inlet losses, the NPSH available should be checked by recalculation.

DESIGN GUIDES

As a preliminary design guide (Section 13.1), published information may be used to establish pump sump dimensions and a minimum desired submergence.[7–10] Figures 1 and 2 of Section 13.1 show the basic layout of a pump sump and present typical dimensions in relation to the flow required per pump, as published by the Hydraulic Institute.[7] Some of these references may express sump dimensions in terms of bell mouth diameter, and Figure 3 shows the typical relationship between design flow for a given wet-pit pump and the bell mouth diameter required to achieve reasonable and desirable velocities approaching the impeller. One should be aware that the sump dimensions and minimum derived submergences given by design guides are considered applicable for the ideal condition of a simple, straight approach flow with a constant, low approach velocity to the pump sump.

The need for a physical model study still exists when site or operating conditions make the ideal condition impossible. For example, for a particular pump intake configuration, a hydraulic model study indicated that a strong submerged floor vortex existed with floor clearance of 0.5 times the bell diameter (a usual design value) and that a reduced floor clearance of 0.3 times the bell diameter eliminated the vortex. Typically, for pump intakes of the following types, model studies should be considered essential:

1. Intakes with nonsymmetric approach flow; for example, an offset in the approach channel

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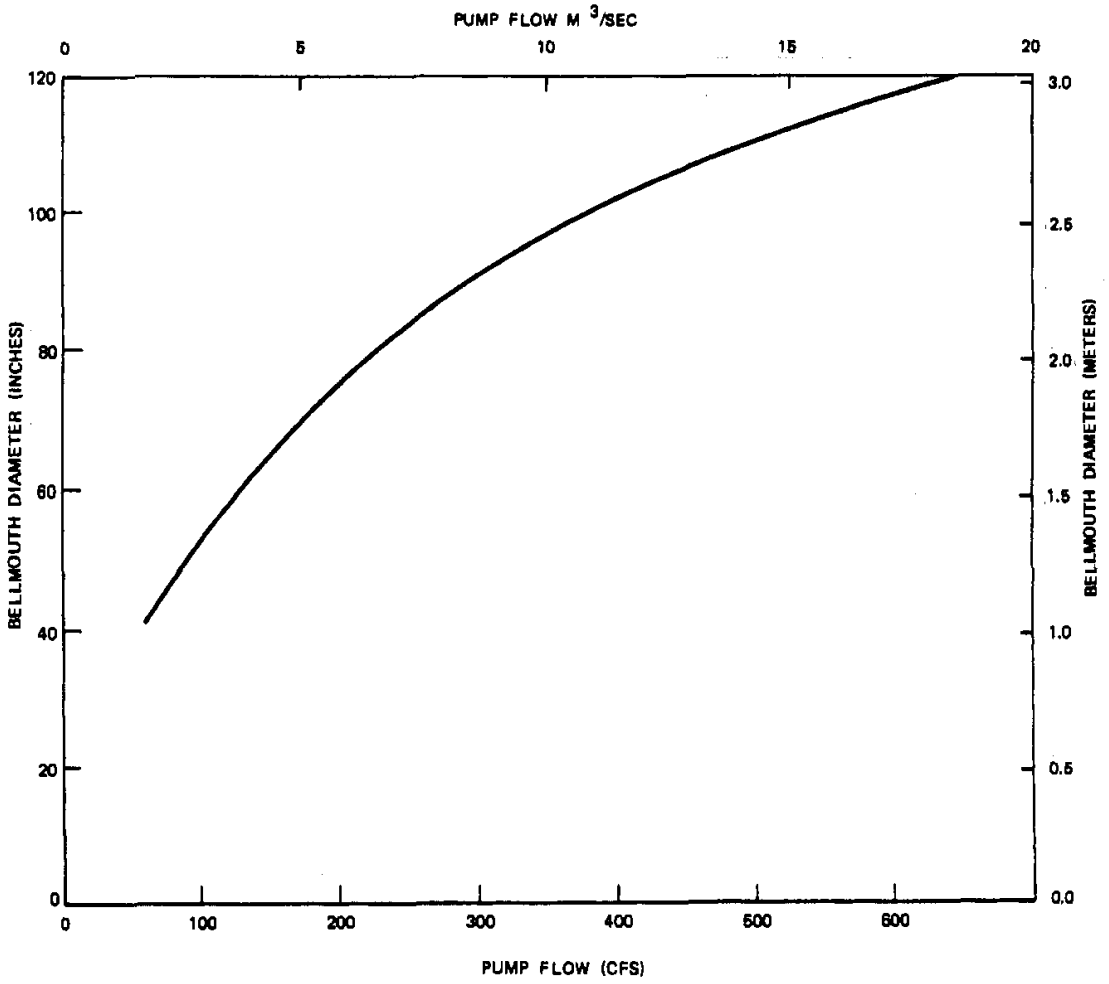


FIGURE 3 An approximate relationship between design flow and required suction bell diameters for vertical propeller pumps for preliminary evaluations

2. Intakes with multiple pump bays with a common approach channel and a variety of pump operating combinations
3. Intakes with pumps of capacities greater than 40,000 gpm (2.5 m^3/s) per pump
4. Intakes with expanding approach channel
5. Intakes with possibilities of screen blockages or obstructions close to suction pipe entrance; for example, reactor containment recirculation sumps, gate guides for dry-pit pumps
6. Dual flow screens intake

For items 1, 2, 4, 5, and 6, a model study is recommended because of the unknown effects a nonuniform approach flow can have on vortexing and swirl. For item 3, considering the cost of large pump installations and the cost of backfit, should problems occur, a model study is recommended to ensure proper pump operation.

MODEL SIMILITUDE

The principle of dynamically similar fluid motion forms the basis for the design and operation of hydraulic models and the interpretation of experimental data. The basic concept

of dynamic similarity is that two systems with geometrically similar boundaries have similar flow patterns at corresponding instants of time.[11,12] To achieve this, all individual forces acting on corresponding fluid elements must have the same ratios in the two geometrically similar systems. The condition required for complete similitude may be developed from Newton's second law of motion:

$$F_i = F_p + F_g + F_v + F_t \tag{1}$$

where F_i = inertial force, defined as mass m times acceleration a
F_p = pressure force connected with or resulting from the motion
F_g = gravitational force
F_v = viscous force
F_t = surface tension force

Additional forces, such as fluid compression, magnetic, or Coriolis forces, may be relevant under special circumstances, but generally these forces have little influence and are, therefore, not considered in the following development.

Equation 1 can be made dimensionless by dividing all the terms by F_i. Two systems of different size that are geometrically similar are dynamically similar if both satisfy the same dimensionless form of Eq. 1. We may write each of the forces of Eq. 1 as

F_p = net pressure difference × area = $\alpha_1 \Delta p L^2$
F_g = specific weight × volume = $\alpha_2 \gamma L^3$
F_v = shear stress × area = $(\alpha_3 \mu \Delta u/\Delta y)$ (area) = $\alpha_3 \mu u L$
F_t = surface tension × length = $\alpha_4 \sigma L$
F_i = density × volume × acceleration = $\alpha_5 \rho L^3 u^2/L = \alpha_5 \rho u^2 L^2$

where

α_1, α_2, and so on = proportionality factors
Δp = net pressure difference
L = representative linear dimension
γ = specific weight = ρg
μ = dynamic viscosity
$\Delta u/Dy$ = depth-wise velocity gradient
u = representative velocity
σ = surface tension
ρ = density
g = acceleration due to gravity

Noting that the kinematic viscosity, ν, is given by μ/ρ, substituting the above terms in Eq. 1 and making it dimensionless by dividing by the inertial force, we obtain

$$\frac{\alpha_1}{\alpha_5} E^{-2} + \frac{\alpha_2}{\alpha_5} F^{-2} + \frac{\alpha_3}{\alpha_5} R^{-1} + \frac{\alpha_4}{\alpha_5} W^{-1} = 1 \tag{2}$$

where

$$E = \frac{u}{\sqrt{\Delta p/\rho}} = \text{Euler number} \propto \frac{\text{inertial force}}{\text{pressure force}}$$

$$F = \frac{u}{\sqrt{gL}} = \text{Froude number} \propto \frac{\text{inertial force}}{\text{gravitational force}}$$

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$$R = \frac{uL}{\sqrt{\mu/\rho}} = \text{Reynolds number} \propto \frac{\text{inertial force}}{\text{viscous force}}$$

$$W = \frac{u^2L}{\sigma/\rho} = \text{Weber number} \propto \frac{\text{inertial force}}{\text{surface tension force}}$$

Because the proportionality factors α_1, α_2, and so on, are the same in model and prototype, complete dynamic similarity is achieved if the values of each dimensionless group, E, F, R, and W are equal in model and prototype. In practice, this is difficult to achieve. For example, to have $F_{\text{model}} = F_{\text{prototype}}$ and $R_{\text{model}} = R_{\text{prototype}}$ requires either a 1:1 'model' or a fluid of very low kinematic viscosity in the reduced-scale model. Hence, the accepted approach is to select the predominant force and then design the model according to the appropriate dimensionless group. The influences of the other forces become secondary and are called scale effects.[11,12]

Froude Scaling Pump intake models are generally designed and operated using Froude similarity because the flow is controlled by gravitational and inertial forces. The Froude number is, therefore, made equal in model and prototype:

$$F_r = F_m/F_p = 1 \tag{3}$$

where m, p, and r denote model, prototype, and ratio between model and prototype.

In modeling a pump intake sump to study the formation of vortices, it is important to select a reasonably large geometric scale to achieve large Reynolds numbers. At a large Reynolds number, energy loss coefficients usually behave asymptotically with Reynolds number. Hence, with $F_r = 1$ and a sufficiently high Reynolds number, the Euler number E will be equal in model and prototype. This implies that flow patterns and loss coefficients may be considered similar in model and prototype. From Eq. 3, the velocity, flow, and time scales are

$$u_r = L_r^{0.5} \tag{4}$$

$$Q_r = L_r^2 u_r = L_r^{2.5} \tag{5}$$

$$t_r = L_r^{0.5} \tag{6}$$

For example, if a model-to-prototype scale of 1:10 is used, a prototype velocity of 1.0 ft/s (0.3 m/s) becomes a model velocity of $(1 \div 10)^{1/2} = 0.32$ ft/s (0.09 m/s). In the model, all physical dimensions of the pump bays and approach channel are scaled to the ratio 1:10. Submergence, being a linear dimension, is also scaled to 1:10. A flow of 100 gpm (0.006 m^3/s) in the model corresponds to a flow of $100 \times 10^{2.5} = 31{,}623$ gpm (2 m^3/s) in the prototype. Similitude parameters and laws are treated in detail in References 11 and 12.

Similarity of Vortices The fluid motions involving vortex formation in pump sumps have been studied by several investigators. It can be shown by principles of dimensional analysis that the dynamic similarity of fluid motion that could cause vortices at an intake is governed by the following dimensionless parameters:

$$\frac{ud}{\Gamma}, \frac{u}{\sqrt{gd}}, \frac{d}{s}, \frac{ud}{\nu}, \text{ and } \frac{u^2d}{\sigma/\rho}$$

where u = average axial velocity at the bell entrance
Γ = circulation contributing to vortexing
d = diameter of the bell entrance
s = submergence at the bell entrance
ν = kinematic viscosity of water
g = acceleration due to gravity

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σ = surface tension of water air interface

ρ = water density

The influence of viscous effects is defined by the parameter ud/ν = R, the Reynolds number. Surface tension effects are indicated by $u^2d\rho/\sigma$ = W, the Weber number. As strong air-core type vortices, if present in the model, would have to be eliminated by a modified sump design, the main concern for interpretation of model performance involves the similarity of weaker vortices. If the influence of viscous forces and surface tension on vortexing is negligible, dynamic similarity is obtained by equating the parameters ud/Γ, $u/\sqrt{gd}$, and d/s in model and prototype. A Froude model satisfies this condition, provided the approach flow pattern in the vicinity of the sump, which governs the circulation, Γ, is properly simulated.

Considerable research on scaling free surface and submerged vortices has been conducted in the past few years. From a study of horizontal outlets for a depressed sump, it was determined that for pipe Reynolds numbers above 7×10^4, no scale effect on vortex strength, frequency, or air withdrawal existed.[30] Another study indicated that an inlet Reynolds number of 3×10^4 is sufficient to obtain a good model to prototype correlation of vortices.[6] Surface tension effects on vortexing have been shown to be negligible for Weber numbers greater than 120 based on laboratory experiments.[14]

A review of all available data on model versus prototype vortex intensity indicated negligible scale effects for weak vortices and small scale effects for air drawing vortices, and that this effect could be overcome by a relatively small increase in model flow rate.[27] The model flow rate should only be increased by an amount such that sufficient Reynolds and Weber numbers result. Excessively increasing the model flow, particularly to prototype velocity, produces highly exaggerated vortices incompatible with prototype observations.

DESIGN AND OPERATION OF MODELS

Model Scale Scale effects are less as model size increases but construction and operation costs increase with model size, so a compromise must be made. In general, the formation of vortices, both free surface and submerged, is highly responsive to approach flow patterns, and it is important to select a geometric scale that achieves Reynolds numbers large enough to keep the flow turbulent and to meet the fluid mechanic criteria for minimizing scale effects.[13] Also, one should consider other factors such as access for instruments, accurate flow measurements, and ease of modification in selecting a proper scale.

Information on preferred minimum values of Reynolds number and Weber number, discussed earlier, may be used in designing a model and deciding geometric scale. However, adhering to these limits does not, in itself, guarantee negligible scale effects in a Froude model because these limits are based on tests run under ideal laboratory conditions. In real situations, there is usually more than one source of vorticity generation of unknown extent, and a generalization of scale effects for all cases would be inappropriate. To compensate for such unknown scale effects, a common practice is to test a model at higher-than-Froude scaled flows.

A special test procedure involving high temperatures may be used to determine any scale effects and to project the model results to prototype ranges of Reynolds numbers.[2] The water temperature in the model is varied over a range, say 50 to 120°F (10 to 49°C), and flow velocities in the pipes are varied over a range of values, if possible, up to the prototype velocities. Vortexing and other flow patterns over a range of Reynolds numbers are obtained from these tests and can be used to evaluate any possible scale effects. A prediction of the prototype performance can be made based on these tests.

Extent of Model It is very important to include a sufficient length of approach channel in the model because approach flow nonuniformities contribute greatly to vortex formation and swirl. The decision on what approach length should be included is usually based

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on the experience and engineering judgment of the model designer. The requirement is essentially the proper simulation of approach velocity profile to the intake and then on to the pump bays. In some cases, considering the cost and time involved in including a sufficient channel length in a pump intake model, a separate, smaller model can be built to determine the approach flow patterns to the intake and these flow patterns then simulated in the pump intake model.[18]

Figure 4a shows a 1:10 geometric scale model of a four-bay pump intake, with each pump designed to draw 140,000 gpm (312 cfs or 8.83 m^3/s) flow. Figure 4b shows a 1:18 geometric scale model pump intake for a flood control project involving several low head high flow pumps—each handling 360,000 to 450,000 gpm (800 to 1,000 cfs) at 12 ft head (22.68 to 28.35 m^3/s at 3.65 m head).

Modeling of Screens and Gratings In addition to providing protection from debris, screens suppress nonuniformities of approach flow. The aspects of flow through screens that are of concern in a model study are (1) energy loss in the fluid passing through the screen, (2) modification of the velocity profile, and (3) production of turbulence. As all these factors could affect vortex formation in a sump with approach flow directed through screens, a proper modeling of screen parameters is important.

The fluid passing through the screen loses energy at a rate proportional to the drop in pressure, and this loss dictates the effectiveness of the screen in altering velocity profiles. The pressure drop across the screen is analogous to the drag induced by a row of cylinders in a flow field and can be expressed in terms of a pressure drop coefficient K (or, alternately, a drag coefficient), defined as[19]

$$K = \Delta p/(0.5\rho u_a^2) = \Delta H/(u_a^2/2g) \tag{8}$$

where Δp = drop in pressure across screen
u_a = mean velocity of approach flow to screen
ΔH = head loss across screen

The loss coefficient is a function of three variables: (1) screen pattern, (2) screen Reynolds number $R_s = u_a d_w/\nu$, where d_w is the wire diameter of the screen, and (3) solidity ratio S', the ratio of closed area to total area of screen (Section 11.1).

If S' and the wire mesh pattern are the same in the model and prototype screens, the corresponding values of K are a function of R_s only. This is analogous to the drag coefficient in a circular cylinder. At values of R_s greater than about 1000, K becomes practically independent of R_s.[19] However, for models with low approach flow velocity and fine wire screens, it is necessary to ascertain the influence of R_s on K for both model and prototype screens before selecting screens for the model that are to scale changes in velocity distribution.

Velocity modification equations relating the upstream and downstream velocity profiles usually indicate a linear relationship between the two, the shape and solidity ratio of the screen, and the value of K.[20] If wire shapes and solidity ratios are the same in model and prototype, it is possible to select a suitable wire diameter to keep the values of K approximately the same for the model and prototype screens in the corresponding Reynolds number ranges. This produces a head loss across the model screen that is scaled to the geometric scale of the model and that produces identical velocity modifications in model and prototype. Some model designers consider it a conservative approach to leave out the screens in the model altogether, under the assumption that this omission will only worsen the nonuniformity of the approach flow. This approach is not recommended unless it is not practically possible to select an appropriate model screen.

Modeling of Pumps The exterior submerged surfaces of the pump bowl assembly and, for wet-pit pumps, the column including the bell mouth must be modeled to scale as well as the interior geometry from the bell mouth perimeter to the impeller eye. This is to ensure that flow patterns approaching the impeller are properly simulated.

Prerotation induced by the pump's rotating element is discussed in Sections 2.1.3 and 13.1 and in References 21 to 24. It has been shown that the rotating element does not affect upstream flow patterns when the pump is operated at design flow, and hence,

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FIGURE 4A A 1:10 geometric scale model of a four pump-bay pump intake (Courtesy of Alden Research Laboratory, Inc.)

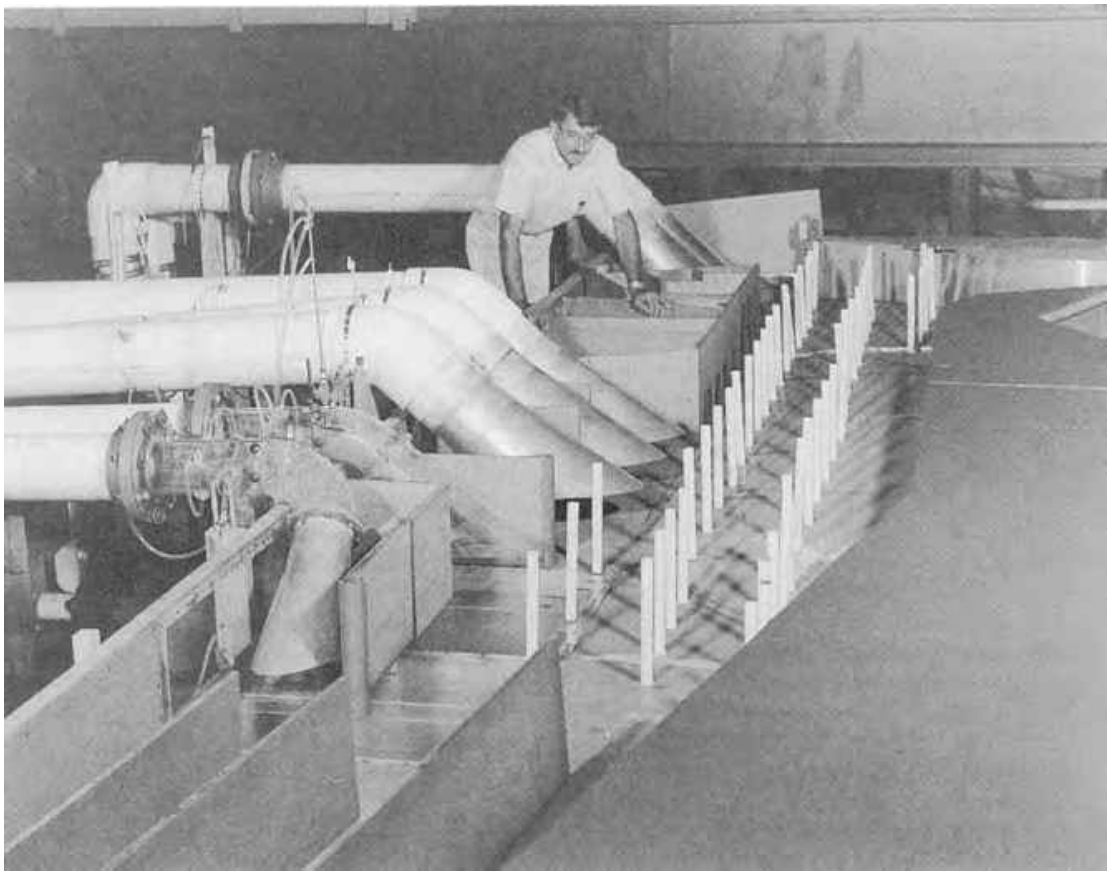


FIGURE 4B A 1:18 geometric scale model of a flood control pumping station with high capacity (800 to 1000 cfs/22.7 to 28.3 m^3/s), low head (about 12ft/3.65 m) pumps (Courtesy of Alden Research Laboratory, Inc.)

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including the rotating element in a pump intake study is not necessary. When the pump is operated at less than rated flow, a degree of swirl is induced upstream of the rotating element. This swirl increases rapidly at flow rates less than 45% of rated flow as reversed flow out of the impeller intensifies, and this may affect vortex activity in the pump well and flow distribution to the pump.

Model Operation Operating a model at the prototype suction pipe velocity is thought to be a conservative method to compensate for excessive viscous energy dissipation and the consequent less intense model vortices in a Froude model.[17] This method is often referred to as the Equal Velocity Rule. Operating a model at a higher than Froude scaled velocity should be considered a reasonable procedure for evaluating scale effects. However, operating a model based on the equal velocity rule may not be advisable unless the model is large enough, say at least a 1:4 scale model, because increasing the flow to many times the Froude scaled flow while keeping a scaled submergence could distort the approach flow patterns and turbulent intensities and, thus, cause unrealistic results. In general, a velocity increase to about 1.5 times the Froude scaled velocity can be considered reasonable. More appropriately, the information contained in References 25 and 26 may be used to decide the velocity ratio for exaggeration, which often is considered a function of model scale. If the final recommended design does not show any coherent core vortices in a model, no large scale effects should be expected, and operating the model at higher than Froude scaled flows may be unnecessary in such cases.[27]

Model Cost The cost of model studies varies considerably and is dependent on such factors as number of pump bays, number of operating conditions, and complexity of approach flow. Typically, $40,000 to $90,000 (in 1999 U.S. dollars) can be expected to cover the range from simple one- or two-bay sumps to multibay installations with complex approach flow and several operating modes.

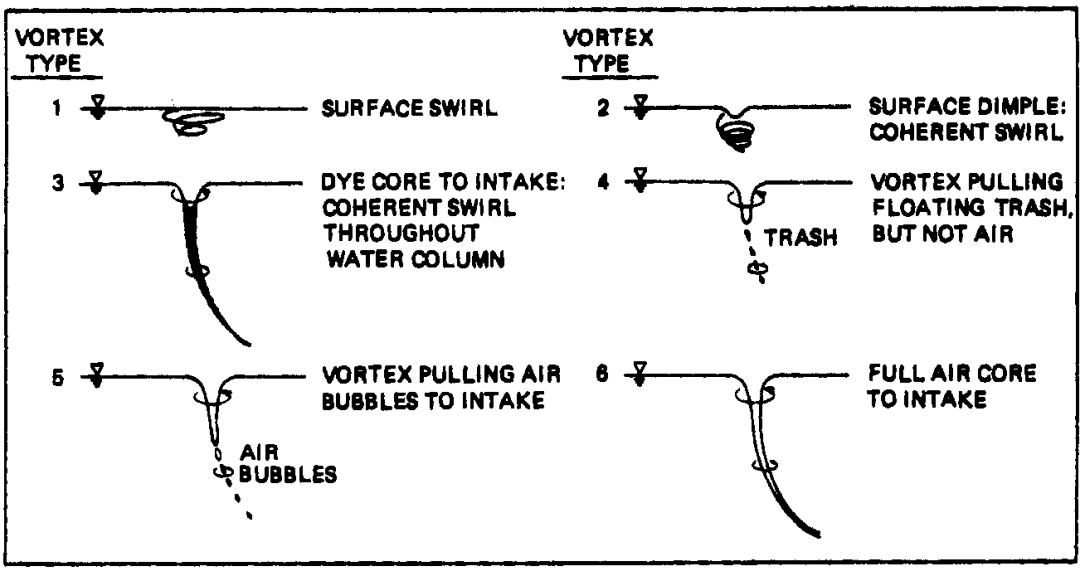
MODEL OBSERVATION TECHNIQUES

Free Surface Vortices Vortices in a model usually contain little energy, and therefore, vortex circulation cannot readily be measured because any measuring device will affect the strength of the vortex. Therefore, vortex strength is best characterized by visual observation and classified by comparison with a vortex strength scale, such as the one shown in Figure 5a. Observations are carried out continuously for a representative period of time, usually one hour prototype, and the resulting data are analyzed in terms of the maximum observed vortex strength and in terms of persistence; that is, the percentage of time a given vortex strength occurred.[28] Vortices of Type 5 and 6 are clearly objectionable in an intake design because of the effect air entrainment and localized pressure reductions have on pump performance. Usually in model studies, a vortex type of 3 is considered the maximum allowable, as prototype strength may be slightly underestimated as a result of possible scale effects.

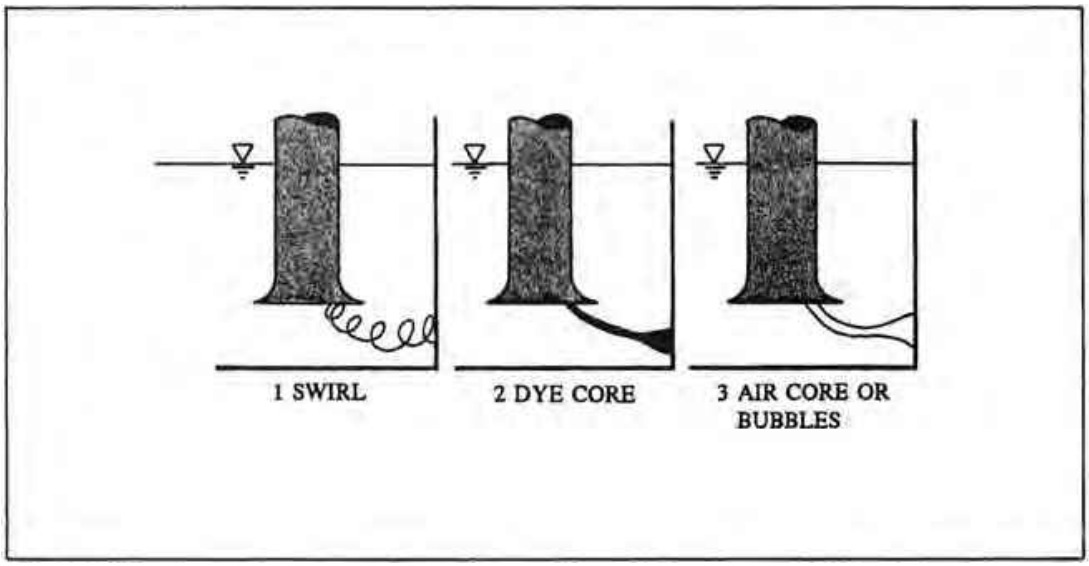
Subsurface Vortices Vortices attached to fixed boundaries below the surface can be difficult to detect and may require a careful search with a dye tracer or an air-injection technique. The vortex strength is best characterized by visual observations and classified by comparison to a vortex strength scale such as one shown in Figure 5b. A strong subsurface vortex may reduce the pressure locally within its core sufficiently low to allow dissolved air to come out of solution resulting in an air core (Type 3 in Figure 5b).

Submerged vortices are most commonly found below the center of wet-pit pumps and around splitter plates introduced to even out flow into the bell mouth. Pressure measurements made with electronic pressure transducers located in the floor under the pump may provide information on rapidly forming and disappearing vortices that cannot be detected with flow visualization. Subsurface vortices should, as far as practical, be eliminated from any design because their presence imposes fluctuating loads on the impeller and the resulting low-pressure areas may cause local cavitation damage to the pump.

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a. FREE-SURFACE VORTICES



b. SUBSURFACE VORTICES

FIGURE 5A and B Vortex strength scale (Courtesy of Alden Research Laboratory, Inc.)

Prerotation The tangential velocity component may be measured by a swirl meter, or vortimeter, which is an impeller with zero pitch in the axial direction mounted axially in the pump column two to four column diameters downstream from the suction bell. A typical vortimeter is shown in Figure 6. Vortimeter rotation is measured by either electronic or manual counting of rotation over a given time. Prerotation can also be determined from velocity traverses obtained with a two-dimensional pitot tube.

Intake Loss Coefficient Water manometers connected to piezometric taps along the pipe are used to measure the hydraulic gradient along the suction pipe. An intake loss coefficient is then determined by extrapolating the measured hydraulic gradient to the suction pipe inlet and computing the head loss $h_L = \Delta h - u^2/2g$, where Δh is the difference

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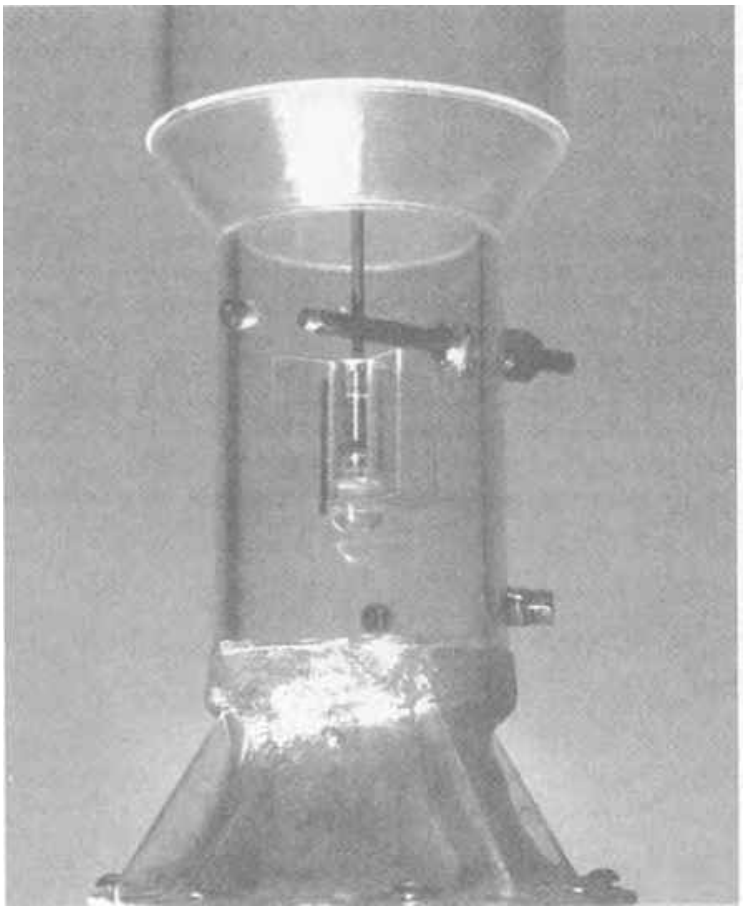


FIGURE 6 Swirl meter in a model pump column (Courtesy of Alden Research Laboratory, Inc.)

between the pump pit water surface and the extrapolated pressure gradeline at the pipe inlet. With this procedure, the computed head losses will not include any pipe frictional losses and consequently any model pipe Reynolds number influence is avoided. At the same time, pressure measurements are done far downstream from the region of flow separation near the entrance, and hence, more accurate measurements of total pressure loss, including entrance losses, are obtained. The loss coefficient C_L is usually defined as $C_L = h_L/(u^2/2g)$, where $(u^2/2g)$ represents the velocity head in the pipe. The computed model loss coefficient is valid for the prototype, and hence, the prototype inlet losses are C_L times the prototype pipe velocity.

Flow Distribution Radial flow under the perimeter of the pump bell is required to avoid vibrations and loss in efficiency caused by uneven impeller loading. Flow direction is usually indicated by six to eight yarn streamers equally spaced and mounted on wire supports 0.2 bell mouth diameter below the bell mouth perimeter. Figure 7 shows a typical installation. For an acceptable design, these streamers should not deviate more than 10 to 15° from radial. Velocity distribution approaching the pump is usually measured with miniature propeller meters.

CORRECTIVE MEASURES

Problems in pump sumps such as severe vortexing, intense swirl, or uneven flow distribution at the bell can be reduced or eliminated by deriving proper corrective methods using models. Because no single remedial method to correct these problems exists, it is important to try several suitable methods in the model to derive an effective and practical one.

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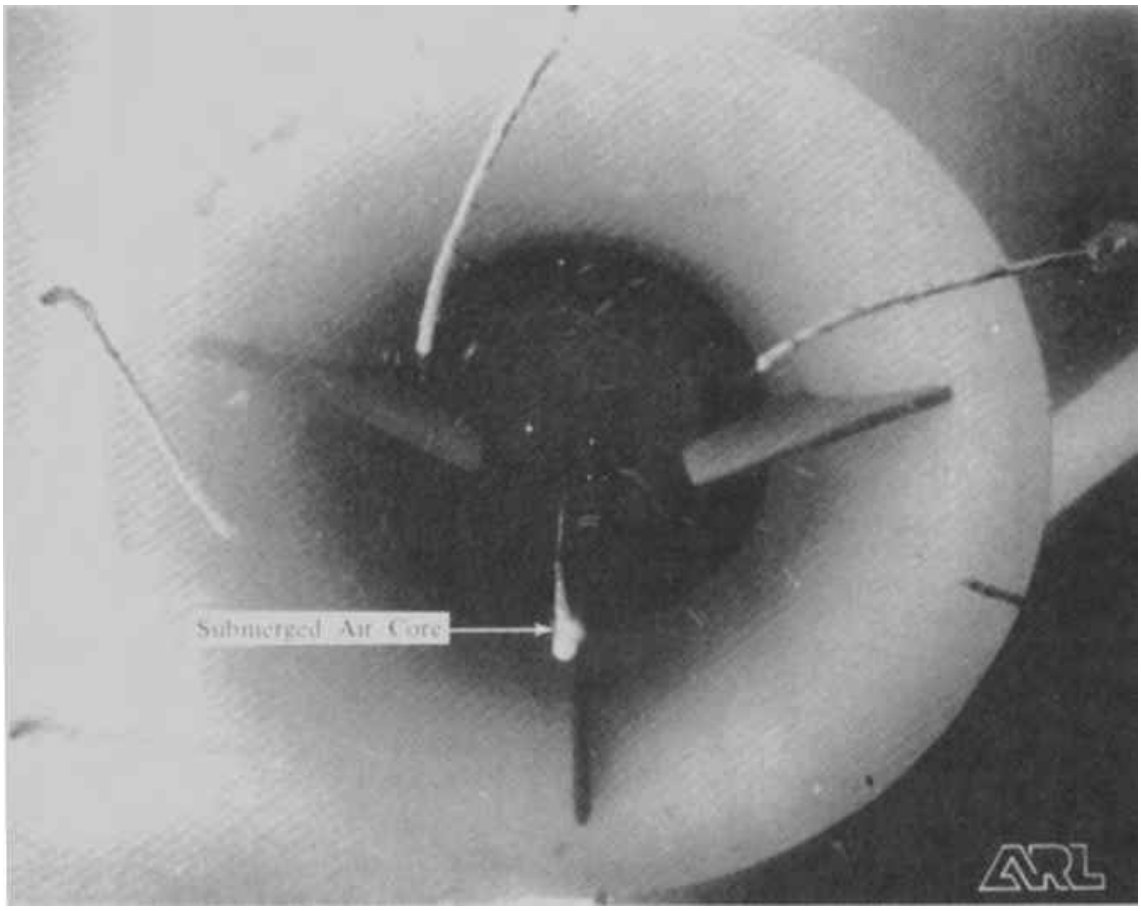


FIGURE 7 Yarn streamers around the suction bell of a model intake to observe flow direction into the bell (Courtesy of Alden Research Laboratory, Inc.)

Even though the pump bell velocity and submergence are very important parameters contributing to problems, it is seldom possible to change these values. Hence, the corrective measures are usually incorporated in the geometry of the pump pit and include the introduction of suitable appurtenant devices.

Severe free surface vortices may be prevented by providing a uniform approach flow to the pumps while maintaining a sufficient submergence. Required submergences can be predicted by empirical equations available in the literature.[31] Relocation of pumps, introduction of horizontal grids below the water surface, changing of wall and floor clearances, improvements in approach channel configurations, and changes in lengths and spacing of piers are some of the common techniques for reducing vortex activity. Surface vortices can also be controlled by installing a curtain wall immediately upstream of the pump. Figure 8 indicates various methods of eliminating poor flow conditions as given in Reference 7.

Submerged vortices are usually eliminated by installing splitter vanes or floor cones under the bell. Variations in floor and wall clearances can also be effective and should be tried in the model. Figure 9 shows a 1:4 scale model of a two-bay pump intake—each pump designed to draw 25,000 gpm (1.575 m^3/s). Flow distributor walls are installed to assure a nearly uniform flow to the pumps, and splitters and fillets are provided to guide the flow into the pump bells without any subsurface vortices.

The choice of corrective measures should always be made with the pump use in mind. For example, in testing pump intakes that will be used for sewerage, corrections that will allow trash build up, such as vanes or grids, should be avoided. It is also important that a continuous dialogue exist between the intake designer and the test laboratory so the most practical and inexpensive solution for a specific installation can be obtained with a minimum of effort.

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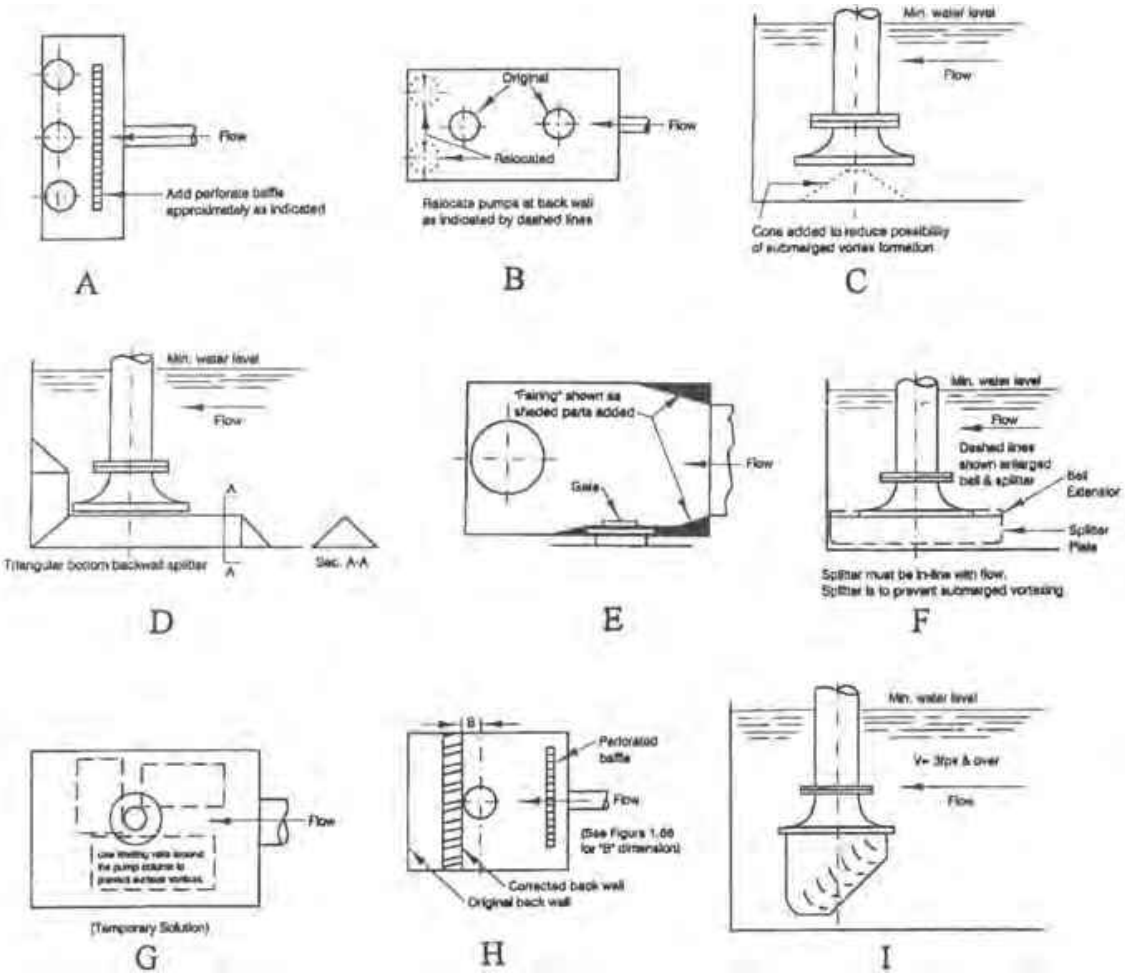


FIGURE 8A through I Common method of eliminating vortexing problems in pump pits (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 7)

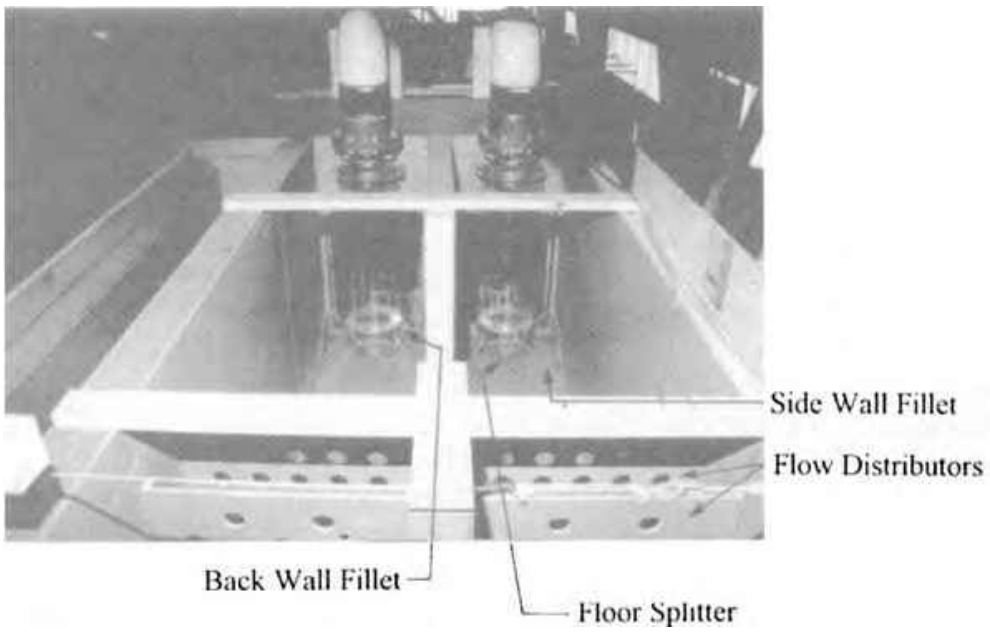


FIGURE 9 A 1:4 geometric model of a two-bay intake showing flow distributors, splitters, and fillets to eliminate objectionable vortices (Courtesy of Alden Research Laboratory, Inc)

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C • H • A • P • T • E • R • 14

SELECTING AND PURCHASING PUMPS

Trygve Dahl

Vinod Patel

INTRODUCTION

Enterprises engaged in the application, design, manufacture, and installation of pumping machinery are operating in an environment of increasing worldwide competition. The competitiveness of the firm's pumping machinery is substantially determined as a consequence of the system design, the selection and purchasing process for pumping equipment, as well as the ongoing operating and maintenance practices employed during the life cycle of the system. The design of the pump system, including the piping network and the components of the pumping system are performed first. Then, during the pump selection and purchasing process, purchasers expect manufacturers to design and offer high-quality equipment which meet their defined pumping requirements. Alternative pump designs that meet the technical requirements are compared based on total cost. This is either a simple evaluation based on total equipment cost or a detailed analysis taking into account the cost of energy, maintenance, and other total life cycle costs (LCCs). Finally, preferences are typically given to manufacturers that have a good reputation for quality and service and that deliver their product within an acceptable lead time.

The majority of the market for pumping equipment utilizes modular product families which are offered as premanufactured or assemble-to-order products, or customized into engineered-to-order (ETO) design variants. In such a design process, the "best" design is not necessarily globally optimum but rather, one that technically satisfies all of the functional requirements, is delivered in an acceptable period of time at the lowest evaluated cost, and meets other commercial constraints imposed by the purchaser. This "front-end" selection and purchasing process is crucial since the overall pump performance and long-term operating and maintenance costs of the equipment are determined by the purchasing decision.

This chapter is organized into two sections:

- *Steps in the Process*. This section describes the steps involved in specifying, selecting, and procuring a pumping system with particular emphasis on the pump and its associated appurtenances.
- *Emerging Trends in Selecting and Purchasing Pumps*. The information age is transforming the way the selection and purchasing process is performed. The emerging technologies and processes that will be more prevalent in the near future are outlined in this section.

STEPS IN THE PROCESS

The process in which a pump is selected and purchased varies considerably depending on factors such as the pumping application, the product architecture, the manufacturing quantity, and the degree of innovation required in the design process. These factors are illustrated in the following examples:

1. Consider a small pump used in a washing machine. These pumps are designed once to meet a single application and manufactured in quantities of hundreds of thousands of units a year. The pump is a fairly simple device, comprised of only a few parts, and requires little or no engineering design activity during the life span of the product. The major emphasis is on manufacturing and minimizing production costs.
2. Consider a line of bilge pumps used in recreational boats. A range of pumps are designed to meet a variety of pumping capacities and developed head (or pressure). Each model is designed based on preferred sizes established over years of sales. These pumps are designed for a well-understood application (i.e., bilge pumps for either fresh- or saltwater) and generally for a forecast demand. These pumps are not uniquely designed for each particular application but instead, chosen based on a catalog selection process.

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3. Consider a line of pumps conforming to API Standard 610/ISO 13709 "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries." The applications addressed by these specifications are broad covering a wide range of centrifugal pump classes. Pump manufacturers satisfy this wide range of pumping requirements by configuring or adapting pre-engineered part modules to build up a unique pumping system. This adaptation of preexisting product range designs to meet unique application requirements is known as an ETO design process.
4. Finally, consider a boiler feed pump used in a central station combined cycle power generation application. There are only a limited number of these plant constructions underway at any given time in the world. The low manufacturing quantity of these types of pumps and specialized conditions of service precludes pre-engineering a line of pumps to satisfy this market. As a result, each pumping system is built from substantially new parts designed specifically to meet the needs of the application. Although each pump is new, the design concepts are fairly well known and rely to a large extent on modifying design concepts that were successfully applied in previous applications. This type of application also requires an ETO design process but one that contains more innovation than the previous example.

The last two examples typify the selection and purchase of ETO pumping equipment where each pumping system is usually a unique design that transforms a specific set of pumping requirements into a system that fulfills those requirements. This is the type of selection and purchasing process described in this section.

Once the initial decision that pumping equipment is required, the selection and purchase of the equipment can be divided into the following general steps:

- Engineering the pumping system
  - Defining the pumping requirements
  - Designing the pumping system
  - Pump specification and data sheet preparation
- Pump selection and configuration
  - Pump selection
  - Pump configuration
  - Pump quotation
- Evaluating and purchasing the pumping equipment
  - Evaluation of bids and negotiation
  - Purchase of the selected pump and driver

In the process of specifying pumping equipment, the engineer must determine system requirements and system head curves, select pump type, write the pump specification, complete the pump data sheet, determine testing, inspection and vendor drawing and data submittal requirements and develop all the data necessary to define the required equipment from the supplier.

Having completed this phase of work, the engineer is then ready to take the steps necessary to purchase the equipment. These steps include issuing the pump inquiry to the bidders, technical and commercial evaluation of pump bids, selection of the supplier, and release of data necessary to issue the purchase order. The ultimate result of this process is the selection of a pump/driver combination that satisfies both the process and mechanical requirements.

ENGINEERING THE PUMPING SYSTEM

Defining the Pump System Requirements The governing functional requirements of the overall pumping system must be defined. Generally speaking, a pumping system requires transporting a fluid between two given locations with sufficient pressure to overcome static elevation changes, system friction losses, and to provide the desired dynamic

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pressure at the end location. These functional requirements are generally described by the service, fluid type, operating conditions, site conditions, and industry specification.

SERVICE Service is a general description of the particular pumping system. Examples might include booster pump, boiler feed pump, gasoline transfer pump, and so on.

OPERATING CONDITIONS Operating conditions define the general fluid capacity, pressure, and elevation change required by the pumping system. At this point, the number of pumps required for the service may not be known, so the specific operating conditions or each pump in the system is not fully defined.

SITE CONDITIONS These describe the location of the pumps, the altitude, and utility conditions for the driver (steam, electrical, and so on).

FLUID TYPE AND PROPERTIES A thorough description of the fluid to be handled must be developed. This includes properties such as viscosity, density, vapor pressure, corrosiveness, erosiveness, volatility, flammability, and toxicity. Depending on the process and the system, some or all of these properties may have an important effect on the pump and system design. For example:

- The temperature, pressure, and corrosiveness of the fluid will strongly influence the materials of construction.
- The fluid viscosity will influence the choice of pump technology, where low viscosity fluids are generally suitable for centrifugal pumps while higher viscosity fluids are better handled by positive displacement pumps.
- If the fluid contains solids in suspension, suitable types of pump seal designs and abrasion-resistant pump construction must be considered.
- Erosion due to high particle content may cause premature performance decline. Large particles may favor open impeller design.
- Fluid toxicity may necessitate the use of dual (tandem or double) mechanical seals due to government regulations or safety considerations.
- Entrained gases may affect the pump's ability to produce the required differential pressure.

The specified fluid physical and chemical properties must cover the entire expected operating range of the pumping system. Influences such as varying temperatures and pressures must also be defined.

INDUSTRY SPECIFICATION In many cases, the pumping application is governed by, or conveniently referenced to some type of industry specification. The design, construction, rating, and testing of most pumps used in refining and chemical industries are governed by standards such as API (American Petroleum Institute), ASME (American Society of Mechanical Engineers), the Hydraulic Institute, NFPA (National Fire Protection Association), PIP (Process Industry Practices), ISO (International Organization for Standardization), and various other international standards. The severity of the service in which the pump will be applied, as well as the location of the plant, will determine which industry standard (or standards) will be used, if any.

In the case of a fire pump service, NFPA compliance might be mandatory to meet the user's insurance company requirements. If a pump will be installed in an oil refinery or chemical plant, API and/or ASME standards will be applied depending on the severity of the service and client preferences. International standards such as DIN (German), BS (British), JIS (Japanese), or ISO can also be used. These standards are intended to provide a pump with a level of quality to match the needs and expectations of the end user of the equipment. It is obvious that the quality requirement for an emergency feedwater pump in a nuclear power plant needs to be much more stringent than a potable water booster pump in an office building. The quality issues covered by these codes/standards range from detailed design issues to inspection and performance testing requirements.

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Many end users, engineering consultants, and EPCs (engineering, procurement, and construction) firms have also developed their own specifications for different types of services that document best practices for those applications. These should always be referenced before commencing with the overall system design.

EVALUATION CRITERIA Before starting the pump system design, it is important to formalize the evaluation criteria. The engineer and the pump manufacturer shall be given guidance with respect to technical requirements, commercial requirements, lead time requirements, and the relative weighting of these factors. Further, as a "systems perspective" is gaining more recognition, the total LCC is becoming more recognized in the commercial evaluation, considering initial capital costs as well as ongoing operating and maintenance costs.

Designing The Pumping System

SYSTEM HEAD CURVES The engineer must have a clear understanding of the process and system in which the pumping equipment will operate. A preliminary design of the system should be made and should include an equipment layout and a piping and instrumentation diagram (P&ID). These preliminary drawings will show the various fluid flow paths for system operation, preliminary pipe diameters and lengths, relative elevations of system components and all valves and other piping components that will be used to establish the system head losses. These drawings will be used by the engineer to calculate the final piping sizes and pumping system head requirements.

With this information, the engineer can develop system head curves that show the relationship between flow rate and hydraulic losses in the piping system. In determining the hydraulic losses, the engineer must include adequate allowances for future corrosion and scale deposits in the piping system over the plant life.

Because hydraulic losses are a function of flow rate, pipe size, and layout, each individual flow path alignment in a given system will have its own characteristic operating curve. Care must be taken when specifying the required pump characteristics to take into account all possible system operating flow paths. It is convenient to add the effects of static pressure and elevation differences in the system to form a combined system head curve. This combined curve shows the total head required of the pumping equipment to overcome system resistance as well as differential static pressure and elevation. The pump head must be at or above the combined system curve at all required operating points and fluid conditions for the various system flow paths. Refer to Sections 11.1, 11.2, 12.1, and 12.2 for guidance in constructing system head curves.

MODES OF SYSTEM OPERATION System operating modes are important considerations when specifying pumping equipment. Will the pump be used in continuous or intermittent operation? Will the pump operate in parallel or series with other pumps? Will there be significant differences in head or flow rate requirements in different system alignments? Will a single pump be used as a common spare for two different pumping applications? These and other questions arising from analyzing the different modes of operation will help influence decisions as to the number of pumps needed, heads and capacities and whether booster pumps are desirable in some systems. It should be noted that unnecessarily conservative hydraulic requirements may increase pump complexity (such as the selection of a more elaborate multistage or double suction pump in place of simpler single stage, overhung pump) and increase the associated acquisition and operating cost.

The engineer should also consider plant maintenance service intervals in the design of the pumping system. This factor will influence the decision of quantity, pump type, requirement for installed spare(s), and the manufacturing quality required of the specified pumps. Frequently, due to the critical nature of a pumping service where high reliability is necessary, installed spares are provided. In some cases, 2 to 100% pumps are provided. When system flow rate requirements fluctuate, 3 to 50% pumps may be called for. When reduced flow rates will not adversely affect operations, 2 to 50% pumps can be specified. Plant operating philosophies will dictate if automatic start of a spare pump is required.

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PUMP FLOW/HEAD MARGINS Pumps are normally specified with a capacity margin above what has been determined necessary for the process. In addition, the calculated system head losses are also determined conservatively. The reasons for this include the following:

- During system design, many assumptions are made while determining pump requirements, some of which might eventually be determined to be incorrect.
- During the plant life cycle, process conditions are likely to change due to aging catalyst, changes in feed stock, or seasonal feed temperature variations.
- Final piping design may be significantly different from preliminary design.
- System hydraulic losses may change due to corrosion.
- Plans for future operating condition changes may be hampered if the piping system does not provide for adequate net positive suction head at the future flow conditions.

During preliminary system design, these potential future changes in head/capacity must be studied to determine the required design margin. On the other hand, care must be taken when applying margins to ensure that the purchased pump is not oversized. If the total head produced is too large, impellers may be trimmed within the allowable range of the pump model provided. Similarly, a pump should be selected to operate close to its best efficiency point, it is important to minimize the selected margin. Margins of 5 to 10% on flow are typical but even 20% is common, for example, in reflux tower service. In cases where the process is well proven and understood, and system operating requirements are well defined, a zero margin is sometimes appropriate. If the flow is significantly oversized, a costly energy penalty can result over the life of the plant. This is caused by lower pump operating efficiency at off-design flow rates and discharge throttling losses that may be necessary to control the flow rate to the desired value.

TYPE OF PUMP CONTROL The type of control for the required pump is also an important consideration for pump specification and selection. Since the actual piping system usually incorporates a design margin, a control valve is normally employed in centrifugal pump applications. This control valve (not typically supplied by the pump manufacturer) is used to adjust the system curve over the life of the unit. Flow sensing control provides the most stable operation for most systems. Pressure control can have very large swings in flow when operating in a flat or drooping area of a centrifugal pump curve. For this reason, most centrifugal pump specifications incorporate the requirement for the centrifugal pump operating curve to rise continually to pump discharge valve shut in (also known as the pump shut off). Variable speed pump operation is used more commonly as advances in variable frequency drives and control methods are more commercialized (see Figure 1). Also, the benefits in operating control, the possible elimination of control valves, and the lower energy costs obtained by an improved choice of operating speed is also contributing to the popularity of variable speed pumping (Hydraulic Institute and Europump, 2004).

Both temperature and level sensing control can lead to a pump running at shutoff or at the end-of-curve condition during upset or failure modes. Centrifugal pumps with a 10 to 20% head rise between the specified operating point and the shutoff point may be preferred for some services such as for parallel pump operation.

Pumps that may operate in the shutoff condition may need a continuously open bypass to prevent pump damage. Pump manufacturers will advise of the minimum required flow for an offered pump, but many end users require an additional margin above the manufacturers recommended minimum flow. If a continuous flow bypass is incorporated in the system design, this additional flow must be added to the specified pump requirement.

FUTURE SYSTEM CHANGES A final factor to be considered is the possibility of providing for future system changes. When future system changes can be predicted with a degree of certainty, the system can be designed with this in mind. Rather than selecting a pump that is operating at the high end of its preferred operating region, a larger impeller diameter or the next larger size pump operating at the beginning of its preferred operating range might be considered. In addition, the capability for installing a larger diameter

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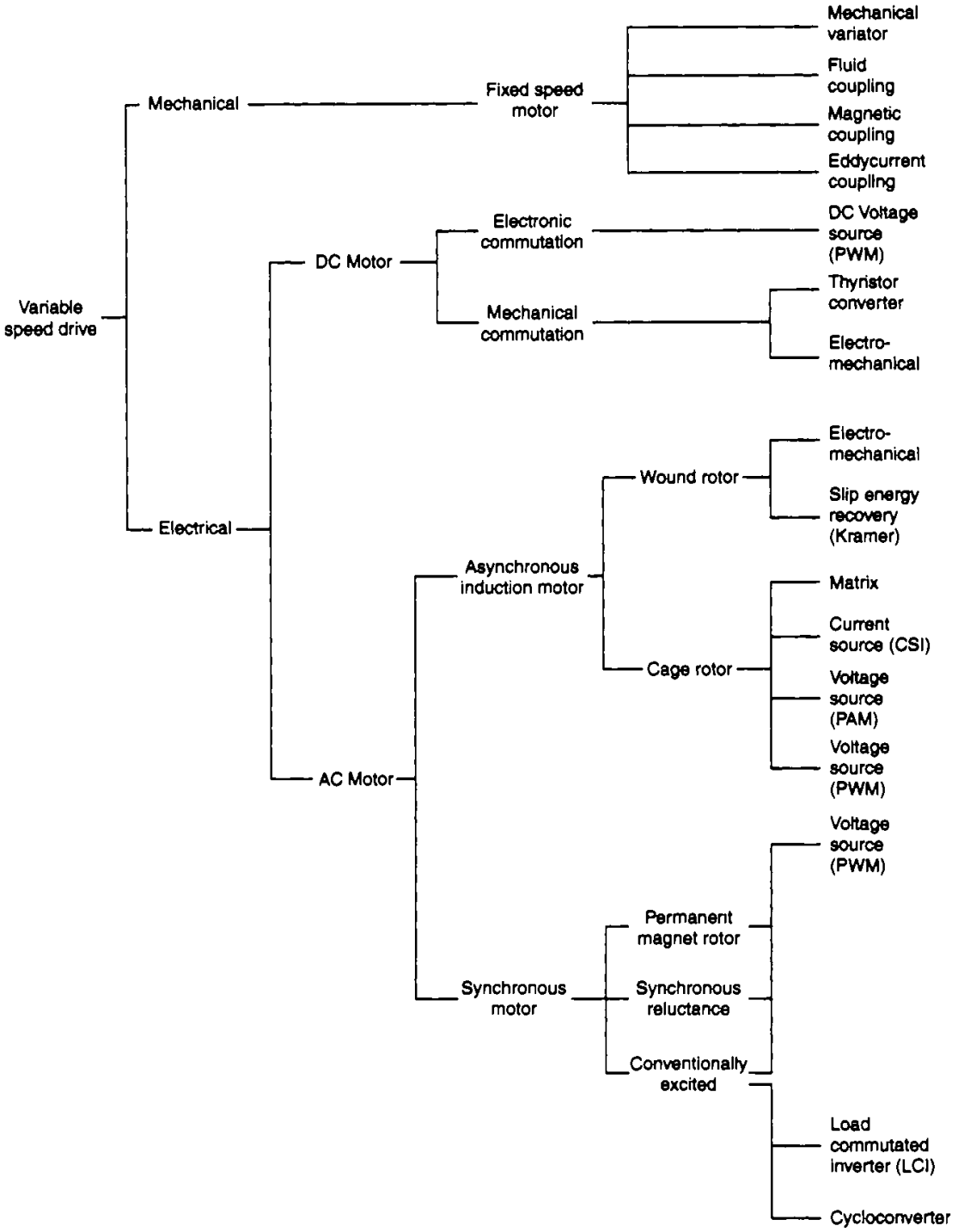


FIGURE 1 Types of variable speed drives (Courtesy Hydraulic Institute)

impeller to handle future higher head requirements must be considered. Because minimizing capital costs for a project is usually the prime consideration, oversizing a pump for future operations is not normal practice. Pumps are expected to operate efficiently and reliably in the present system, and this fact should be noted during the selection process.

Pump Specification and Data Sheet Preparation The pumping system requirements and design are typically documented in the pump specification and pump data sheet. This

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documentation is created by the purchaser and conveyed to the pump supplier, forming the basis for the request for quotation (RFQ).

REQUEST FOR QUOTATION In the RFQ, it is necessary to fully define the scope of supply for the desired piece of equipment. To do so requires the right amount of documentation provided by the purchaser to match the type of equipment being purchased. In some cases, the bid requisition can be as simple as a one-page listing of the pump requirements or it may be a document that incorporates data sheets, technical specifications, shipping specifications, purchasing terms and conditions, vendor drawing requirements, and any other document that will help define the full requirements of the intended purchase. A fully detailed requisition for a complicated pumping service might include over 100 pages of requirements.

As a minimum, the requisition must include a clear scope of supply, applicable specifications and data sheets, and commercial terms and conditions. In addition, the requisition can be used to specify additional requirements that have not been adequately addressed in other documents. These are described in more detail below.

PUMP TECHNICAL SPECIFICATION As discussed previously, there are many different designs of pumps that can be purchased. Engineering and operating companies may have standard technical specifications for all categories of equipment that are commonly purchased. When a pump service is being prepared for bids, a technical specification can be used in standard form, or updated to incorporate special requirements based on client or other project specific needs.

A technical specification is usually written to cover a wide range of equipment within a given category. Because many variations exist within any equipment category, the technical specification is usually written to be applicable to this range of equipment. Data sheets may be used to provide the specific requirements not covered in the technical specification.

Most technical specifications are written as performance specifications rather than design specifications. Care must be exercised to ensure that the pump manufacturer (not the purchaser) is responsible for the actual equipment design. If design requirements rather than performance requirements are specified, the vendor may claim that he is not responsible when performance requirements are not met. Care must be taken to assure that the equipment selected is from qualified vendors with proven design experience in similar applications. The purchaser must resist the temptation to force the vendor to modify a proven design and build a pump with which the vendor does not have proven experience.

As a minimum, a technical specification must list the following:

1. Industry codes and standards to which the pump must be designed, constructed, and tested. It is also possible to create a technical specification that stands alone and does not reference (or minimally references) codes and standards. This stand-alone technical specification will require a more thorough listing of needed requirements than one that references common design, fabrication, and testing standards.
2. A list of desired deviations and preferences from the main technical standards used to help define the pump requirements.
3. A definition of any technical terms used in the specifications that are not defined in referenced standards. This will help prevent misunderstandings between the purchaser and pump vendors.
4. A list of documents to be submitted by the successful bidder, both with the quotation and after the order is placed.

Pump Specification and Data Sheet

PUMP DATA SHEET The industry uses various layouts of equipment data sheets to collect, organize and exchange the technical information about the functional and physical requirements for pumping equipment. These data sheets range from company specific formats to industry standard layouts. The data sheet is used to list specific requirements for each individual pump service. These requirements are not general enough to be listed in

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the technical specification. As a minimum, the following items must be included on the data sheets for each service:

1. Pump name and item number
2. All required system design conditions and pump performance values needed to fully define the pump operating requirements, including any known turndown, start-up, or upset conditions
3. Materials of construction (unless the vendor is free to recommend standard materials)
4. All accessory requirements (unless the vendor is free to offer the standard selections) such as driver, coupling, seal, and packaging needs
5. Utility conditions (e.g., electrical voltage, frequency, and so on)
6. Intended site location and annual environmental conditions in this location
7. Electrical hazardous area classification
8. Instrumentation and control requirements (usually out of the vendors scope)
9. Noise requirements
10. Any specific preferences that deviate from requirements of the technical specifications
11. Inspection and testing requirements

In addition, the vendor should be advised to fill in the applicable blanks in the data sheet and submit the proposal. The manufacturer's data will help define the offering. See Figure 2 for excerpts from an API-610, 10th edition / ISO 13709 data sheet.

OTHER REQUISITION DOCUMENTS AND REQUIREMENTS A purchaser may include many additional options in the bid requisition. These requirements may be included in the requisition narrative or data sheets. Some possible options are addressed in the following paragraphs.

1. *Alternates.* It is extremely difficult for a specification to cover all possible pumps offered by various manufacturers. In addition, the pump industry constantly updates their products due to competition and revisions to industry standards. Because of this, it is good practice to allow manufacturers to offer alternatives to the specified pumping equipment. This allows manufacturers to present their best offerings and gives the purchaser the advantage of obtaining commercially and technically attractive alternate offerings. However, the choice of whether to accept an alternative is retained by the purchaser.
2. *Vendor data requirement (VDR) form.* For complicated equipment where many different documents are requested for submittal by the purchaser, a list of these documents is usually included in the requisition. This list should include a listing of each generic document, the number of copies required and the time after placement of purchase order that the document is required by the purchaser. Requirements for drawings are significantly different for an end user maintenance group and an engineering contractor. Because documentation can add a significant amount to the purchase cost, care should be taken when specifying required drawings and drawing quantities.
3. *Inspection and testing checklist.* For critical and special pump applications, the purchaser may require significant shop inspection and testing of the equipment above the vendor's standard procedures. This might include materials certification, NDE, welding inspections, rotor balancing, inspection reports, and mechanical/performance tests.
4. *Noise data sheet.* If the equipment is large enough or of a type known to be a noise concern, the purchaser may specify the maximum allowable noise level attributable to the equipment being purchased. One way to accomplish this is to include a noise data sheet that lists the allowable levels for the equipment provided. This data sheet should be designed to require the vendor to fill in the expected noise level for the equipment being offered. If the expected noise is greater than the specified maximum value, the vendor should be directed to provide special design options (i.e., noise enclosure, WP II motor special noise ducting, and so on) to lower the expected noise level.

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CENTRIFUGAL PUMP
API 610/ ISO 13709 TENTH EDITION
PROCESS DATA SHEET
US CUSTOMARY UNITS

JOB NO. _____ ITEM NO. (S) _____
REQ/SPEC NO. _____ /
PURCH ORDER NO. _____ DATE _____
INQUIRY NO. _____ BY _____

APPLICABLE TO: ○ PROPOSALS ○ PURCHASE ■ AS BUILT
FOR _____ UNIT _____
SITE _____ SERVICE _____

NOTES: INFORMATION BELOW TO BE COMPLETED: ○ BY PURCHASER ☐ BY MANUFACTURER ■ BY MANUFACTURER OR PURCHASER

○ DATA SHEETS

	ITEM NO.	ATTACHED	ITEM NO.	ATTACHED	ITEM NO.	ATTACHED
PUMP		○		○		○
MOTOR		○		○		○
GEAR		○		○		○
TURBINE		○		○		○

REVISIONS

NO.	DATE	BY
1		
2		
3		
4		
5		

APPLICABLE OVERLAY STANDARD(S):

○ OPERATING CONDITIONS (5.1.3)

FLOW, NORMAL _____ (gal/min) RATED _____ (gal/min)
OTHER _____
SUCTION PRESSURE MAX/RATED _____ / _____ (psig)
DISCHARGE PRESSURE _____ (psig)
DIFFERENTIAL PRESSURE _____ (psi)
DIFF. HEAD _____ (ft) NPSHA _____ (ft)
PROCESS VARIATIONS (5.1.4) _____
STARTING CONDITIONS (5.1.4) _____
SERVICE: ○ CONT. ○ INTERMITTENT (STARTS/DAY) _____
○ PARALLEL OPERATION REQD (5.1.13)

○ SITE DATA (5.1.3)

LOCATION: (5.1.30)
○ INDOOR ○ HEATED ○ OUTDOOR ○ UNHEATED
○ ELECTRICAL AREA CLASSIFICATION (5.1.24/6.1.4)
CL _____ GR _____ DIV _____
○ WINTERIZATION REQD. ○ TROPICALIZATION REQD.
SITE DATA (5.1.30)
○ ALTITUDE _____ (ft) BAROMETER _____ (Psia)
○ RANGE OF AMBIENT TEMPS: MIN/MAX. ____ / ____ (°F)
○ RELATIVE HUMIDITY: MIN/MAX _____ / _____ (%)
UNUSUAL CONDITIONS: (5.1.30) ○ DUST ○ FUMES
○ OTHER _____

○ DRIVER TYPE

○ INDUCTION MOTOR ○ STEAM TURBINE ○ GEAR
○ OTHER _____

○ MOTOR DRIVER (6.1.1/6.1.4)

■ MANUFACTURER _____
☐ _____ (HP) ☐ _____ (r/min)
☐ FRAME _____ ■ ENCLOSURE _____
■ HORIZONTAL ■ VERTICAL ■ SERVICE FACTOR ____
■ VOLTS PHASE/HERTZ _____ / _____ / _____
○ TYPE _____
○ MINIMUM STARTING VOLTAGE (6.1.5) _____
■ INSULATION _____ ○ TEMP. RISE _____
■ FULL LOAD AMPS _____
■ LOCKED ROTOR AMPS _____
■ STARTING METHOD _____
■ LUBE _____

BEARINGS (TYPE/NUMBER):
☐ RADIAL _____ /
☐ THRUST _____ /
☐ VERTICAL THRUST CAPACITY
UP _____ (lb) DOWN _____ (lb)

○ LIQUID (5.1.3)

LIQUID TYPE OR NAME _____
○ HAZARDOUS ○ FLAMMABLE ○ _____ (5.1.5)

	MIN.	NORMAL	MAX.
PUMPING TEMP (°F)			
VAPOR PRESS (psia)			
RELATIVE DENSITY (SG):			
VISCOSITY (cP)			

SPECIFIC HEAT. Cp _____ (Btu/lb °F)
○ CHLORIDE CONCENTRATION (6.5.2.4) _____ (ppm)
○ H_2S CONCENTRATION _____ (ppm) WET (5.12.1.12c)
CORROSIVE/ EROSIVE AGENT _____ (5.12.1.9)

MATERIALS (5.12.1.1)

○ ANNEX H CLASS (5.12.1.1) _____
○ MIN DESIGN METAL TEMP (5.12.4.1) _____ (°F)
○ REDUCED HARDNESS MATERIALS REQD (5.12.1.12)
☐ BARREL/CASE _____ IMPELLER _____
☐ CASE/IMPELLER WEAR RINGS _____
☐ SHAFT _____
☐ DIFFUSERS _____

■ PERFORMANCE:

PROPOSAL CURVE NO. _____ ☐ R/MIN _____
☐ IMPELLER DIA. RATED _____ MAX ____ MIN ____ (in.)
☐ IMPELLER TYPE _____
☐ RATED POWER _____ (5hp) EFFICIENCY _____ (%)
☐ MINIMUM CONTINUOUS FLOW:
THERMAL _____ (gal/min) STABLE _____ (gal/min)
☐ PREFERRED OPER. REGION _____ TO _____ (gal/min)
☐ ALLOWABLE OPER. REGION _____ TO _____ (gal/min)
☐ MAX HEAD @ RATED IMPELLER _____ (ft)
☐ MAX POWER @ RATED IMPELLER _____ (BHP)
☐ NPSHR AT RATED FLOW _____ (ft) (5.1.10)
○ MAX. SUCTION SPECIFIC SPEED: _____ (5.1.11)
○ MAX. SOUND PRESS. LEVEL REQD. _____ (dba) (5.1.16)
■ EST MAX SOUND PRESS. LEVEL _____ (dba) (5.1.16)
■ EST MAX SOUND POWER. LEVEL _____ (dba) (5.1.16)

○ UTILITY CONDITIONS (5.1.3)

ELECTRICITY	VOLTAGE	PHASE	HERTZ
DRIVERS			
HEATING			

SYSTEM VOLTAGE DIP ○ 80% ○ OTHER _____ (6.1. 5)

STEAM	MAX. PRESS.	MAX. TEMP.	MIN. PRESS.	MIN. TEMP.
DRIVERS				
HEATING				

COOLING WATER: (5.1.19) SOURCE _____
SUPPLY TEMP. _____ (°F) MAX. RETURN TEMP. _____ (°F)
NORM. PRESS. _____ (psig) DESIGN PRESS _____ (psig)
MIN. RET. PRESS. _____ (psig) MAX. ALLOW. D.P. _____ (psi)
CHLORIDE CONCENTRATION: _____ (ppm)

FIGURE 2 API 610 / ISO 13709 Data sheet—page 1 (Courtesy American Petroleum Institute)

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COST EVALUATION The "total cost" of the pumping equipment, in the most simplistic sense, is the purchase price of the equipment as quoted by the pump supplier. Many purchasers base their economic purchasing decision on this purchase price alone. More commonly purchasers are including an energy evaluation or even an LCC evaluation in their cost evaluation.

ENERGY EVALUATION The purchaser may choose to include operating costs in the equipment evaluation. If so, the vendor needs to be informed of this requirement so the most attractive proposal can be offered. In this case, the overall equipment cost will be a combination of first cost and the differential cost of energy used over a specified period of time. If so, the vendor must be informed of the present worth payback time for this analysis (such as 3 years), cost of a unit of energy, and the number of hours per year that the equipment will operate. Based on the differential power consumed over the payback period and the energy cost of this differential, the penalty or credit can be easily calculated.

LCC EVALUATION A more sophisticated LCC evaluation method is gaining acceptance in the industry. Traditionally, cost evaluations have been limited to the cost of the initial capital equipment plus some evaluation for the cost of energy. An LCC cost equation, developed by the Hydraulic Institute, is represented by the following equation:

$$LCC = C_{ic} + C_{in} + C_e + C_o + C_m + C_s + C_{env} + C_d$$

where

- C_{ic} = initial costs, including the purchase price of the pump set (pump(s), motor, motor starter, and so on)
- C_{in} = installation and commissioning cost (includes installation costs such as foundations, electrical installation, piping, isolation valves, and so on)
- C_e = energy cost (usually electricity because most pumps are driven by electric motors)
- C_o = operating costs (includes labor cost of system supervision and operating costs such as cooling water, steam heating, centralized oil lubrication systems, and so on)
- C_m = maintenance (includes routine servicing and unplanned repairs)
- C_s = down time costs (potential loss of income due to unplanned downtime, sometimes neglected if a spare pump is included in purchase price)
- C_{env} = environmental cost (contamination from pumped liquid and auxiliary equipment)
- C_d = disposal/decommissioning costs (to dispose of the equipment, in an environmentally responsibly way)

The LCC is usually discounted to a present value, based on an assumed discount rate, inflation rate, and expected equipment life. See the section titled "Life Cycle Cost Calculations" and Figure 18 later in this chapter for more information on this topic.

COMMERCIAL TERMS AND CONDITIONS The requisition should include the following commercial terms and conditions:

- Name of buyer, place to which proposals must be delivered, information on ownership of documents, time allotted for submission of bids, governing laws and regulations.
- Location of plant site.
- Site storage conditions and anticipated length of storage (preparation requirements).
- Schedule for submittal of drawings/documentation, and pump delivery.
- Guaranty/warranty requirements.
- Instruction on minimum information to include in the proposal, number of copies that vendor must provide, status of alternative offerings, and a statement on the owners right to accept or reject bids that are not in accordance with the bid package.

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- Acceptable terms of payment.
- Method of transportation to site that establishes the vendor's responsibility. If the vendor will not be responsible for providing shipping, he must be requested to provide enough information in the quote for others to estimate shipping costs.
- Customer shop inspection requirements.
- Any penalty or bonus related to late or early shipping.

The list of specifications and data sheets must include document name or description, document number, revision number, and revision date. This will provide a record of what documents have been sent to a vendor.

Formalizing the RFQ Submittal

BIDDERS LIST PREPARATION The preparation of a bidders list for the required equipment on a project is of utmost importance. This list is comprised of suppliers that the client and purchaser agree are qualified to supply the needed equipment. Qualification requirements may include a variety of categories such as the following:

- Vendor proven successful experiences supplying similar equipment
- Vendor ability to meet required delivery schedule
- Vendor ability to support field installation
- ISO 9001 certification standing
- Location of manufacture or material supply
- After sales service, spare parts supply

If a proposed vendor is not known to the purchaser, a shop survey may be required to satisfy the purchaser of a potential supplier's ability. This type of survey will usually include an inspection and assessment of the manufacturer's shops, engineering and design facility, and quality assurance organization. The complexity of the required equipment and the project needs will determine the level of qualification required of the vendors. Remember that a vendor is required to provide not only the specified equipment, but may also need to provide civil, piping, and electrical interface information required for project design. A vendor must possess the skills necessary to support all of these requirements.

The number of vendors that are to receive bid requests is usually between three and five. This number will provide for effective competition and will limit the quantity of quotes that require evaluation.

Pump alliance agreements, or the need to duplicate existing equipment, may reduce the list to a single bidder. In addition, unique design or vendor experience may dictate a sole source supplier.

EVALUATION CRITERIA Upon preparing the formal RFQ submittal to the selected pump suppliers, the purchaser should clearly define the criteria in which the suppliers' quotations will be evaluated. This helps the suppliers optimize their pump technical and commercial offer to meet the purchasers' requirements.

In summary, the evaluation criteria for the selection and purchase of pumping equipment should include:

- *Satisfaction of technical requirements*. This is found in the pump specification, data sheets, and other requisition documents and requirements.
- *Minimum total cost*. This is described by the purchaser as either a purchase price evaluation, energy evaluation, or the more sophisticated LCC evaluation.
- *Commercial terms and conditions*

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PUMP SELECTION AND CONFIGURATION

Once the RFQ (or *inquiry*, as it is often called) has been prepared by the pump purchaser, a pump quotation (or *bid)* must be obtained from one or more qualified pump manufacturers. The quotation will contain both technical and commercial information, sufficiently detailed to allow the pump purchaser to make a purchasing decision.

The primary tool that pump manufacturers have traditionally used to prepare a quotation is commonly known as the "price book." One might envision a price book as a simple price list which sales engineers use to quote pumps. In the most routine case, this is true. More often, however, the price book is a technical manual which contains extensive technical information such as pump performance curves, materials of construction tables, engineering data including dimensions and cross-sectional drawings, a plethora of guidelines on proper application of pumping equipment, and yes, pricing information. Essentially, the price book is an engineering design, specification, and pricing manual traditionally used by pump manufacturers to support the preparation a pump quotation.

Generating a pump quotation is described by the following three-step process:

1. *Pump selection*. Generating the "best" pump selection for the given set of customer requirements and specifications
2. *Pump configuration*. Generating the pump scope of supply, including supporting documents such as data sheets, price calculations, and drawings
3. *Pump quotation*. Preparing a complete quote package, including sufficient technical and commercial data to support a purchasing decision

Pump Selection Pump selection is a process that follows the following major steps:

1. Establish the conditions of service (COS)
2. Identify the most suitable product lines that might satisfy the COS
3. Search all candidate pump sizes within the selected product lines toward satisfying the COS
4. Size and calculate the complete performance characteristics for all candidate pumps

Skilled pump applications engineers typically perform the pump selection process. This process has been historically accomplished with aid of a technical manual and hand calculations. More recently, computer-aided selection programs are used to automate the procedure and assure a more accurate and optimized result. Examples are shown in the sections below.

Pump Conditions of Service The pump conditions of service are initially acquired directly from the purchasers pump data sheet and specifications. Unless the specification dictates the type of pump, the COS are described without prejudice to the type of pump for the service. This allows the manufacturer to consider a broader range of pump designs, each having their own inherent strengths and weaknesses. A typical COS list is shown in Figure 3 and outlined in the following paragraphs.

BASIC OPERATING CONDITIONS Hydraulic performance for pumps are established by the rated pump capacity (flow rate), the rated differential head (or pressure) required, and driver speed selection strategy to bound the allowable range of operating speeds of the pump. These parameters are sufficient to perform a basic hydraulic selection defining the pump size and type, impeller design and impeller trim, casing or diffuser design, the number of pump stages and the operation speed.

ADVANCED OPERATING CONDITIONS Advanced operating conditions extend beyond the basic hydraulic selection by introducing parameters that affect the pump design. These

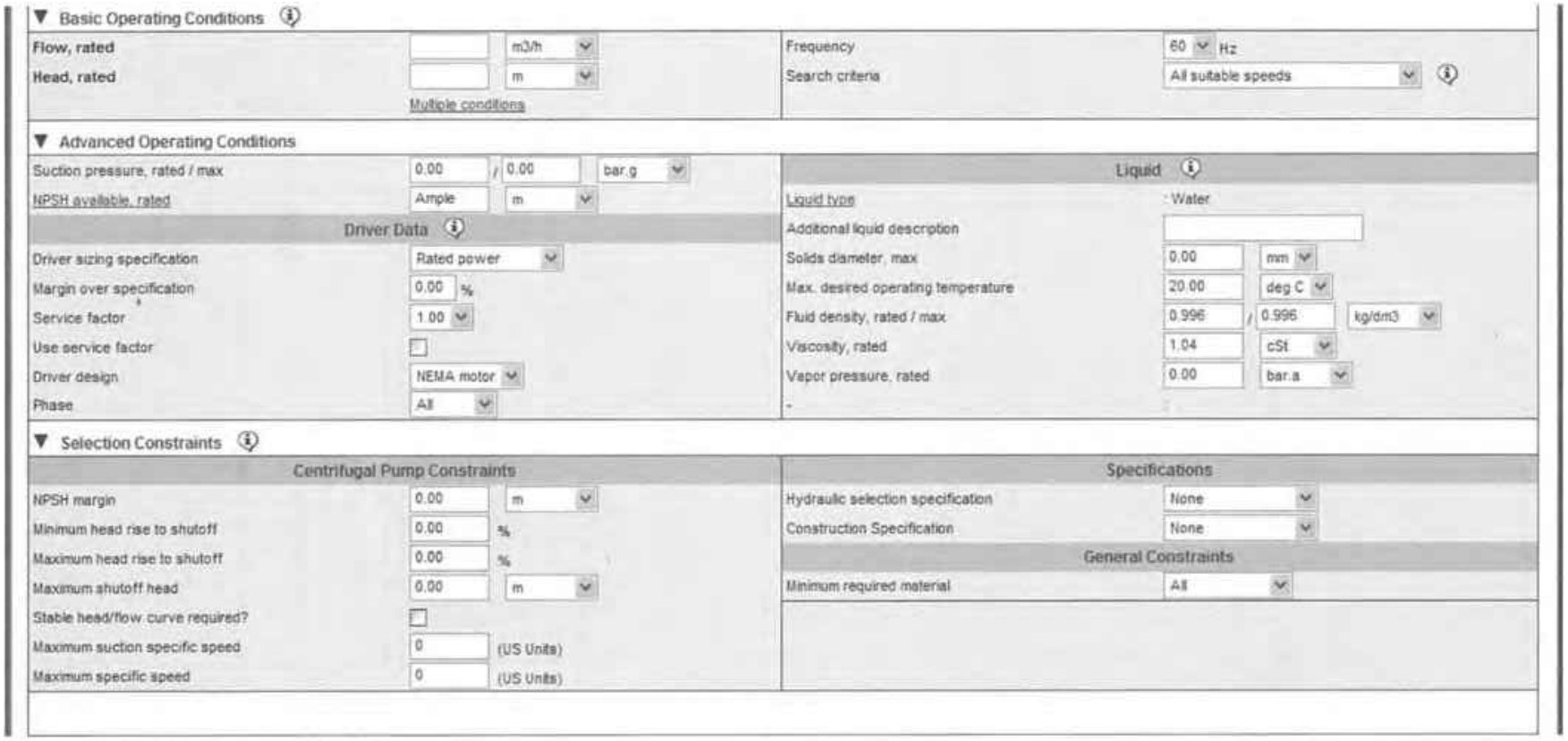


FIGURE 3 Pump conditions of service (Courtesy Intelliquip, LLC)

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parameters include suction pressure, fluid characteristics, net positive suction head available (NPSHA) and driver design. These parameters strongly influence the suitability of various pump types.

FLUID CHARACTERISTICS Fluid characteristics such as viscosity, density, vapor pressure, volatility, chemical stability, solid content, and entrained gases are important factors to be considered for proper pump selection. Pumps are available to handle a full range of fluid types, but each pump type has its limitations. A positive displacement progressing cavity pump can be used to pump toothpaste, peanut, butter, and shampoo, but it will not usually be a good choice for pumping water or gasoline. A rotary gear pump or variable displacement piston pump is a good choice for a hydraulic control system, but not for a potable water application. A rotary sliding vane pump can be successfully applied for pumping hot asphalt and for limited application in a lube oil system.

Making the best pump selection for a certain fluid application is often difficult. Previous successful experience is usually the best guideline for proper pump selection. This information can be obtained from end users, from process licensers, and from pump manufacturers. Recommendations from all of these sources should be carefully considered.

PUMP MATERIALS Material selection is affected both by the pumped fluid and the environment. Resistance to corrosion and erosion are of prime importance. The engineer must determine which material is most suitable and economical for a particular service. This requires that an evaluation comparing the more expensive longer life material to a less expensive material, which may provide a shorter pump life. Requirements such as continuous or intermittent operation, critical or noncritical service and plant life cycle should be considered when selecting materials.

Pumps are commonly available in cast iron, ductile iron, bronze, carbon steel, alloy steels, and in some cases composite materials or special alloys such as Monel, Hastelloy, or titanium. In addition to the importance of pump design life, safety must also be considered when selecting materials. Cast iron construction is not used for pressure casing parts of pumps that are to handle flammable or hazardous liquids because cast iron is more subject to fracture when thermally shocked. For these services, pressure-casing parts must be high strength ductile materials such as carbon or alloy steel.

DRIVER SELECTION The choice of driver type for a pumping service is as important as the pump selection. Factors that affect the driver choice are capital cost, driver type availability, operating reliability and the availability and cost of utilities.

Constant speed electric motors are most economical when only the first cost is considered. Often there is excess steam available within a facility that, when compared to the cost of electricity, will justify the extra cost of a steam turbine. Reliability requirements may necessitate the use of both a steam driven main pump and an electric motor-driven backup pump. In the case of firewater pumps, a battery-start, diesel-fueled internal combustion engine is needed to be completely independent of plant utilities. More expensive variable speed electric motors can sometimes be justified if the pump is operated well below its design conditions and there is the potential for significant savings in power.

There are other factors that should be considered when selecting the pump driver. The capital cost as well as the installation cost is more expensive for a steam turbine due to required piping. Steam turbines also require more maintenance during plant life, which may be undesirable to the owner. Selection based on past proven performance and selection to match existing plant equipment to minimize spare part inventory is a common consideration.

Air-operated diaphragm-type pumps are available in relatively small capacities, and these can be particularly effective in hazardous area classifications where use of electric motors may be undesirable.

SPECIFICATION REQUIREMENTS Specification requirements that affect the pump selection need to be recognized early in the selection process. For example, conformance to an API specification for a process pump application might invoke API 610 requirements for centrifugal process pumps or API 676 for rotary pumps. These specifications have implied

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requirements that will limit the range of manufacturers and product lines suitable for the service. Other examples include suction specific speed limits and minimum head rise to shutoff limits, as shown in Figure 3.

Product Line Selection Now that the conditions of service are defined, one must select those product lines that satisfy the conditions of service. The suitability of a pump product lines is based on many factors, including basic product design, reliability, previous installations under similar service, and of course, cost. A brief outline of different pump types and their relative design features and characteristics are described below.

PUMP TYPES There are several different types of basic pump designs. Each design can be used for a range of flow and head combinations. Refer to Chap. 1 for a general overview of the type of pump that can be used for various heads and flow rates.

In addition to capacity and head considerations, other operating characteristics of a pump will help the engineer to select a pump type for a given application. These are typically based on maximum temperature, viscosity, and liquid handling capability, available materials of construction, solids handing, and general pump design. These various types of pumps are shown in Figure 4, each with their respective design advantages.

The two main branches in pump types are kinetic pumps and positive displacement pumps. Most applications engineers consider pumps from the either the centrifugal or positive displacement pump branches. Centrifugal pumps, like all dynamic pumps, add energy continuously to the fluid to increase the fluid velocities within the pump such that a velocity decrease near the pump discharge results in an increase in fluid pressure. Alternatively, positive displacement pumps periodically add energy to the liquid by direct application of force to moveable enclosed fluid volumes, resulting in an increase in pressure up to the point where the fluid passes through a port or valve into the discharge line. An important feature of a centrifugal pump is that for a given operating speed, the discharge flow will vary from the rated flow condition to zero flow, depending on the discharge pressure of the pump. Conversely, a positive displacement pump delivers nearly the same discharge flow rate at a given operating speed, irrespective of the discharge pressure of the pump.

In general, centrifugal pumps are favored by specifying engineers when both types of pumps are suitable for the conditions of service. This is due to the belief that centrifugal pumps are more reliable than a corresponding positive displacement pump since they generally have fewer moving parts, no check valves, and lower pressure pulsations than the positive displacement pump. There are certain applications that are certainly more suitable for positive displacement pumps, such as applications with:

- Low flow and high pressure
- High viscosity
- Low shear
- Self-priming
- Accurate flow rate (flow metering)
- Multiphase flow
- Solids handling
- High efficiency

A more detailed description, including design and application details of kinetic and positive displacement pumps are described in detail in Chap. 3. Some useful considerations in pump-type selection are detailed in the following sections.

VARIABLE HEAD/FLOW REQUIREMENT Centrifugal and axial flow pumps are able to operate in variable head/flow conditions. By reviewing the performance curve for a given pump, the head/flow range capability for these types of pumps can easily be determined. Refer to the centrifugal pump performance curve in Figure 5. For a specific impeller size, a centrifugal pump will produce any flow rate within its head-flow rate characteristic curve

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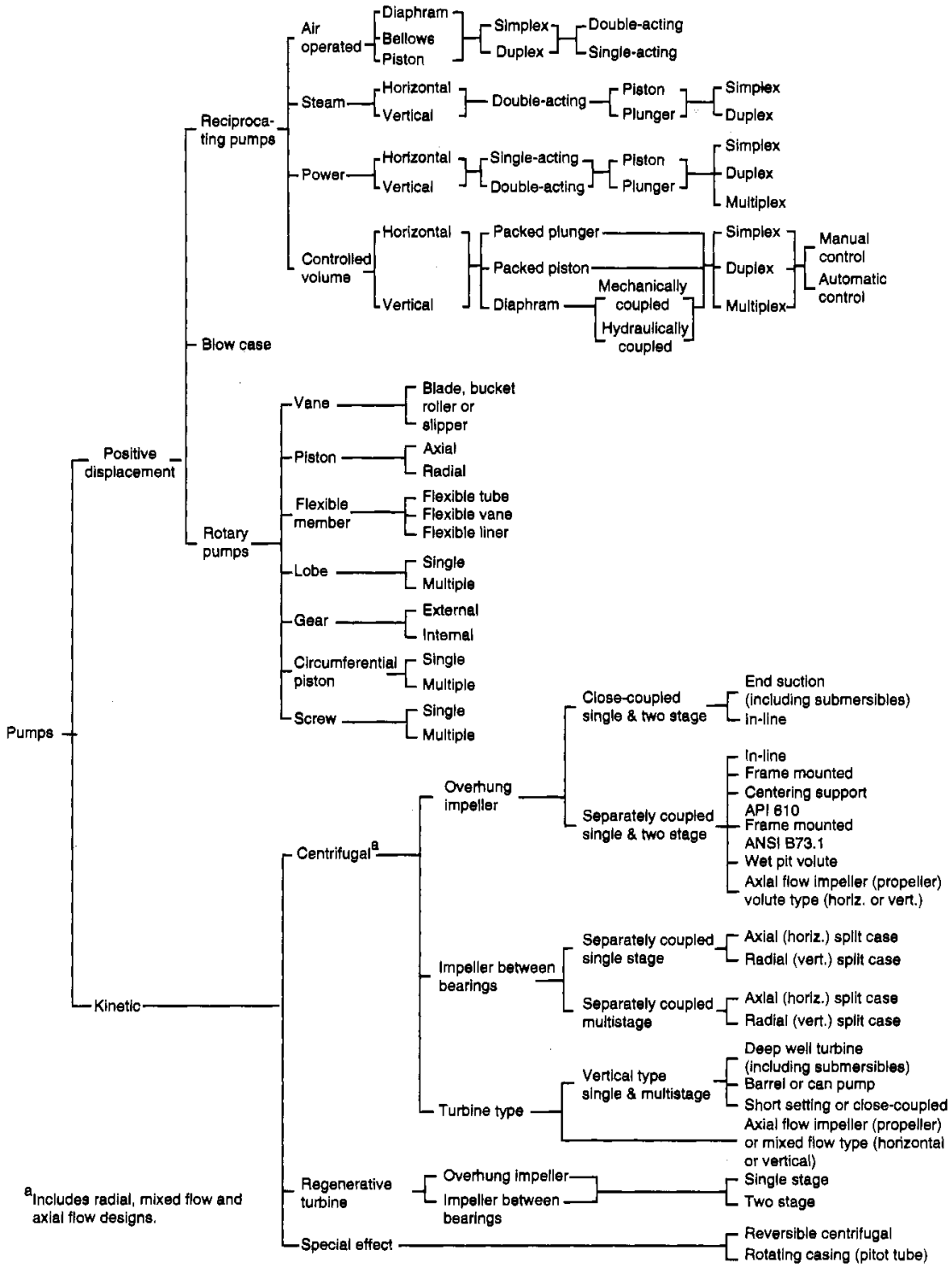


FIGURE 4 Types of pumps (Courtesy Hydraulic Institute)

which corresponds to intersection with the system head curve (see Figure 5), if sufficient NPSH is available. The system head characteristics can be changed to vary the flow by discharge throttling or by varying pump speed.

CONSTANT FLOW REQUIREMENT Positive displacement pumps deliver a nearly constant flow rate as a function of the operating speed. A slight decrease in flow rate accompanies

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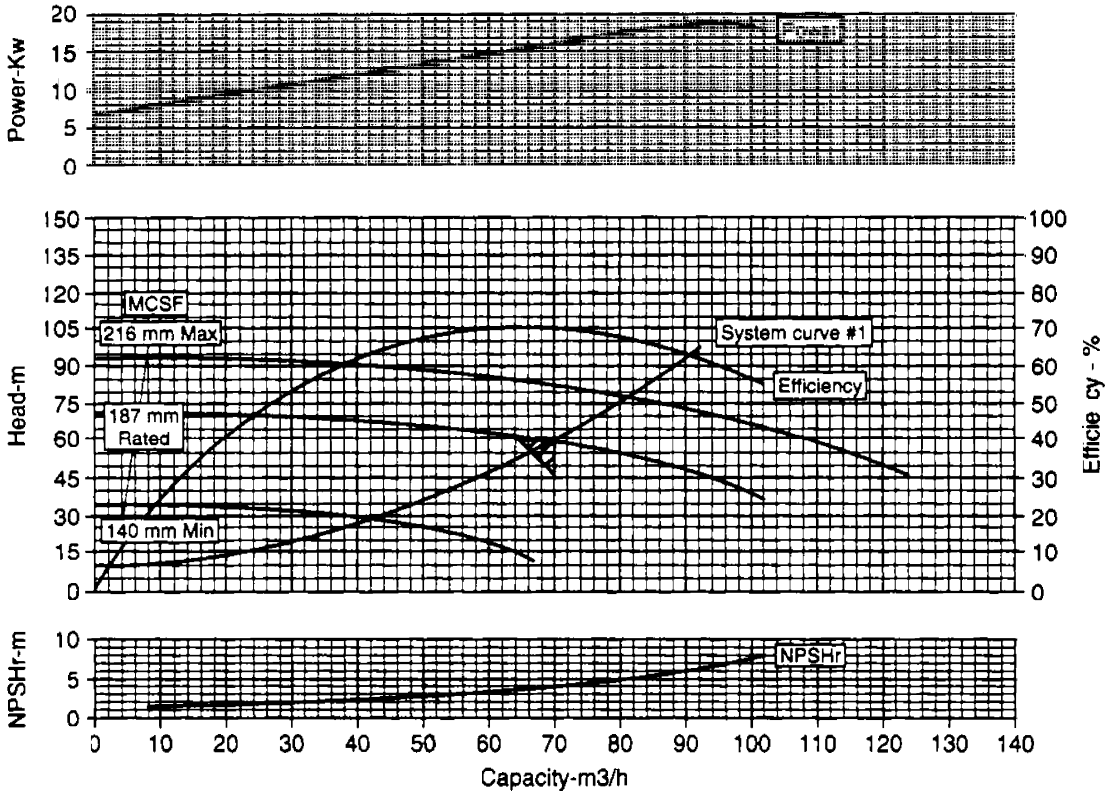


FIGURE 5 Centrifugal pump performance characteristic (Courtesy Intelliquip, LLC)

a larger differential pressure, due to internal leakage or "slip" within the pumping cavities (see Figure 6). Since a positive displacement pump will attempt to deliver a constant flow, irrespective of the discharge pressure level, a relief valve is usually installed to prevent over pressurizing the discharge piping system.

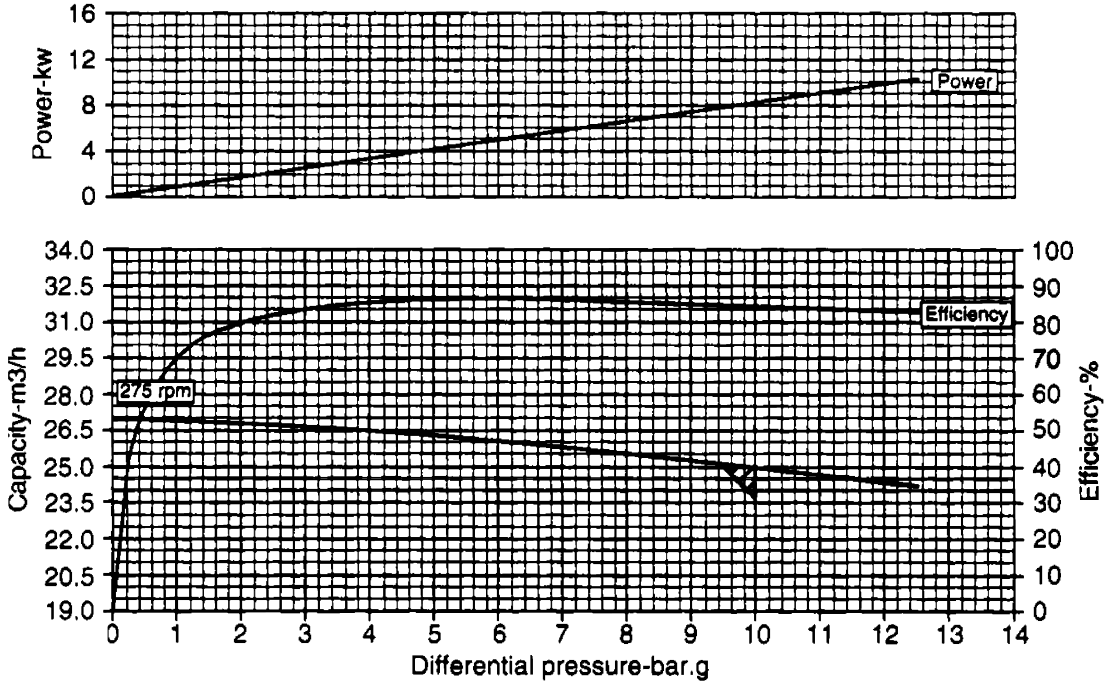


FIGURE 6 Positive displacement pump performance characteristic (Courtesy Intelliquip, LLC)

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Notice that the conventional format for centrifugal pump performance in Figure 5 and the positive displacement pump performance in Figure 6 inverts the abscissa and ordinate curve axis. If a positive displacement pump flow versus pressure characteristic is plotted on a centrifugal pump curve axis (as in Figure 5), the line curve would appear nearly vertical.

HIGH HEAD REQUIRED (ABOVE SINGLE STAGE CENTRIFUGAL PUMP ABILITY) Depending on the required flow rate, either a centrifugal, regenerative turbine pump, or a piston pump may fulfill the need for high differential head. If a relatively small flow is required, either an integrally geared high-speed centrifugal pump, regenerative turbine pump, or a piston pump may be applied. When selecting between these choices, other questions to ask might be:

- Will the pulsating flow of a piston pump be detrimental to system operation? Will a pulsation dampener take care of this problem?
- Is the liquid clean enough to avoid premature wear on pistons and cylinders?

For medium to high flow and high head combinations, a multistage centrifugal pump can be used. These are commonly available in horizontal, between bearing design in either a split case or radially split configuration. Alternatively, vertical multistage pumps, often known as turbine-type pumps are also used. Various designs of multistage centrifugal pumps are available with a wide range of prices reflecting special designs for a whole range of applications (high temperature, cryogenic, water, hydrocarbon, and so on).

LOW FLOW WITH PRECISE FLOW ADJUSTMENT ABILITY For low-flow applications where accurate flow metering is necessary, a proportioning or metering pump is appropriate. This type of pump can also be provided with variable flow capability. Certain types of gear, plunger, and diaphragm pumps can also be used in combination with a variable speed drive for flow rate regulation.

SELF PRIMING REQUIREMENT If a pump is taking suction from a source below the pump suction nozzle, a self-priming capability may be necessary. Positive displacement pumps such as a piston pump, rotary screw, or gear pump are able to self-prime within limits in the smaller capacity range. There are also special centrifugal pump designs that will self-prime in this situation.

LOW AVAILABLE *NET POSITIVE SUCTION HEAD* If the NPSHA is low, specially designed centrifugal pumps can be considered. Depending upon how low the NPSHA is, either a horizontal end suction with a suction inducer or a horizontal double suction arrangement may be applied. A vertical turbine pump may also be used, either immersed in the process fluid, possibly in a tank or vessel, or in a specially designed vessel (known as a suction can) that can be installed below grade to increase the NPSHA.

Evaluating Alternative Pump Selections Once the COS are established, a list of candidate pump product lines are identified (see Figure 7). The candidate product lines are evaluated or "filtered" based on technical criteria such as flow, head, suction pressure, temperature, viscosity, and solids handling capability. Further, other criteria such as material compatibility to the process fluid, pump-type classification, industry specification, pump orientation (vertical vs. horizontal), and industry application experiences also filter the choices to the "best" candidates.

Next, each pump model within the candidate product line families are evaluated against the conditions of service. These pump models are sized to suite the COS. For centrifugal pumps, this includes defining the operating speed, hydraulic design, impeller trim, number of stages, driver size, and materials of construction (see Figure 8). For positive displacement (PD) pumps, the hydraulic design may include size and number of plungers, gear size, or screw diameter and pitch. Equally important, the operating speed of a PD pump generally requires a speed reducer between the pump and the driver. Examples include gearbox drive, V-belt drives, or more popularly today, variable frequency drives (VFD).

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▼ Product Line Filters

Industry	Product Classification	Orientation	Minimum Material	Construction Specification
All	All	All	All	None

▼ Product Line List

Select	Product Line (All)	Maximum Flow (USgpm)	Maximum Head (ft)	Max. Suction Pressure (psi.g)	Temperature Limits (deg F)	Maximum Viscosity (cP)	Max. Solids Diameter (in)	Orientation
☐	Axial flow	252000.0	30.75	50.00	-50.00 to 275.0	2191.9	0.98	All
☐	End-suction, foot mounted (basic)	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
☐	End-suction, foot mounted (loaded motor speed)	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
	End-suction, foot mounted (variable speed)	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
☐	Radially split multistage	4402.9	6561.7	290.1	-40.00 to 392.0	1992.7	0.39	Horizontal
☐	Submersibles	4402.9	492.1	348.1	-4.00 to 572.0	1992.7	0.59	Horizontal
☐	Vertical turbine	35000.0	2000.0	150.0	35.00 to 180.0	1992.7	2.24	Vertical

FIGURE 7 Example of candidate pump product lines (Courtesy Intelliquip, LLC)

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Results List					Thumbnail Curves (All)					
Result	Pump Size	Display Thumb-nails	Stages	Speed Criteria	Pump speed, rated (rpm)	Pump Efficiency (%)	NPSHr (m)	Max. Power (kW)	% BEP	% Max Diameter or Pitch
✓	3X8 MULTI	☐	4	Sync.	1,775	73.77	1.19	21.37	80.51	98.53
✓	3X8 MULTI	☐	11	Sync.	1,185	72.27	1.03	17.85	120.42	98.53
✓	1.5X6 MULTI	☐	2	Sync.	3,550	71.17	2.79	20.57	105.27	100.00
✓	3X10 MULTI	☐	6	Sync.	1,185	70.70	0.66	17.76	107.09	98.26
?	2X6 MULTI	☐	9	Sync.	1,775	68.43	2.26	18.61	121.55	100.00
✓	2X8 MULTI	☐	5	Sync.	1,775	68.37	1.51	18.17	129.79	97.79
✓	2X6 MULTI	☐	10	Sync.	1,775	66.94	2.29	18.02	126.89	97.00
✓	80x50x200 IEQ	☐	1	Sync.	3,500	66.26	2.69	27.76	65.82	97.79

FIGURE 8 *Example of pump search results list (Courtesy Intelliquip, LLC)*

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Choices among different models are generally based first on technical requirements; namely, does the pump model satisfy all of the conditions of service? (Note: In Figure 8 this is designated by the "check mark" in the left most column.) Then, among technically compliant candidate pump models, the "best" pump is chosen based on criteria established by the purchaser. Typically, this is balance between cost (as depicted by the "Price Index" in Figure 8), pump efficiency, and pump reliability. Pump reliability and operating performance have been correlated to many factors, including the "fit" within the operating range with particular emphasis on how closely the rated flow corresponds to the best efficiency flow rate for the rated impeller diameter. These features are easily visualized using a family curve presentation as shown in Figure 9. Here, the rated condition is depicted graphically between the operating boundaries of the minimum and maximum impeller trim, and the minimum flow and runout flow.

LCC considerations of pumps and their included pumping systems are becoming more important. These will be discussed in a later section.

Upon selecting the "best" pump model, final pump performance calculations are generated, as shown in Figure 10. These calculations are necessary for the pump system engineer to use in making detailed comparisons between alternative pump offerings and in sizing the final piping and foundation design.

Pump Configuration In the previous section, the pump selection process was described, resulting in a pump model choice with supporting technical and pump performance data. The subsequent pump configuration process involves choosing among an array of option choices to complete the pump scope-of-supply. While the design, performance, and operating characteristics of the pump are substantially fixed by the pump selection, the pump configuration has a substantial influence on installation costs, maintenance, reliability, lead time, and quality of the pump package.

CONFIGURATION WORKUP The manufacturer's application engineer begins by preparing the configuration workup. This configuration workup is a methodical process of:

1. Interpreting the customer's pump specification,
2. Comparing the specification to the product standards available for the manufacturers product line
3. Making scope-of-supply choices that simultaneously adhere to the customer's specification and the product line standards, while providing an economical and quality technical solution

Seasoned applications engineers for pump manufacturers provide this expertise and perform this function. The configuration workup is accomplished by making option choices from different option categories, as described below.

MATERIALS OF CONSTRUCTION Most product lines have pre-engineered material choices that are offered. These range from standard materials specified by the American Petroleum Institute for various pump types (see API Standard 610, 674, 675, 676, 685) to nonmetallic choices offered for specialty services. Configuration choices often allow upgraded material choices for impeller, shafts, wearing rings, and so on to improve the service life and corrosion resistance of those components. Pump specifications often dictate the materials based on prior applications experience. Otherwise, the manufacturer is responsible to offer material recommendations or upgrades considering fluid compatibility, corrosion, applications experience, or operating experience. See Chap. 8 for more information on "Materials of Construction."

SHAFT SEAL AND SEAL FLUSH This choice of shaft seals is sometimes straightforward for well-known pump applications using off-the-shelf mechanical shaft seals, pre-engineered for use with each pump. Alternatively, some demanding process applications involving flammable and hazardous fluids at extreme temperature conditions (either cryogenic or high temperature) require very careful seal selections that can be more costly than the

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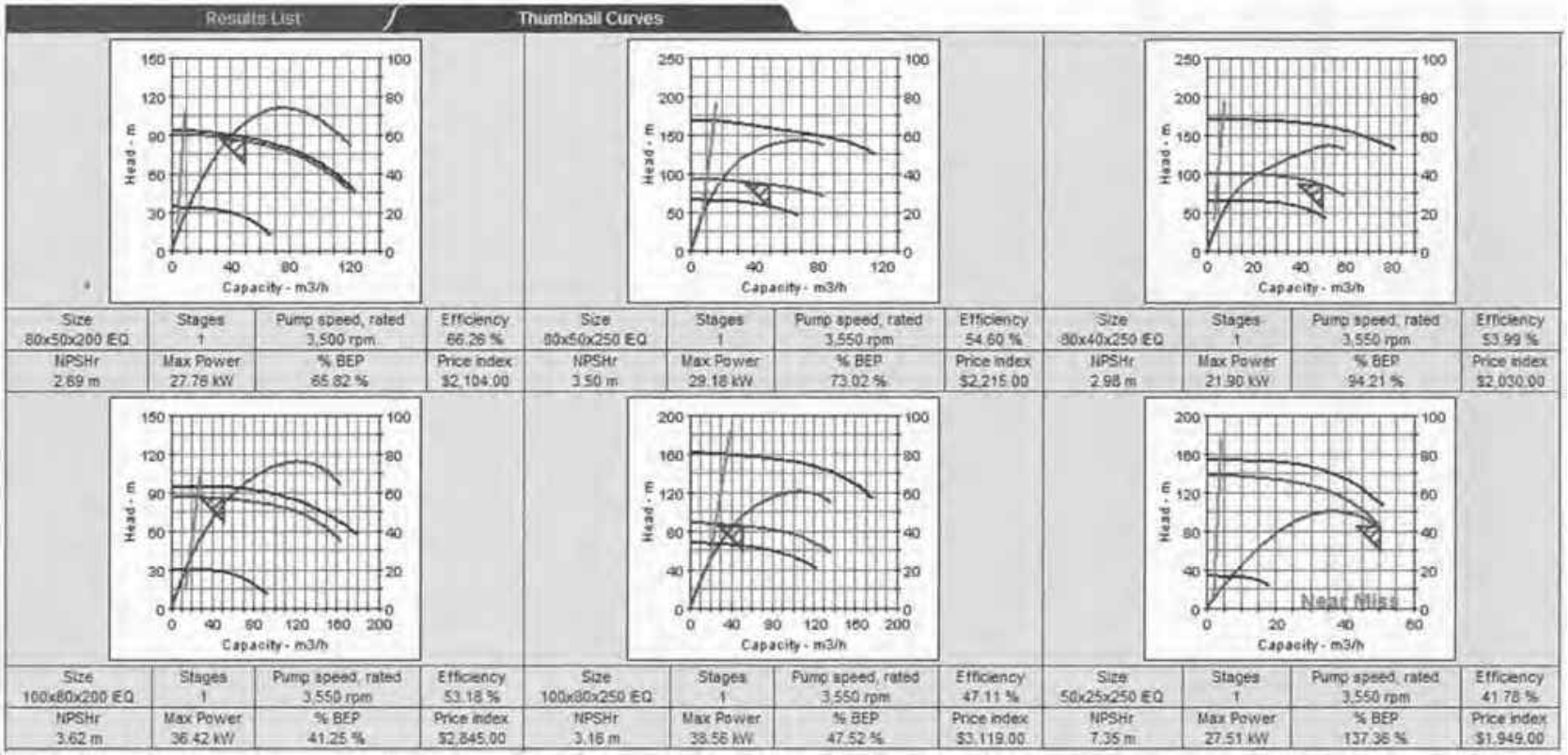


FIGURE 9 Collection of candidate pump family curves (Courtesy Intelliquip, LLC)

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Pump Performance Datasheet

Customer		Quote number	SP9_2
Customer reference		Pump size	80x50x200 IEQ
Item number	003	Stages	1
Service		Based on curve number	200IEQ-13 Rev 0
Quantity of pumps	1	Date last saved	07 Apr 2007

Operating Conditions		Liquid	
Flow, rated	50.0 m3/h	Liquid type	Water
Head, rated (requested)	85.0 m	Additional liquid description	
Head, rated (actual)	84.5 m	Solids diameter, max	0.00 mm
Suction pressure, rated / max	1.0 / 1.5 bar g	Temperature, max	20.00 deg C
NPSH available, rated	8.0 m	Fluid density, rated / max	0.996 / 0.996 kg/dm3
Frequency	60 Hz	Viscosity, rated	1.04 cSt
		-	

Performance		Material	
Pump speed, rated	3,500 rpm	Material requested	Grey cast iron
Impeller diameter, rated	211 mm	Material selected	Grey cast iron
Impeller diameter, maximum	216 mm	**Pressure Data**	
Impeller diameter, minimum	140 mm	Maximum working pressure	10.3 bar.g
Efficiency	66.3 %	Maximum allowable working pressure	19.7 bar.g
NPSH required / margin required	2.7 / 0.0 m	Maximum allowable suction pressure	19.7 bar.g
Specific speed / Suction specific speed	994 / 9,565 US units	Hydrostatic test pressure	31.0 bar.g
MCSF	7.7 m3/h	**Driver & Power Data**	
Head, maximum, rated diameter	90.2 m	Driver sizing specification	Rated power
Head rise to shutoff	5.9 %	Margin over specification	0.0 %
Flow, best eff. point (BEP)	76.0 m3/h	Service factor	1.00
Flow ratio (rated / BEP)	65.8 %	Power, hydraulic	11.5 kW
Diameter ratio (rated / max)	97.8 %	Power, rated	17.4 kW
Head ratio (rated dia / max dia)	96.8 %	Power, maximum, rated diameter	27.8 kW
Viscous coefficients (CQ / CH / CE)	1.00 / 1.00 / 1.00	Minimum recommended motor rating	18.6 kW / 25.0 hp
Selection status	Acceptable		

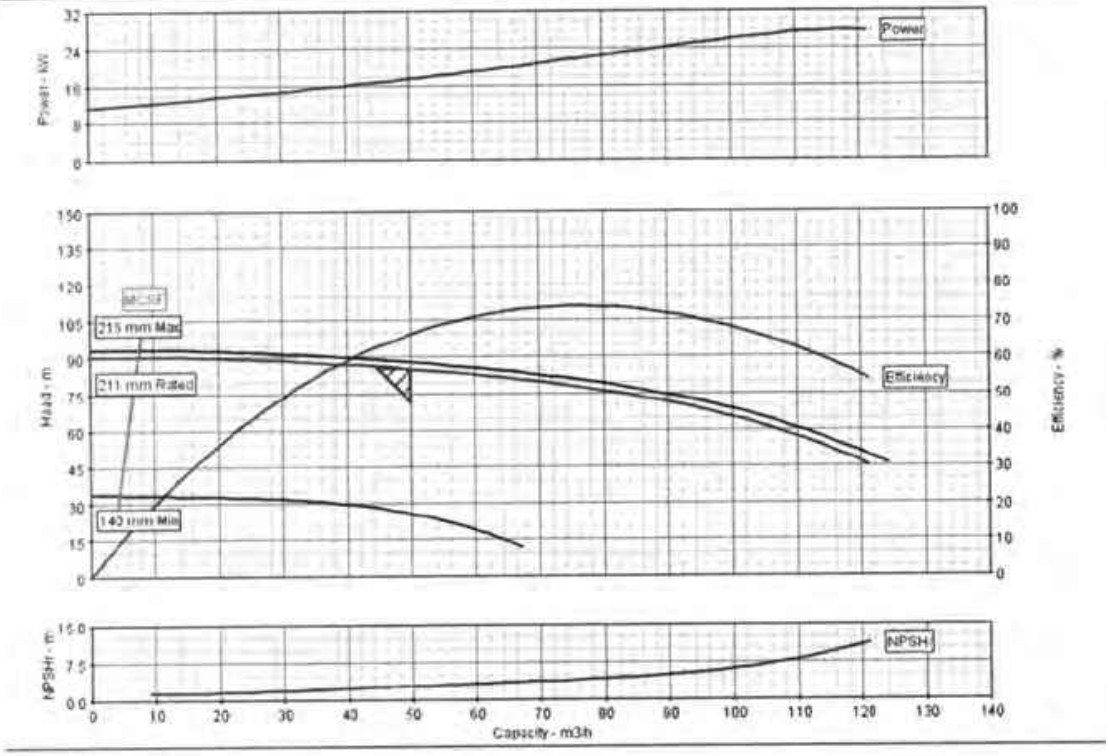


FIGURE 10 Pump performance data sheet (Courtesy Intelliquip, LLC)

pump itself. Mechanical seal vendors are generally consulted by the manufacturer to assure proper selection and configuration of the seal choice and corresponding seal flushing plans. See Chap. 5, or the API 682 standard for shaft sealing systems for centrifugal and rotary pumps is a useful industry reference.

DRIVER AND DRIVER MOUNTING The driver choice is typically outlined in the purchasers pump specification. The vast majority of drivers are synchronous speed electric motors. Variable speed drivers offer greater operational flexibility to vary output flow and

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performance. These include electric motors with variable frequency drives, engine drives, or steam turbines.

For both technical and commercial reasons, the purchaser may decide to purchase the pump/driver combination in various ways.

The pump and driver may be purchased separately. This may be advantageous if either the pump or driver (but not both) can be purchased locally. This will save shipping costs and possibly allow the purchaser to meet client requirements for locally manufactured content for a project. Separate purchase of drives on large capital projects can also lead to quantity price discounts and limited spare part inventories.

The purchaser must consider the risk associated with the separate purchase of the pump and driver. Equipment installation and alignment problems (with resulting start-up delay) are more probable than when the pump vendor takes single source responsibility for the purchase and skid mounting of all components.

If the decision is made to purchase the driver separately from the pump, the pump manufacturer can provide the equipment for block mounting or provide a skid on which the driver can be installed in the field. Either way, additional shop inspection is recommended to verify dimensions for field interface. It should be noted that pump manufacturers may purchase a high volume of electric motors and obtain greater discounts than most operating companies and engineering contractors.

See Chap. 9 for more information on "Pump Drivers."

QUALITY AND TESTING Depending on the pump service and technical risk associate with the equipment purchase, the pump specification may require special pump testing or quality requirements. Typical testing options are performance tests (hydrotest, pump performance, or *NSPHR*) that may provision for a customer to be present on-site to witness the test. See Chap. 16 for more information on "Pump Testing." Similarly, quality requirements often require material certifications or nondestructive examination of critical areas. The API Mechanical Equipment Standards such as API 610 and the Process Industry Practices (PIP) offer recommendations for pump quality requirements.

COMMERCIAL PRICE MAKEUP The configuration workup is used to prepare a commercial price makeup, as depicted in Figure 11. Various option categories and configuration option choices are depicted in this figure, as described in the previous section. In addition, the commercial price implications are also described. All price makeups involve engineering trade-offs between technically sufficient offers versus more robust but costly offerings. This is dictated by the pump specification and the skill of the applications engineer.

GENERAL ARRANGEMENT DRAWINGS General arrangement drawings are typically required to be provided by the manufacturer within the quotation submittal package. A sample drawing is found in Figure 12. The general arrangement drawing provides the overall outline dimensions of the pump package, as well as any other connections or interfaces that the purchaser/engineer must consider upon pump installation. These interfaces include the driver and electrical requirements, seal flush and cooling water piping connections. Weights of the pump, baseplate, and motor are also used in finalizing the foundation design.

For complex installations with specially procured drivers, seal flush plans, or baseplates, certified general arrangement drawings are often provided after the order is confirmed since design details are not yet finalized.

CONSTRUCTION DATA SHEET A construction data sheet is also provided as part of the quotation submittal package. Most manufacturers provide a proprietary data sheet automatically created from their quote generation system. An example is shown in Figure 13. In lieu of accepting the manufacturer's data sheet, many purchasers or specifiers require completing their own proprietary data sheet. Alternatively, an industry recognized data sheet like API or PIP is used. These alternative data sheet formats are laborious to fill out and to maintain through the life cycle of the quotation and procurement process. Improved methods using *electronic data exchange* are under development as described in a later section in this chapter.

Description	Qty	Extended Sell Price
80x50x200 IEQ	1	$ 2,442.00
Manufacturing Details		
Booking location:: Houston		
Bareshaft pump supply: Houston		
Unit packaging: Houston		
Construction		
Material		
Grey cast iron		Included
4140 carbon steel, sleeved shaft		
Cast iron impeller material		
Carbon steel shaft sleeve		
Standard material casing & gland gaskets		
Cast iron support head		
Cast iron bearing housing		
Mechanical		
PN10 flat face flanges		
Double row thrust bearings - oil lube		
Sealing Options		
Crane Standard Materials Seal 1A	1	$ 152.77
Casing cover type 2 - No flush tap		
No casing cover bushing		
Flush gland for flat gasket	1	$ 357.00
Seal flush piping options		
No seal flush piping		

FIGURE 11 Configuration options and pricing (Courtesy Intelliquip, LLC)

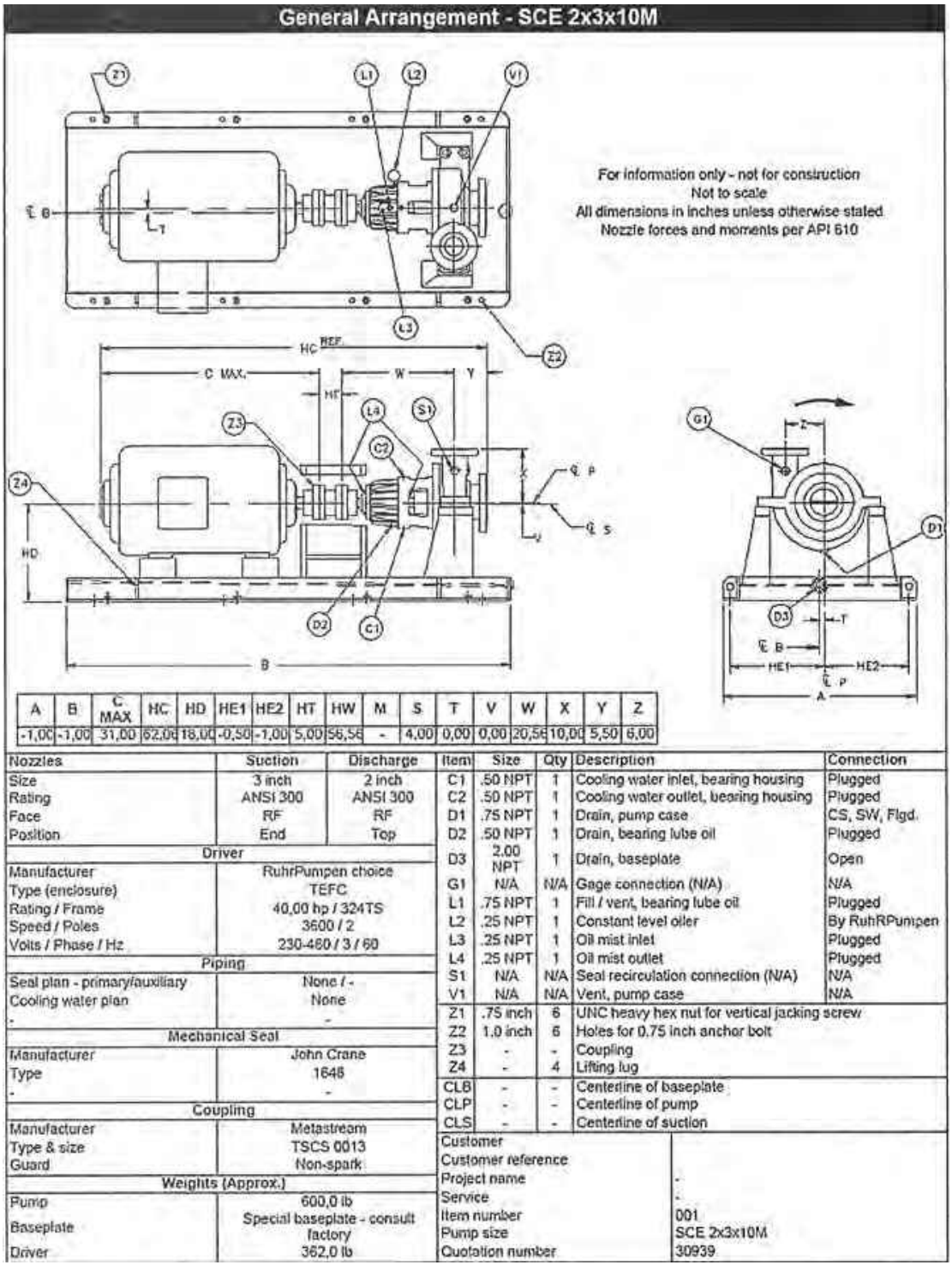
PUMP QUOTATION A final quotation submittal package is provided to the purchaser (the initiator of the RFQ). This contains documents such as a price makeup, performance curves, data sheets, and general arrangement drawings. Depending on specification requirements, additional documents such as a vendor drawing and data requirements, installation lists, or quality control plans may be required.

EVALUATING AND PURCHASING THE PUMPING EQUIPMENT

After the bid requisition has been prepared and sent to the approved pump vendors for quotation, purchasers begin preparing bid review documents. Both commercial and technical evaluations are necessary and can be accomplished on the same or separate documents.

Technical and Commercial Bid Evaluation Review strategies differ between projects and clients. Often, technical reviewers are not made aware of the quoted costs and are concerned only with technical specification compliance and scope of supply. In this case, the commercial evaluation is done by others. The final comparison of costs is made after the various vendors quotes are conditioned to meet all technical requirements. If the cost is withheld from the technical reviewer, an important piece of evaluating information is missing, which increases the chance of missing key requirements. Based on a quick

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A	B	C MAX	HC	HD	HE1	HE2	HT	HW	M	S	T	V	W	X	Y	Z
-1,00	-1,00	31,00	62,00	18,00	-0,50	-1,00	5,00	56,56	-	4,00	0,00	0,00	20,56	10,00	5,50	6,00

Nozzles	Suction	Discharge
Size	3 inch	2 inch
Rating	ANSI 300	ANSI 300
Face	RF	RF
Position	End	Top
Driver		
Manufacturer	RuhrPumpen choice	
Type (enclosure)	TEFC	
Rating / Frame	40,00 hp / 324TS	
Speed / Poles	3600 / 2	
Volts / Phase / Hz	230-460 / 3 / 60	
Piping		
Seal plan - primary/auxiliary	None / -	
Cooling water plan	None	
-	-	
Mechanical Seal		
Manufacturer	John Crane	
Type	1648	
-	-	
Coupling		
Manufacturer	Metastream	
Type & size	TSCS 0013	
Guard	Non-spark	
Weights (Approx.)		
Pump	600,0 lb	
Baseplate	Special baseplate - consult factory	
Driver	362,0 lb	

Item	Size	Qty	Description	Connection
C1	.50 NPT	1	Cooling water inlet, bearing housing	Plugged
C2	.50 NPT	1	Cooling water outlet, bearing housing	Plugged
D1	.75 NPT	1	Drain, pump case	CS, SW, Flgd.
D2	.50 NPT	1	Drain, bearing lube oil	Plugged
D3	2.00 NPT	1	Drain, baseplate	Open
G1	N/A	N/A	Gage connection (N/A)	N/A
L1	.75 NPT	1	Fill / vent, bearing lube oil	Plugged
L2	.25 NPT	1	Constant level oiler	By RuhRPumpen
L3	.25 NPT	1	Oil mist inlet	Plugged
L4	.25 NPT	1	Oil mist outlet	Plugged
S1	N/A	N/A	Seal recirculation connection (N/A)	N/A
V1	N/A	N/A	Vent, pump case	N/A
Z1	.75 inch	6	UNC heavy hex nut for vertical jacking screw	
Z2	1.0 inch	6	Holes for 0.75 inch anchor bolt	
Z3	-	-	Coupling	
Z4	-	4	Lifting lug	
CLB	-	-	Centerline of baseplate	
CLP	-	-	Centerline of pump	
CLS	-	-	Centerline of suction	

Customer	
Customer reference	
Project name	-
Service	-
Item number	001
Pump size	SCE 2x3x10M
Quotation number	30939

FIGURE 12 General arrangement drawing (Courtesy Intelliquip, LLC and Ruhrpumpen)

comparison of cost, it is easy for the document reviewer to establish the quoted general scope for each vendor. If large cost differences are noted between vendors, the reviewer must carefully determine what major differences in quoted scope likely exist.

The number of quotes that will be fully evaluated will depend partly on the number received. The goal is to ensure that the least expensive, technically acceptable bidder can be determined while also minimizing evaluation effort.

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Centrifugal Pump Data Sheet

1	Customer:	Abastible	Sulzer Tender ID:	001
2	Project Name:	-	Inquiry Date:	February 16, 2006
3	Inquiry Number/ID:		Bid Submitted Date:	-
4	Customer Item Number:	004	Revision Date:	May 9, 2006
5	Application:	-	Quantity:	1
6	**Operating Conditions**		**Pump Design Data**	
7	Liquid Type:	Water	Pump Type:	No Data
8	Temperature, Max:	20.00 deg C	Product Line:	AhlstarUp
9	Fluid Density, Rated/Max:	0.998 kg/dm3	Pump Size / No. of Stages:	A 11-50_O / 1
10	Vapor Pressure, Rated:	0.02 bar.a	Rotation (viewed from drive end):	No Data
11	Viscosity, Rated:	1.00 cSt	Suction Impeller Type:	Open
12	Consistency:	-	Casing Mounting:	-
13	Air Content:	-	Casing Split:	-
15	Discharge Flow, Rated:	68.46 m3/h	Casing Type:	-
16	Discharge Head, Rated/Actual:	50.84 m / 50.96 m	*Nozzle* / *Size* / *Rating* / *Face* / *Posn*	
17	Suction Pressure, rated/max:	0.00 bar.g / 0.00 bar.g	Suction: - / - / - / -	
18	NPSH Available:	Ample	Discharge: DN50 / PN16 / - / -	
19	**Performance**		Lineshaft Diameter:	-
20	Performance Curve No.(s):	K18709	Bearing Type, Radial:	-
21	Pump Speed:	2,980 rpm	Lineshaft Bearing / Bowl Bearing (Vert. pumps only):	-
22	Frequency:	50 Hz	Bearing Type, Thrust:	-
23	Fixed/Variable Speed:	-	Bearing Lubrication:	-
24	Impeller Diameter, Rated:	190 mm	Baseplate Type:	PMS 3
25	Impeller Diameter, Maximum:	210 mm	**Materials**	
26	Impeller Diameter, Minimum:	150 mm	API Material Class:	-
27	Efficiency:	70.44 %	Barrel/Can:	-
28	NPSH Required (3%) / Margin Required:	2.26 m / 0.00 m	Case / Bowls:	-
29	Ns / Nss:	1,177 US units / 11,175 US units	Column:	-
30	Head, Maximum, Rated Diameter:	54.12 m	Discharge Head:	-
31	Head Rise to Shutoff:	6.44 %	Impeller:	-
32	Flow, Best Eff. Point (BEP):	86.56 m3/h	Case/Impeller Wear Rings:	-
33	Diameter Ratio (Rated / Max):	90.48 %	Shaft:	-
34	Head Ratio (Rated Dia. / Max Dia.):	77.89 %	Diffusers:	-
35	Viscous Coefficients (CQ / CH / CE):	1.00 / 1.00 / 1.00	**Shaft Sealing, Flush & Cooling Piping Plans**	
36	**Pressure Data**		Seal Size / Type:	DS11 / -
37	Maximum Working Pressure:	5.30 bar.g	Seal Code:	-
38	Working Pressure Limit:	16.00 bar.g	Seal Manufacturer:	Sulzer
39	Suction Pressure Limit:	0.00 bar.g	Seal Flush Piping, Primary:	-
40	Hydrostatic Test Pressure (Suction/Discharge):	0.00 bar.g	Seal Flush Piping, Secondary:	-
41	Suction Pressure, rated/max:	0.00 bar.g / 0.00 bar.g	Cooling Water Piping:	-
42	Discharge Pressure, Rated:	4.98 bar.g	**Driver & Power Data**	
43	Differential Pressure, Rated:	4.98 bar.g	Driver Size:	18.00 kW
44	**Equipment Weights (Approximate)**		Volts/Phase/Frequency:	- / 2 / 50 Hz
45	Pump:	66.00 kg	Service Factor:	n.a.
46	Driver:	0.00 kg	Power, Rated:	13.42 kW
47	Baseplate:	140.0 kg	Power, Maximum, Rated Diameter:	17.80 kW
48	Total Package:	206.0 kg	Enclosure:	-
49	**Accessories**			
50	Driver:	-		
51	Coupling:	-		

FIGURE 13 Construction data sheet (Courtesy Intelliquip, LLC and Sulzer)

SHORT LISTING OF QUOTES When bids are first received, a quick commercial and technical tabulation is often prepared that lists the main scope of supply, extent of specification compliance, major deviations, and the associated total cost. This is done to get a general idea of the completeness of the quotes as well as to provide information to determine which of the quotes will be fully evaluated.

TECHNICAL BID EVALUATION Quotations must be evaluated against the specified requirements. A rigorous feature-by-feature comparison between alternative bids is often difficult,

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since a centrifugal pump may be compared against a positive displacement pump. In addition, each manufacturer's submittal requires a significant amount of effort and ingenuity in providing a cost-effective yet technically suitable pumping solution. Professional ethics must be respected to maintain the intellectual effort provided by each manufacturer.

After the decision is made on which bids to fully evaluate, the formal review and conditioning process can begin. A technical bid tabulation form is necessary to ensure that the equipment is provided in compliance with the requisition. This tabulation will list the important points of each vendor quote in table form. Each quote is then compared in a technical bid tab that includes:

- Equipment model number and size
- Compliance with process duty requirements (flow, head, NPSHR, and so on)
- Mechanical design limitations (pressure and temperature)
- Mechanical and hydraulic operating parameters (such as absorbed power, head rise to shutoff, capacity rise to run out, percentage of best efficiency point)
- Mechanical design features (such as materials of construction, seal type, seal flush type, bearing and lubrication type)

When the technical bid evaluation form is being completed, the reviewer must highlight all items that do not meet requisition requirements or that are not clearly defined in the vendor's quote. In addition, features of the vendor's quotation should be highlighted to clearly indicate advantages, disadvantages, ambiguities, and noncompliances. A typical API-610 pump technical bid evaluation is shown in Figure 14.

TECHNICAL QUOTE CLARIFICATION When the initial review is complete, a list of clarification questions needs to be prepared and sent to each vendor being evaluated. Questions should be direct and concise to ensure that responses answer the questions asked. The vendors must be encouraged to provide cost adders, if necessary, to meet requisition requirements. It is usually appropriate to allow the vendors 1 week to reply.

When answers to the clarification questions are received, the bid tabulation must be updated to include any revisions in the vendors quotation. Care must be exercised in tracking revisions to the original quotation. If additional clarification is needed, there are various ways to go about it, depending on the situation. Some suggestions are detailed in the following sections.

TECHNICAL PURCHASE RECOMMENDATION When all issues of bid requisition noncompliance have been satisfactorily addressed, the engineer must recommend which vendor or vendors may be considered for purchase of the required pump. Usually, part of this recommendation is the completed and signed technical bid tabulation. The final bid tabulation may show that all (or none) of the vendors are in complete compliance with all requisition requirements. The bid tab should clearly show where all noncompliance occurs. On the tabulation, a statement of technical compliance or noncompliance for each quote that has been reviewed must be provided. Notes that clarify any ambiguous line item on the tabulation are highly recommended so reviewers will have as few questions as possible on the bid tabulation content.

The bid tab should be attached to a recommendation memo or letter that states the recommendations and provides adequate justification. If a vendor is not in full compliance with the bid requisition, this should be noted, along with other reasons that support the recommendation. All revisions made since the original submission of the quotations that are necessary for purchase must be addressed. This might take the form of a scope of supply listing for each recommended vendor that clearly defines what must be purchased to meet the requisition requirements.

If a vendor is clearly favored over others for technical reasons, this must be clearly stated. Because the purchase order is usually given to the lowest cost technically acceptable bidder, this type of preference must be compelling.

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Bid Tab Comparison
Centrifugal Pump : P1252

Operating Conditions	Specified	Pump Mfg # 1	Pump Mfg # 2	Pump Mfg # 3
Flow, Rated (gal/m)	6,417	6,417	6,417	6,417
Discharge Pressure (psig)	155	155.1	155.1	155.1
Differential Pressure (psi)	101	101	101	101
Differential Head (ft)	323	323	323	323
NPSHA/NPSHR (ft)	85	18.7	18.7	18.7
Manufacturer		Pump Mfg # 1	Pump Mfg # 2	Pump Mfg # 3
Model Number		4 x 6 – 11	4 x 6 – 11	4 x 6 – 11
Material				
Barrel/Case		Carbon steel	Carbon steel	Carbon steel
Impeller		12% Chrome	12% Chrome	12% Chrome
Case/Impeller Wear Rings		12% Chrome / 12% Chrome	12% Chrome / 12% Chrome	12% Chrome / 12% Chrome
Shaft		AISI 4140	410SS	410SS
Performance				
RPM		3560	3560	3560
Impeller Diameter, Rated (in)		9.3125	9.3125	9.3125
Impeller Diameter, Max (in)		11.125	11.125	11.125
Impeller Diameter, Min (in)		8.0	8.0	8.0
Rated Power (bhp)		66.1	66.1	66.1
Efficiency (%)		71.5	71.5	71.5
Min Continuous Flow, Thermal (gal/m)				
Min Continuous Flow, Stable (gal/m)		2,671	2,671	2,671
Max Head @ Rated Impeller (ft)		351	351	351
Max Power @ Rated Impeller (bhp)	85	85	85	85
NPSHR @ Rated Flow (ft)		18.7	18.7	18.7
Max Suction Specific Speed	10660	10660	10660	10660
Coupling				
Coupling Manufacturer	Thomas	Thomas	Thomas	Thomas
Coupling Model Number	71	71	71	71

FIGURE 14 Pump technical bid evaluation

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For vendors that have been listed as technically acceptable, recommended for purchase, a list that addresses vendor's exceptions to requisition requirements should be attached. These exceptions should be justified in the technical purchase recommendation.

COMMERCIAL PURCHASE RECOMMENDATION Prior to completion of the technical evaluation, certain commercial items can be reviewed and negotiated with the vendors being evaluated. Terms and conditions of payment, guarantee/warranty, and proposed delivery schedule can usually be discussed in parallel with the technical conditioning of the bids.

After the technical recommendation for purchase is complete, the commercial purchase recommendation may be completed. In addition to including the technical bid analysis, this recommendation should cover the following items (if applicable):

1. Listing of original vendor bid scope of supply and the associated cost(s)
2. Listing of revisions to the vendor's original quotes needed for requisition compliance with the associated cost(s)
3. Required spare parts cost
4. Shipping costs
5. Miscellaneous costs associated with documentation, inspection, and so on
6. Utility consumption cost evaluation
7. Agreed delivery schedule
8. Exceptions taken to commercial terms and conditions
9. Economic adjustments for different terms of payment

At this point, the purchase recommendation can be made, usually based on the lowest cost from a technically acceptable bidder.

Final Purchase Decision and Purchase Order Issue Now that the commercial purchase recommendation is complete, it is forwarded to the parties responsible for making the purchasing decision. After the selection is made, the engineer must decide if additional discussions with the vendor are needed prior to issue of the purchase order. For complex, costly equipment, and large orders, this extra discussion often takes place in a pre-award meeting.

During this meeting, the final agreed-upon equipment, technical, and commercial requirements are confirmed. A final review of the vendor's technical exceptions to the requisition can also be completed at this point. As with previous meetings, recording all discussions and agreements is very important. Both parties should read and sign the meeting notes.

After final discussions are complete, the last step in purchasing the pump is the issue of the purchase order. Because the paper trail for the purchasing cycle can be very complex, it is strongly suggested that the purchase order be a stand-alone document. It may or may not be sufficient to only reference previously transmitted narrative specifications. Many options are likely to have been discussed and agreed to during the process. All these agreements, even if properly documented during the process, should be incorporated into the purchase order. This includes revising data sheets as necessary and also incorporating agreed vendor exceptions to the inquiry. Establishing "clean order requirements" at the purchase order will avoid costly interpretation, manufacturing errors, or shipment delays later in the process.

Often it takes several days (or weeks) to incorporate all agreements to the various documents that are contained in the requisition. In order to permit the vendor to proceed in advance of the formal purchase order issue, a letter can be written by the purchaser that confirms the order and requests the vendor to begin work immediately.

SUMMARY

Selecting and purchasing pumps is a complex but routine process that involves rich engineering experience from both the purchaser and supplier of the pumping equipment. The

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multistep process outlined in the chapter, involving: (1) engineering the pumping system, (2) pump selection and configuration, and (3) evaluating and purchasing the pumping equipment routinely followed for each new pumping system.

Many firms, including owners, operators, engineering/procure/construct (EPC) contractors, pump manufacturers, and pump distributors/packagers have all developed competencies and internal procedures to streamline the selection and purchasing process. This engineering effort can represent a significant portion of the overall equipment cost and has seen increased attention in recent years toward improving the overall efficiency.

The methods outlined in the previous sections represent the recommended approach using modern and effective practices. In the next section, emerging trends that seek to further improve the effectiveness of the selection and purchase of pumping equipment will be explored.

EMERGING TRENDS IN SELECTING AND PURCHASING PUMPS

Selecting and purchasing pumps involves the same basic process steps today as used in the recent past. While the process steps are the same, an engineer who performed those tasks 50 years ago would find many differences in the pump technology, the way purchasers and suppliers now interact, and the process automation tools used today. Globalization has had a profound effect in the way business is conducted. Rapid economic growth is now occurring outside of industrial nations in North America, Western Europe, Japan, and Australia. Information technology is enabling some of that growth as borders and distance are diminished through better communication technology and logistics to move people and materiel. On the other hand, this growth is placing a growing burden on the world's energy and natural resources. All of these factors are having an affect on the pump industry. The following sections will explore emerging trends affecting "how" the selection and purchase of pumps are taking place, and how they will continue to change in the coming years.

Pumping Systems Perspective

ENERGY EFFICIENCY The emphasis on energy conservation is becoming increasingly relevant. The world's demand for energy continues to expand, particularly in rapidly growing economies. In some cases, energy demand is exceeding availability. Energy costs continue to rise as sources of nonrenewable energy are depleted. Further, greenhouse gases created by industry, power production, and transportation are also affecting the world's environment. A Department of Energy report in 1998 reported that electric motor systems in the United States consumed over 679 billion kWh of electricity per year. Interestingly, it was found that pumping systems represented about 25% of that total (DOE 1998) in applications such as HVAC (heating, ventilation, and air conditioning), municipal water and wastewater treatment, and chemical process and refinery applications, to name a few.

The purchase and operation of pumping systems is a significant expense for most facilities. As energy costs rise, these expenses will reflect an increased proportion of overall facility costs. Ironically, most pumping system procurement decisions are based primarily on the initial purchase price of the pumping equipment. This is particularly prevalent in the purchase of new equipment where major expenditures are made on new capital facilities with business driven on fixed price contracts. However, it is well known that the initial cost of a pump is typically a small portion of the total LCC to operate the pumping system. Energy, maintenance, and other operating costs generally outweigh these initial costs. Therefore, it makes economic and environmental sense to consider these total lifetime costs when designing and procuring these pumping systems.

The selection and purchasing practices in the pump industry can thus have a relevant effect on worldwide energy consumption. Pumping system, energy efficiency improvement opportunities fall into two types, the purchase of new systems or the modification of existing systems. New systems need to be examined from an LCC perspective. Existing systems should not be ignored and actually represent a greater opportunity since the number of existing systems far exceeds the number of new systems installed each year. Efficiency

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improvements in existing systems normally require changes to the pump, the control system, or the incident piping systems. In many cases, changes to these systems are not economically practical. However, the DOE estimates that the average energy savings available in a typical installed system is 20%, through better pump selection, control systems, or piping system design. Certain installations may even enjoy much greater energy savings opportunities of 25 to 50%.

A SYSTEMS APPROACH The importance of proper pumping system design is becoming more evident as energy efficiency and LCCs are more broadly evaluated in the purchasing decision. While this transition has been evolutionary rather than revolutionary, the effort to reorient the evaluation process in the pump industry is underway. At the macroeconomic level, energy costs and diminishing resources will motivate change. At the industry level, new initiatives are under development to educate and provide tools to help engineers to evaluate pumps from a broader systems perspective. The Hydraulic Institute (HI) has historically provided pump standards but is now extending its reach to include standards, guidelines, and educational programs on pumping systems. The HI and Europump, a similar organization of European pump manufacturers, collaborated to create two user guides on the subject; *Pump Life Cycle Costs: A Guide to LCC Analysis for Pumping Systems;* and *Variable Speed Pumping: A Guide to Successful Applications*. More recently, the HI has organized national pump systems educational initiative called *Pump Systems Matter* TM (HI 2004b). This initiative is directed toward shifting the focus to LCC metrics to realize improved energy efficiency and longer-term performance and cost improvements. This initiative is providing educational materials suitable for facilities and maintenance personnel, specifiers, purchasers, and manufacturers alike. The PSM initiative is an industry-driven and industry-organized national market transformation initiative that complements broader energy savings efforts sponsored by government organizations.

PUMP SYSTEMS OPTIMIZATION TOOLS Pump systems optimization has been a labor intensive process requiring expertise in the application and design of pumping systems. The educational component of the "systems approach" is important, but tools (i.e., computer software) to aid in the detailed calculations of LCCs and pump system analysis and optimization are also needed. These are discussed in the section on "Process Automation Technologies."

Selecting and Purchasing Pumps in the Information Age Many firms are applying emerging information technologies during the process of selection and purchasing pumps to improve their competitiveness. Computer-based applications are often used to aid in generating pump proposals or to check part inventory status. Design departments use CAD/CAM systems to shorten the design cycle and run simulations using structural finite element methods. However, the use of computer technology alone does not guarantee a measurable economic benefit.

Organizational and process changes are usually necessary to achieve the benefits of computer automation. These process improvements often extend beyond a single firm with the formation of formal and informal alliances between pump users, owners, engineering contractors, architect-engineers, pump suppliers, and other equipment suppliers. These trends place a greater emphasis on the purchaser–supplier interface during the pump selection process. From the purchaser's perspective, the optimal choice of pumping equipment has significant cost implications over the service life of the equipment. From the manufacturer's perspective, configuring the preferred offering of pumping equipment is crucial in securing a competitive advantage during the purchase evaluation.

The information age is redefining virtually all aspects of conducting business including the way pumping systems are designed, evaluated, procured, manufactured, and maintained through their entire life cycle. In this section, the process of selecting and purchasing pumps will be revisited in the context of the way critical information is communicated between pump purchaser and supplier. Then, four of the emerging information technologies that are contributing to improved quality and cycle time in the overall process will be described.

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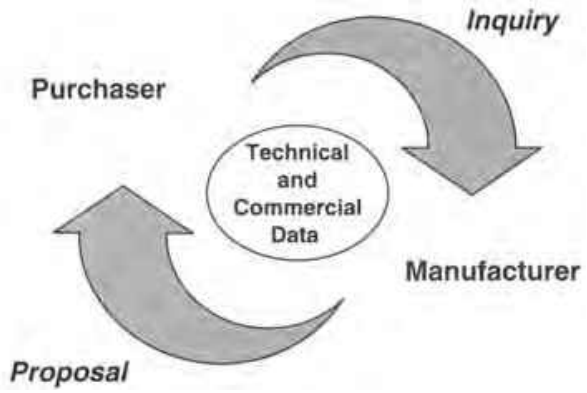


FIGURE 15 Purchaser–supplier information exchange

INFORMATION FLOW BETWEEN PURCHASER AND SUPPLIER Following the decision that pumping equipment is required, a selection and purchasing process is undertaken involving: (1) engineering the pumping system, (2) pump selection and configuration, and (3) evaluating and purchasing the pumping equipment. The entire process is information intensive, consisting of both technical and commercial information, as shown in Figure 15. The first two steps of the process (steps 1 and 2) are technical in nature, involving the exchange of system design, pump specifications, and performance and construction details of the pump. The last step of the process (steps 3) transitions toward the commercial elements of the purchasing decision such as equipment costs, LCC evaluations, terms and conditions, and delivery lead times. This flow of technical and commercial information is described by the inquiry-quotation information exchange.

This inquiry-quotation information exchange is not limited to only one purchaser-supplier interaction. With each new procurement opportunity, this information exchange effects every trading partner participating in the entire supply chain. Consider the simplified example (in Figure 16) of an operating company that gives three engineering/design contractors the opportunity to bid on an expansion of a chemical process plant. If each of these contractors issue three inquiries to pump manufacturers, nine inquiries are issued. Now, if each manufacturer issues three inquiries to their subsuppliers, a total of 27 inquiries are pending for this single plant expansion. In the end, only one contractor, one equipment-supplier, and one subsupplier actually receive orders to supply equipment for the project. Thus, only 3 inquiries out of 27 representing 11% of the total application and

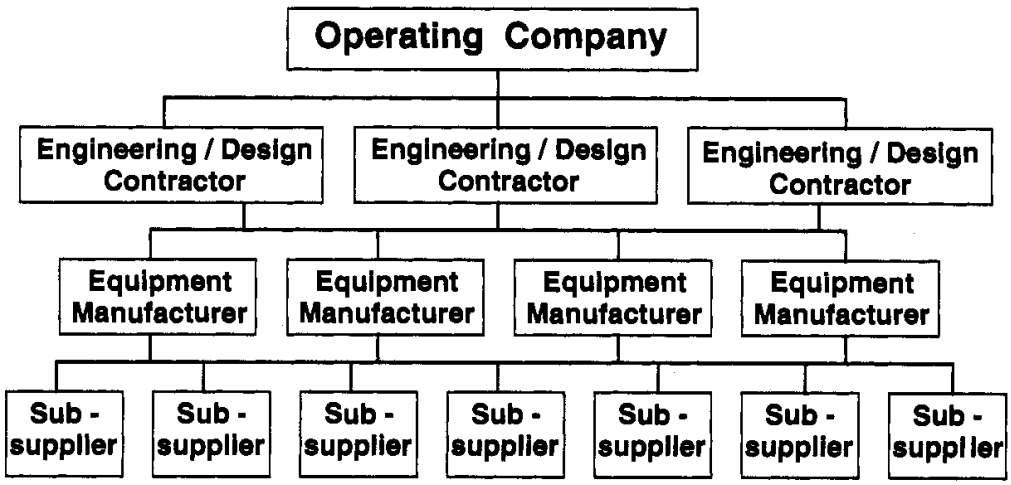


FIGURE 16 Multifirm information flow

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quotation effort represent "useful" work. These engineering costs are recovered only when equipment is actually purchased. The cost of the other 89% of effort by those participants in the inquiry/proposal process who did not receive a customer order are "wasted" and absorbed as a sales and engineering overhead. This example actually understates the total number of inquiries issued indicating that the "wasted" effort is probably much greater than calculated.

To reduce costs, purchasers are seeking ways to reduce the engineering resources required to process this information without compromising the quality of the competitive evaluation. Similarly, suppliers are developing methods for responding to inquiries (or requests-for-quotation [RFQ]) with less resources and effort while striving to still provide high quality quotations to their customers. Simplifying the inquiry-quotation process is an attractive proposition for both purchaser and manufacturer.

STRATEGIES FOR IMPROVING THE SELECTION AND PURCHASING PROCESS Both purchaser and manufacturer in the inquiry/quotation process are striving to enhance their competitiveness by reducing costs through process and information technology-oriented productivity improvements. Intracompany productivity gains have been the focus of many internal reengineering programs within a given firm. By their nature, these gains are limited to changes made within a single company and cannot effect improvements between multiple companies. However, the "ripple effect" on the total cost across multiple firms, described earlier, is a significant cost and represents a relatively unexplored opportunity for intercompany productivity gains. Some of these cooperative strategies engaged by both purchaser and supplier include the following:

- Form alliances to streamline the inquiry/quotation process and flow of information across multiple firms
- Reduce the number of participants in the process by defining a preferred set of customers and suppliers
- Move the pump selection task earlier in the pumping system design process by providing purchasers' access to computerized selection programs allowing the purchaser to optimize their process design
- Structure the flow of information between companies to reduce ambiguity and enhance common work processes

Existing and emerging information technologies offer a technology platform necessary to implement many of these strategies. The next section describes four system and information technologies that are particularly important in improving the selection and purchasing process for pumping equipment.

System and Information Technologies Information technologies are used throughout the design, manufacture, and maintenance of pumping equipment and systems. With respect to the specific activity of selecting and purchasing pumps, there are four system technologies that are gaining broader popularity and are playing a more meaningful role, including: (1) pump system analysis software, (2) pump system and selection programs, (3) pump configuration and quotation systems, and (4) electronic data exchange.

PUMP SYSTEM ANALYSIS AND OPTIMIZATION Designing the piping network and sizing the components for a pumping system are performed very early in the overall pump selection and purchasing process. The piping system design involves numerous components that introduce friction losses in the system. These must be calculated in order to estimate the behavior of the system resistance curve needed to properly size the pumps needed in the system. Changes in process conditions (pressure, temperature, fluid properties, tank elevation, and so on) or multiple/variable branch systems introduce additional operating conditions in the system design that must be predicted. This design process, when done manually, is tedious and time consuming. However, handling design iterations and performing system optimization studies have become more practical with the use of computerized design and analysis programs.

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Use of design and analysis programs became more prevalent in the mid-1980s with the advent of personal computers as powerful engineering workstations on the desktops of pumping equipment designers. These computers, with their easier to use software interfaces, encouraged the development of more robust and capable software applications that were more economical to deploy. Consequently, a substantial amount of effort went into the development of engineering programs dedicated to the sizing of piping systems, pumps, and other components.

Numerous piping system design/analysis computer applications are now available to aid in the design or analysis of piping systems and their components. Most of these programs are based on simultaneous path solutions. The more complete programs easily model piping networks and are capable of calculating friction losses through pipes and fittings. They also contain large libraries of standard pumped liquids and their fluid properties such as density, viscosity, and vapor pressure as a function of temperature. These programs are more time efficient and give more predictable results than former manual methods. An example is shown in Figure 17.

PUMP SELECTION SYSTEMS Most pump selection processes are now performed using computer-aided selection programs. Pump manufacturers performance curves are digitally programmed and accessed by the program to establish preliminary pump performance and design specifications. These programs can be reasonably sophisticated, using specialized mathematical algorithms to predict pump performance under varying operating conditions of speed, temperature, *NFSHA*, pressure, viscosity, and so on. Some are even capable of adjusting performance based on alternative mechanical seal design,

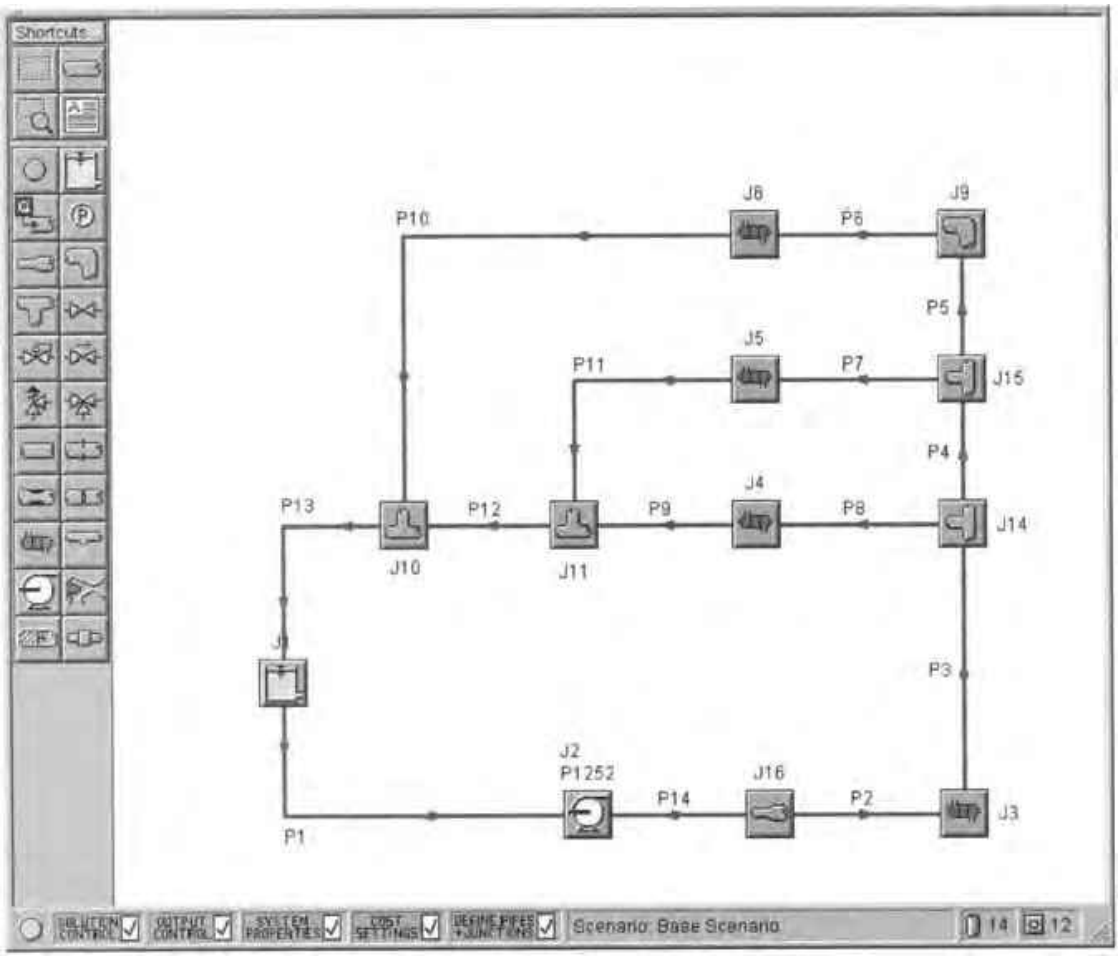


FIGURE 17 Pump system analysis and design schematic (Courtesy Applied Flow Technology)

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wearing ring design and clearance, bearing design, materials of construction, or other mechanical design features. Most pump manufacturers now offer some form of pump selection programs. In some cases, these pump selection programs are specially developed by the pump manufacturer using proprietary selection and searching methods. These programs are available under a licensing agreement directly from the pump manufacturer and are usually the same programs used by their internal applications engineers. Third-party programs are now available incorporating product lines from multiple pump manufacturers or customized to meet the specific needs of each manufacturer. The newest selection programs are designed to work directly from a browser on the Internet, providing immediate access to end users, specifiers, and applications engineers. These are convenient since special software is not required, data are easily shared between purchaser and supplier, and concerns over software updates become irrelevant.

Pump system and selection programs have improved the ability to evaluate large numbers of alternative design alternatives in a short period of time. Accurate performance calculations, even with variable pump speeds, fluid properties (viscosity, temperature, pressure), and mechanical configurations are readily predicted. Design and calculation errors have diminished and the overall quality of the process has greatly improved (Dahl and Ochs, 1997)

LCC CALCULATIONS One of the valuable features of a modern pump selection program is the ability to efficiently calculate LCCs. LCCs contain many different cost elements, as described in the earlier section, *Life Cycle Cost Evaluation.* In the example shown in Figure 18, the LCC equation is organized into the following sections:

Load Profile and Energy Costs The energy cost (C_e) is readily calculated with knowledge of the pump performance characteristics and the different load profiles in which the pump is expected to operate during its lifetime. With the optimization features of the LCC module, a comparison between fixed speed operation and variable speed operation is conveniently calculated, including the recalculation of pump performance coincident with the defined system curve(s).

Additional One Time Costs at Year 0 These are the initial costs (C_{ic}) of the pump and the installation and commissioning costs (C_{in}).

Additional Annual Costs These include the annual operating costs (C_o), maintenance costs (C_m), down time costs (C_s), and environmental costs (C_{env}).

Additional One Time Costs at Year, N The decommissioning costs (C_d) at the end of the service life of the equipment (year "N") is the final component of the cost model.

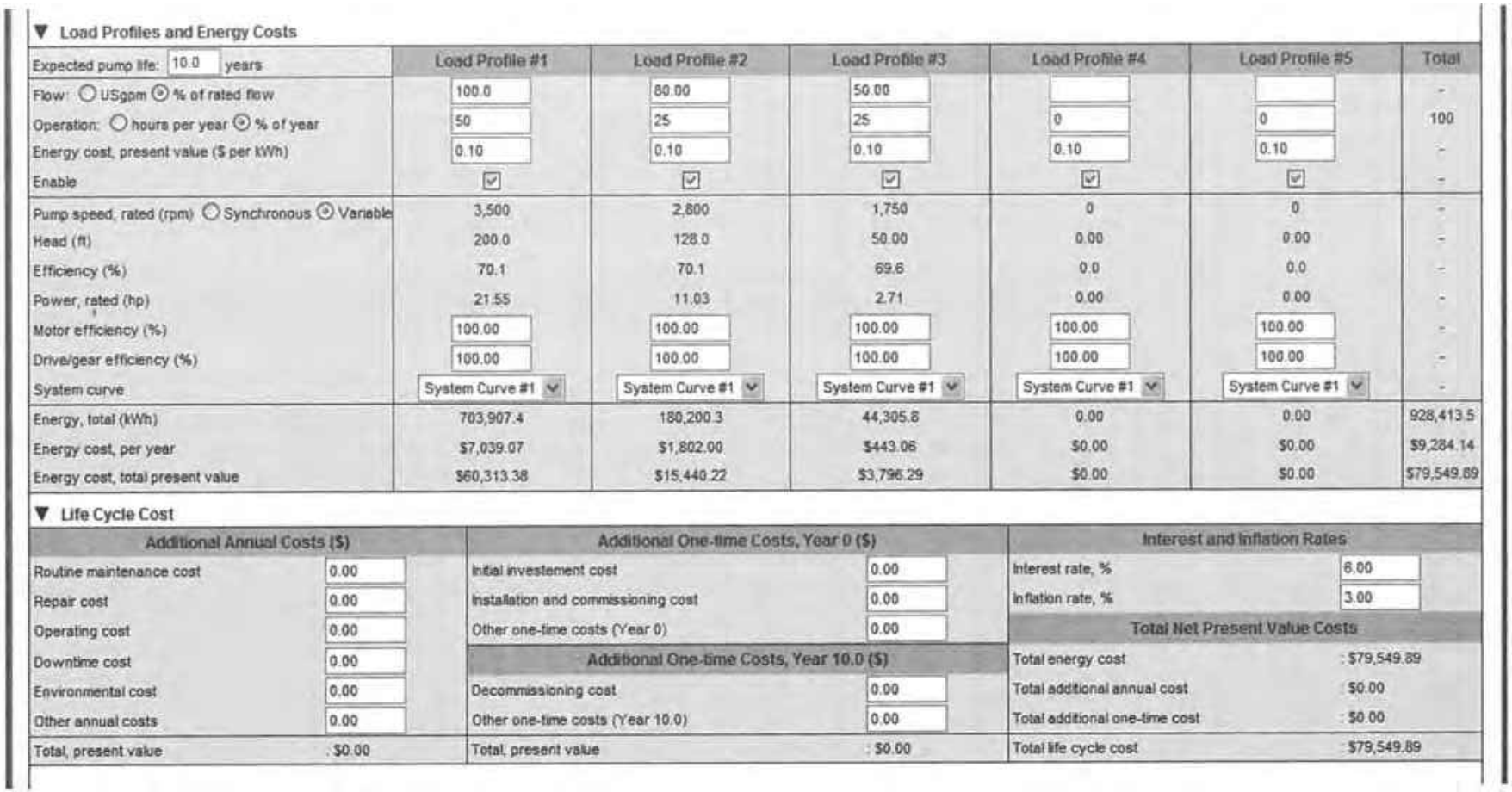
The additional costs are more difficult to model and are generally obtained from the pump manufacturer's quotation, estimates from the construction engineer, or operating experience from the owner/operators maintenance logs (see Hodgson and Walters, 2002).

Finally, while not always considered, the time value of money may be considered for both interest and inflation rates to calculate a net present value for the LCC.

PUMP CONFIGURATION AND QUOTATION SYSTEMS Pump quotations are complex documents containing a myriad of technical and commercial information. These quotations typically include data sheets, performance curves, general arrangement drawings, and a price makeups including terms and conditions. In many cases, alternative selections, comments to the customer's specification, or other supplementary information are also provided. The diverse array of information, expertise, and resources needed to generate a customized proposal has prompted the need to systematize the configuration and quote development process using computer-aided tools.

Some manufacturers have responded by implementing computerized product configurators. Product configurators aid the applications engineer in developing a pump quotation according to a prescribed product configuration model. These configurators are based on design guidelines and configuration rules that limit configuration choices. Examples of design rules are maximum casing pressure, maximum pump torque for a given shaft

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▼ Load Profiles and Energy Costs

Expected pump life: 10.0 years	Load Profile #1	Load Profile #2	Load Profile #3	Load Profile #4	Load Profile #5	Total
Flow: ○ USgpm ◉ % of rated flow	100.0	80.00	50.00			-
Operation: ○ hours per year ◉ % of year	50	25	25	0	0	100
Energy cost, present value ($ per kWh)	0.10	0.10	0.10	0.10	0.10	-
Enable	☑	☑	☑	☑	☑	-
Pump speed, rated (rpm) ○ Synchronous ◉ Variable	3,500	2,800	1,750	0	0	-
Head (ft)	200.0	128.0	50.00	0.00	0.00	-
Efficiency (%)	70.1	70.1	69.6	0.0	0.0	-
Power, rated (hp)	21.55	11.03	2.71	0.00	0.00	-
Motor efficiency (%)	100.00	100.00	100.00	100.00	100.00	-
Drive/gear efficiency (%)	100.00	100.00	100.00	100.00	100.00	-
System curve	System Curve #1	System Curve #1	System Curve #1	System Curve #1	System Curve #1	-
Energy, total (kWh)	703,907.4	180,200.3	44,305.8	0.00	0.00	928,413.5
Energy cost, per year	$7,039.07	$1,802.00	$443.06	$0.00	$0.00	$9,284.14
Energy cost, total present value	$60,313.38	$15,440.22	$3,796.29	$0.00	$0.00	$79,549.89

▼ Life Cycle Cost

Additional Annual Costs ($)		Additional One-time Costs, Year 0 ($)		Interest and Inflation Rates	
Routine maintenance cost	0.00	Initial investement cost	0.00	Interest rate, %	6.00
Repair cost	0.00	Installation and commissioning cost	0.00	Inflation rate, %	3.00
Operating cost	0.00	Other one-time costs (Year 0)	0.00	**Total Net Present Value Costs**	
Downtime cost	0.00	**Additional One-time Costs, Year 10.0 ($)**		Total energy cost	$79,549.89
Environmental cost	0.00	Decommissioning cost	0.00	Total additional annual cost	$0.00
Other annual costs	0.00	Other one-time costs (Year 10.0)	0.00	Total additional one-time cost	$0.00
Total, present value	$0.00	Total, present value	$0.00	Total life cycle cost	$79,549.89

FIGURE 18 Life cycle cost evaluation sheet (Courtesy Intelliquip, LLC)

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design, or the allowable temperature range for a given material. Using these design rules, a configurator could automatically upgrade a flange rating on a casing based on casing pressure or restrict the use of a gasket material as a function of temperature. Similarly, configuration rules ensure that a complete and permissible pump scope-of-supply is generated. For example, the mechanical seal type dictates the list of allowable seal flush piping options or the choice of pump and motor automatically defines coupling size required. These rules are coded into a knowledge base and serve to enforce product standardization and reduce configuration errors.

The pump selection program and the product configurator are the hub of the computer-aided tools used by the pump supplier to produce customized quotations. Supplementary programs are used to develop additional information such as pump performance curves, data sheets, general arrangement/outline drawings, and price makeups. These programs have reduced the quotation cycle time while improving the quality and accuracy of the technical and commercial information required by the purchaser.

A graphical description showing the interface of a pump selector and configurator are found in Figure 19. Important features of this model are:

- Workflow modeled after the well-known steps in the selection and configuration of pumping systems starting with defining the conditions of service, product selection and pump selection, and ending with configuration and quote generation.
- A rules engine and associated knowledgebase. Together, these assure reliable and consistent application of design rules, applications guidelines, and expert calculation methodologies traditionally known to only the most seasoned applications engineers.
- One knowledgebase referenced to produce all relevant quotation documents, including technical data sheets, general arrangement drawings, and price makeups.

ELECTRONIC DATA EXCHANGE The communication of inquiry and quotation information is still predominately managed through paper-based methods in spite of the increasing use of computer-aided design, selection, inquiry, and quotation systems. The exchange of pump technical data has been traditionally handled using the pump data sheet. Both purchaser and manufacturer have developed their own data sheet formats. Consequently, each data sheet exchange requires a laborious translation and interpretation of information from one data sheet format to another. This interpretation and translation requires highly skilled engineers and can be a source of errors and delays in completing the selection and purchase decisions.

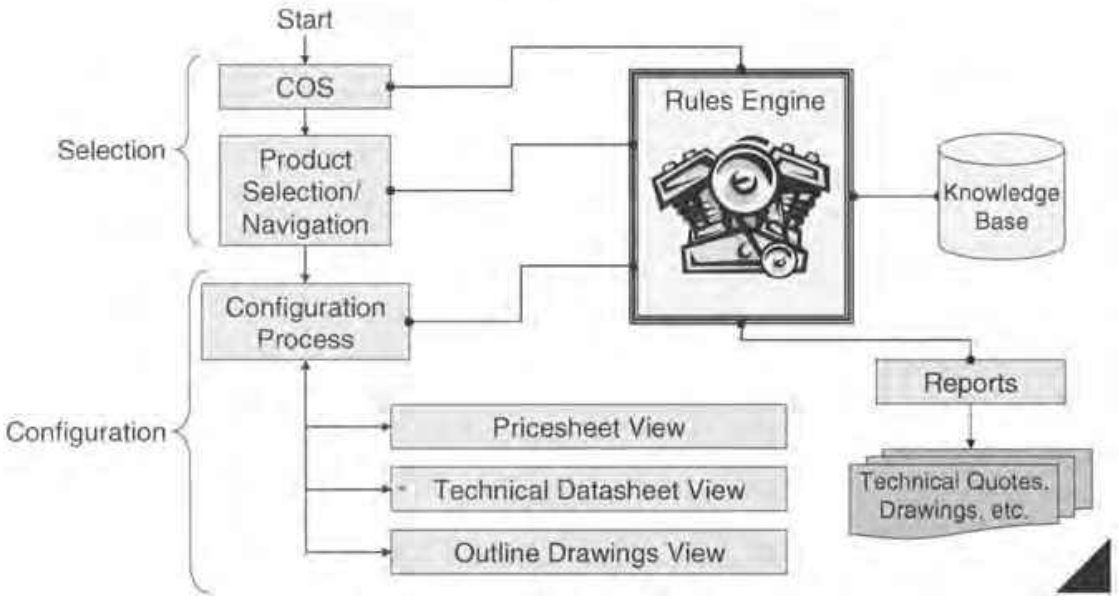


FIGURE 19 Selection and configuration workflow process (Courtesy Intelliquip, LLC)

With the implementation of computerized selection programs and bid-tab programs, this manual translation is an untapped opportunity to streamline the data communication and purchasing process. In response to this situation, some firms have developed proprietary data exchange formats to leverage their own proprietary systems. Unfortunately, that approach has only shifted the burden to their trading partners who must develop special data translators to accommodate this wide variety of data sheet formats. It has been shown that the total number of translators needed for M data exchange formats follows the relationship, $M \sim (M - 1)$. Therefore, 3 unique data formats require 6 translators, whereas 10 unique formats require 90 translators.

To address this problem, standards organizations have developed neutral data exchange specifications to perform electronic data interchange (EDI). Well-known standards include UN/EDIFACT and ANSI X12 to support basic buyer and seller transactions. However, exchange of technical information such as that found in purchaser–supplier transactions in the pump industry is not supported by these EDI standards. Electronic data exchange of pump technical data is only possible through the efforts of pump industry related standards organizations. One such specification was developed under the auspices of the American Petroleum Institute's Standard 610, Centrifugal Pumps for Petroleum, Heavy Duty Chemical, and Gas Industry Services (eighth edition, August, 1995). Since that time, the emergence of the World Wide Web on the Internet, and XML have provided a more fertile foundation for pump-related electronic data exchange.

To address these challenges and the need to automate the reliable exchange of equipment information, the FIATECH organization started the AEX (Automating Equipment Information Exchange) Project. This nonprofit industry consortium is working with other industry organizations and standards organizations to provide electronic data exchange methods to support the design, selection, and procurement of equipment involved in capital facilities projects, including not only pumps but valves, heat exchangers, compressors, motors, seals, and so on. AEX is now delivering XML schemas for a number of these equipment types. Simply stated, XML is a user-definable text file format that provides the capability to store highly structured digital information in a standard, *software neutral* text file. XML will work on virtually any computer hardware and operating system platform and with any software program. As a result XML can be used both for software neutral data exchange today, and for long-term data archival and reuse. This work, in coordination with other standards organizations such as the Hydraulic Institute (HI), the American Petroleum Institute (API), American Society for Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE), Design Institute for Physical Properties (DIPPR), and Process Industry Practices (PIP) are expected to be used in actual purchaser–supplier transactions in the next few years.

THE INTERNET The use of the Internet has grown dramatically since the advent of the World Wide Web in the mid-1990s. The rate of change on the Internet is witnessed in terms of days rather than years. Information or Web sites that exist on the Internet today may be eliminated or replaced by tomorrow. As a consequence, any references about the use of the Internet for the selection and purchasing of pumping equipment will most likely be obsolete at the time the reader sees it. Nonetheless, it is necessary to anticipate the effect the Internet will have on this process in the pump industry.

The Internet is rapidly becoming the medium by which business-to-business (B-to-B) or electronic commerce will be conducted between trading partners in the future. The pump system design, selection, configuration, and purchasing systems are rapidly becoming Web-enabled, allowing users to perform these activities directly on the Internet. Similarly, the process of electronic data exchange will be performed across the Internet using structured data exchange methods that eliminate the manual reentry of technical information between different systems. Standards created by either international standards bodies or a de facto standard created by a consortium or commercial entity will provide the necessary framework for this data exchange.

Finally, the actual business transactions involving inquiry, quotation, purchase order, and invoice will expand over the Internet. This process will evolve starting with commodity-type pumps and eventually will involve more sophisticated types of pumping equipment. Interactions during the order fulfillment cycle will be available such as order

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status information. Drawings, certifications, test data, and other technical information will be available as secure business transactions. The use of voice and video are already supplanting many face-to-face meetings or equipment inspections that often involve detailed coordination and expensive travel.

SUMMARY

The basic activities involved in the selection and purchasing of pumps as described at the beginning of this chapter are substantially the same now as in the past, and will be in the future. However, the processes and technologies employed in performing these activities are changing rapidly because of new system and information technologies, the influence of globalization and the cost of energy. The motivations surrounding these changes are driven by the desire for shorter cycle times, higher quality, and lower costs in the entire pump system life cycle. These objectives are driven by the availability of new and emerging information technologies that offer a seamless and structured flow of information between the purchaser's and the supplier's sales, applications, engineering, and manufacturing functions. The availability of computer systems guarantees only that the infrastructure is in place to achieve the anticipated benefits. However, common work practices in both an intercompany and intracompany environment must be adopted and adapted to these new technologies. These process changes, not the availability of new information technologies, will govern the speed at which the pump industry changes in the future. The information age will support, but not replace, the basic requirement to design, select, configure, and manufacture high-quality and reliable pumps that support a substantial part of our modern life and industry today.

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C • H • A • P • T • E • R • 15

INSTALLATION, OPERATION, AND MAINTENANCE

Igor J. Karassik

C. C. Heald

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The information contained in this chapter is general and should be supplemented by the specific instructions prepared by the manufacturer of the pump in question.

INSTALLATION

Instruction Books Instruction books, or Technical Manuals, are intended to help keep the pumps in an efficient and reliable condition at all times. It is necessary, therefore, that instruction books be available to all personnel involved in this function.

Preparation for Shipment After a pump is assembled in the manufacturer's shop, it should be suitably prepared for the type of shipment the purchaser has specified. This can include blocking of the rotor, when necessary. If the rotor is blocked, this should be identified by weather-resistant tags. As shipped, the equipment should be suitable for at least 6 months of outdoor, uncovered storage from the time of shipment, with no disassembly required at the time of installation and operation except for possible inspection of bearings and seals. If storage for a longer time is required and specified, the manufacturer and purchaser should agree on the preparation and storage procedures and requirements prior to shipment.

The pump and driver should be prepared for shipment only after all testing and inspection processes have been completed and the equipment has been released for shipment by the purchaser. If packing was used for testing, it should be removed before shipment. Usually pumps are not disassembled after performance testing. The pump must be completely drained and dried, and all internal parts should be coated with a suitable rust preventative within 4 hours of testing. Alternatively, within 4 hours of testing, the pump and seal chamber should be drained to the extent practical, filled with a water-displacing inhibitor, and redrained before shipment.

Flanged openings should be provided with covers, usually metal, at least $\frac{3}{16}$ in (5 mm) thick and sealed with an elastomeric gasket. Threaded openings should be closed with steel caps or plugs. Any openings that have been prepared for welding in the field should be provided with closures designed to protect against damage to the welding surface as well as the entrance of foreign materials into the equipment.

Exposed shafts and shaft couplings should be wrapped with waterproof cloth or paper and sealed with oil-proof adhesive tape. Bearing assemblies should be protected from moisture and dirt (dust). If vapor phase inhibitor crystals in bags are installed to absorb moisture, the location of the bags should be tagged so they will be removed before installation of the equipment in the field.

Usually a copy of the manufacturer's standard installation instructions is packed inside the equipment prior to shipment.

Care of Equipment in the Field The manufacturer should provide the purchaser with the instructions necessary to preserve the integrity of the original storage preparation when the equipment is received at the job site and prior to start-up. All equipment and materials should be stored free from direct ground contact and away from areas subject to collecting water. Indoor storage should be used whenever possible.

All carbon and low alloy steel surfaces should be protected from any contact with corrosive environments to prevent rust formation. All items with machined surfaces should be stored to facilitate periodic examination for damage or rust. Storage areas should be kept clean and free from such contaminants as concrete chipping, sanding, and painting.

Periodic rotation of equipment shafts should be performed in accordance with the equipment manufacturer's instructions for the specific equipment type and preservation methods used. When rotation is performed, determine first that all shipping blocks on rotating components have been removed and that there is adequate lubrication before rotation.

Certain preservatives and storage lubricants can affect safety and operating life of the equipment, especially if they react with the process fluid or operating lubricant. The

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installer should ensure that all preservative and storage lubricants are suitable for the specific application. Preservatives should not be used on surfaces where prohibited by the process or application.

Pump Location Working space must be checked to assure adequate accessibility for maintenance. Axially split casing horizontal pumps require sufficient headroom to lift the upper half of the casing free of the rotor. The inner assembly of radially split multistage centrifugal pumps is removed axially. Space must be provided so the assembly can be pulled out without canting it. For large pumps with heavy casings and rotors, a traveling crane or other facility for attaching a hoist should be provided over the pump location.

Pumps should be located as close as practicable to the source of liquid supply. Whenever possible, the pump centerline should be placed below the level of the liquid in the suction reservoir.

Foundations Foundations may consist of any structure heavy enough to afford permanent rigid support to the full area of the baseplate and to absorb any normal strains or shocks. Reinforced concrete foundations built up from solid ground are the most satisfactory. Although most pumping units are mounted on baseplates, very large equipment may be mounted directly on the foundations by using soleplates under the pump and driver feet. Misalignment is corrected with shims.

The space required by the pumping unit and the location of the foundation bolts are determined from the drawings supplied by the manufacturer. Each foundation bolt (Figure 1) should be surrounded by a pipe sleeve three or four diameters larger than the bolt. After the concrete foundations are poured, the pipe is held solidly in place but the bolt may be moved to conform to the corresponding hole in the baseplate.

When a unit is mounted on steelwork or some other structure, it should be placed directly over or as near as possible to the main members, beams, or walls and should be supported so the baseplate cannot be distorted or the alignment disturbed by any yielding or springing of the structure or of the baseplate.

Mounting of Vertical Wet-Pit Pumps A curb ring or soleplate must be used as a bearing surface for the support flange of a vertical wet-pit pump. The mounting face must be machined because the curb ring or soleplate will be used in aligning the pump.

If the discharge pipe is located below the support flange of the pump (belowground discharge), the curb ring or soleplate must be large enough to pass the discharge elbow during assembly. A rectangular ring should be used (Figure 2). If the discharge pipe is located above the support flange (aboveground discharge), a round curb ring or soleplate should be provided with clearance on its inner diameter to pass all sections of the pump below the support flange (Figure 3). A typical method of arranging a grouted soleplate for vertical pumps is shown in Figure 4.

If the discharge is belowground and an expansion joint is used, it is necessary to determine the moment that may be imposed on the structure. The pump casing should be attached securely to some rigid structural members with tie rods. If vertical wet-pit pumps are very long, some steadying device is required irrespective of the location of the discharge or of the type of pipe connection. Tie rods can be used to connect the unit to a wall, or a small clearance around a flange can be used to prevent excessive displacement of the pump in the horizontal plane.

Alignment When a complete unit is assembled at the factory, the baseplate is placed on a flat, even surface; the pump and driver are mounted on the baseplate; and the coupling halves are accurately aligned, using shims under the driver mounting surfaces where necessary. The pump is usually doweled to the baseplate at the factory, but the driver is left to be doweled after installation at the site.

The unit should be supported over the foundation by short strips of steel plate or shim stock close to the foundation bolts, allowing a space of $\frac{3}{4}$ to 2 in (2 to 5 cm) between the bottom of the baseplate and the top of the foundation for grouting. The shim stock should extend fully across the supporting edge of the baseplate. The coupling bolts should be

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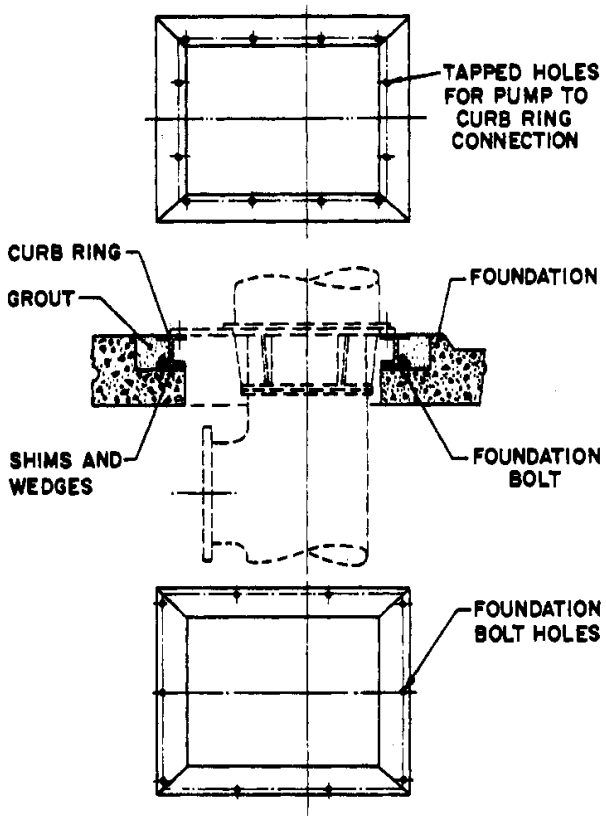


FIGURE 1 Foundation bolt

FIGURE 2 Rectangular curb ring for belowground discharge vertical pump

removed before the unit is leveled and the coupling halves are aligned. Where possible, it is preferable to place the level on some exposed part of the pump shaft, sleeve, or planed surface of the pump casing. The steel supporting strips or shim stock under the baseplate should be adjusted until the pump shaft is level, the suction and discharge flanges are vertical or horizontal as required, and the pump is at the specified height and location. When the baseplate has been leveled, the nuts on the foundation bolts should be made handtight.

During this leveling operation, accurate alignment of the unbolted coupling halves must be maintained. A straightedge should be placed across the top and sides of the coupling, and at the same time the faces of the coupling halves should be checked with a tapered thickness gage or with feeler gages (Figure 5) to see that they are parallel. For all alignment checks, including parallelism of coupling faces, both shafts should be pressed hard over to one side when taking readings.

When the peripheries of the coupling halves are true circles of equal diameter and the faces are flat and perpendicular to the shaft axes, exact alignment exists when the distance between the faces is the same at all points and when a straightedge lies squarely across the rims at any point. If the faces are not parallel, the thickness gage or feelers will show variation at different points. If one coupling is higher than the other, the amount may be determined by the straightedge and feeler gages.

Sometimes coupling halves are not true circles or are not of identical diameter because of manufacturing tolerances. To check the trueness of either coupling half, rotate it while holding the other coupling half stationary and check the alignment at each quarter turn of the half being rotated. Then the half previously held stationary should be revolved and the alignment checked. A variation within manufacturing limits may be found in either of the half-couplings, and proper allowance for this must be made when aligning the unit.

A more exact method for checking alignment that is recommended requires the use of a dial indicator. With the indicator bolted to the pump half of the coupling, both radial and

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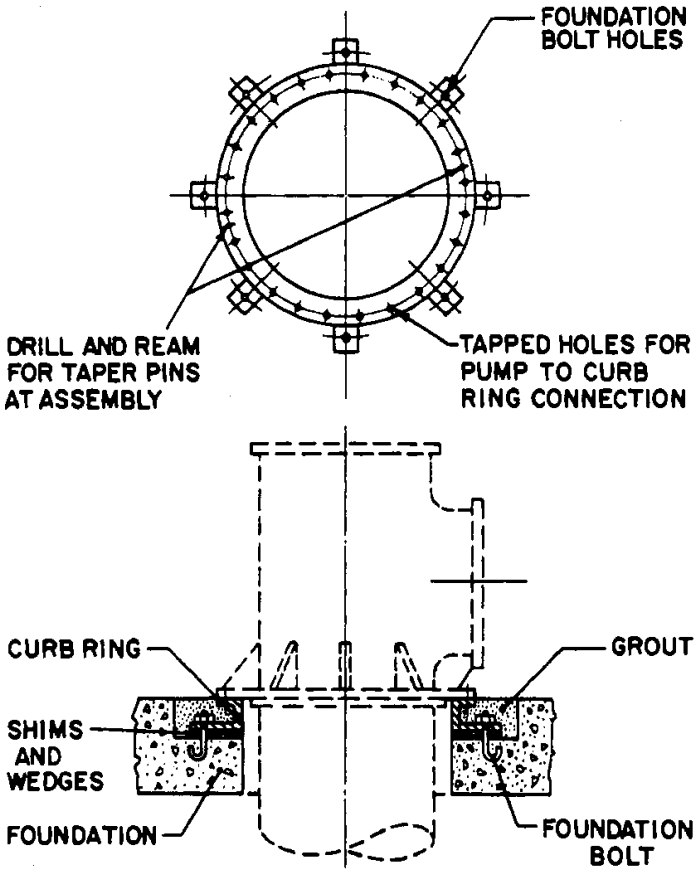


FIGURE 3 Round curb ring for aboveground discharge vertical pump

axial alignment can be checked, as illustrated in Figure 6. This method is called face-and-rim alignment. With the button resting on the periphery of the other coupling half, the dial should be set at zero and a mark chalked on the coupling half at the point where the button rests. For any check (top, bottom, or sides), both shafts should be rotated the same amount, that is, all readings on the dial should be made with the button on the chalk mark. The dial readings will indicate whether the driver must be raised, lowered, or moved to either side. After any movement, it is necessary to check that the coupling faces remain parallel to one another.

For example, if the dial reading at the starting point is set to zero and the diametrically opposite reading at the bottom or sides shows ±0.020 in (±0.508 mm), the driver must be raised or lowered by shimming or moved to one side or the other by half of this reading. The same procedure is used to align gear couplings, but the coupling covers must first be moved back out of the way and all measurements should be made on the coupling hubs.

When a spacer-type coupling connects the pump to its driver, a dial indicator should be used to check the alignment (Figure 7). The spacer between the coupling halves should be removed to expose the coupling hubs. The coupling nut on the end of the shaft may be used

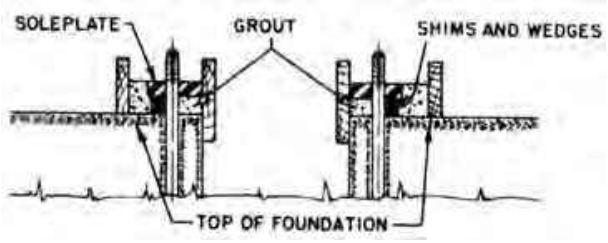


FIGURE 4 Grouting form for vertical pump soleplate (ANSI B-58.1 [AWWAE 101-61])

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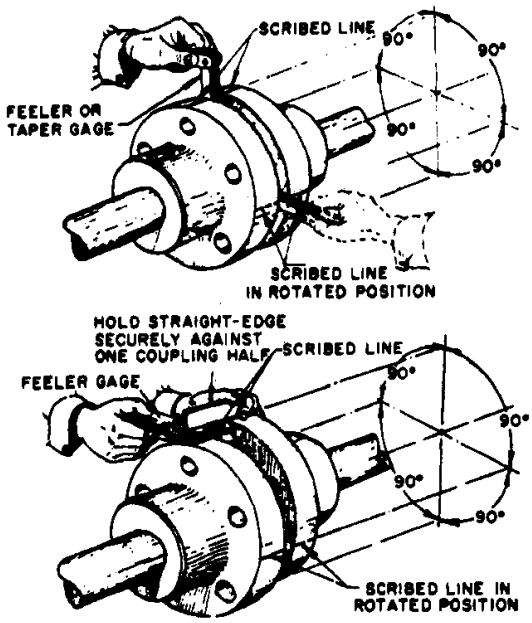


FIGURE 5 Coupling alignment using feeler gages

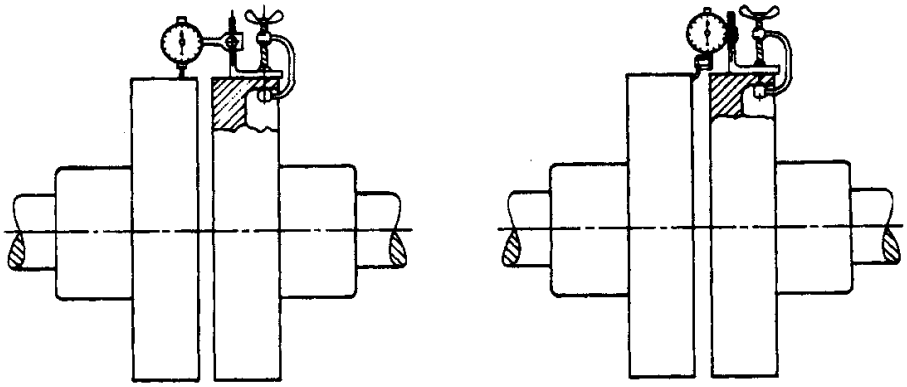


FIGURE 6 Use of dial indicator for face-and-rim alignment of standard coupling

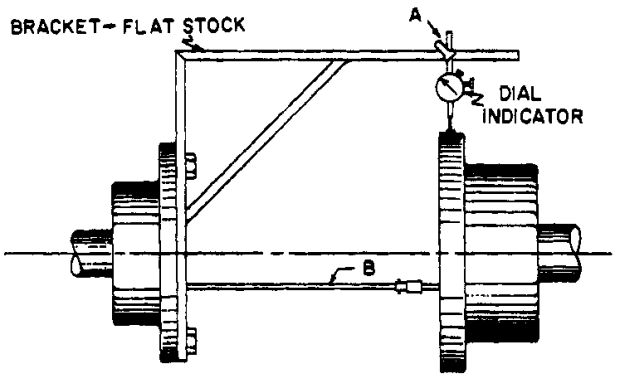


FIGURE 7 Use of dial indicator for face-and-rim alignment of spacer-type coupling

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to clamp a suitable extension arm or bracket long enough to extend across the space between the coupling hubs. The dial indicator is mounted on this arm, and alignment is checked for both concentricity of the hub diameters and parallelism of the hub faces. Changing the arm from one hub to the other provides an additional check. The dial extension bracket must be checked for sag, and readings must be corrected accordingly.

The clearance between the faces of the coupling hubs and the ends of the shafts should be such that they cannot touch, rub, or exert a pull on either pump or driver. The amount of this clearance may vary with the size and type of coupling used. Sufficient clearance will allow unhampered endwise movement of the shaft of the driving element to the limit of its bearing clearance. On motor-driven units, the magnetic center of the motor will determine the running position of the motor half-coupling. This position should be checked by running the motor uncoupled. This will also permit checking the direction of rotation of the motor. If current is not available at the time of installation, move the motor shaft in both directions as far as the bearings will permit and adjust the shaft centrally between these limits. The unit should then be assembled with the correct gap between the coupling halves.

Large horizontal sleeve-bearing motors are not generally equipped with thrust bearings. The motor rotor is permitted to float, and as it will seek its magnetic center, an axial force of rather small magnitude can cause it to move off this center. Sometimes it will move enough to cause the shaft collar to contact and possibly damage the bearing. To avoid this, a limited-end-float coupling is used between the pump and the motor on all large units to restrict the motor rotor (Section 9.3). The setting of axial clearances for such units should be given by the manufacturer in the instruction books and elevation drawings.

When the pump handles a liquid at other than ambient temperature or when it is driven by a steam turbine, the expansion of the pump or turbine at operating temperature will alter the vertical alignment. Alignment should be made at ambient temperature with suitable allowances for the changes in pump and driver centerlines after expansion. The final alignment must be made with the pump and driver at their normal temperatures and adjusted as required before the pump is placed into permanent service.

For large installations, particularly with steam-turbine-driven pumps, more sophisticated alignment methods are sometimes employed, using proximity probes and optical instruments. Such procedures permit checking the effect of temperature changes and machine strains caused by piping stresses while the unit is in operation. When such procedures are recommended, they are included with the manufacturer's instructions.

When the unit has been accurately leveled and aligned, the hold-down bolts should be gently and evenly tightened before grouting. The alignment must be rechecked after the suction and discharge piping has been bolted to the pump to test the effect of piping strains. This can be done by loosening the bolts and reading the movement of the pump, if any, with dial indicators.

The pump and driver alignment should be occasionally rechecked because misalignment may develop from piping strains after a unit has been operating for some time. This is especially true when the pump handles hot liquids because there may be a growth or change in the shape of the piping. Pipe flanges at the pump should be disconnected after a period of operation to check the effect of the expansion of the piping, and adjustments should be made to compensate for this.

For a further discussion of hot and cold alignment, face-and-rim versus reverse dial methods measurement of dial bracket sag, and graphical alignment plotting procedure, refer to Section 2.1.4.

Grouting Ordinarily, the baseplate is grouted before the piping connections are made and before the alignment of the coupling halves is finally rechecked. The purpose of grouting is to prevent lateral shifting of the baseplate, to increase the mass to reduce vibration, and to fill in irregularities in the foundation. Generally, it is recommended that all permanently installed pumping equipment be supported by a reinforced concrete foundation. Some pumps that require an elevated installation may be supported on structural steel structures, but care must be taken to ensure that such structures are of adequate stiffness and strength.

Foundation dimensions must consider the size and arrangement of the pump and driver, the piping arrangement and anticipated piping loads, anchor bolt placement, and minimum

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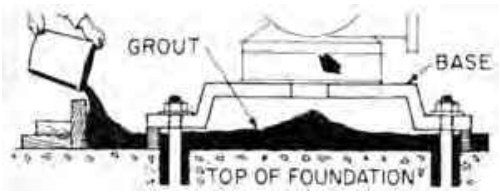


FIGURE 8 Application of grouting. Grout is added until the entire space under the base is filled. Holes in the base (arrow) allow air to escape and permit working of the grout to release air pockets.

dimensions required for servicing the equipment. Vertically suspended canned pump foundations should be designed so the pump can is directly attached to a mounting plate and is removable without disturbing the grout.

Foundation materials must be properly resistant to chemicals and oils. If the environment is aggressive, protective coatings or covers should be considered to protect the foundation and reinforcing steel. Two types of grout are commonly used: epoxy and cement-based. Epoxy grout, although usually more expensive, has significant advantages in higher bond strength and nonporous finished surface characteristics that generally make it the material of choice for most pump installations.

When the foundation has been prepared, the grout material has been selected, and the baseplate has been properly positioned on the foundation, a preliminary coupling alignment is made to ensure that final alignment is possible after the baseplate is finally grouted. After this alignment check is successfully accomplished, the grout is added through the holes in the baseplate. To retain the grout in place, a leak-tight form is built around the outside of the baseplate. Grout is added until the entire space under the baseplate is filled to the top of the underside (Figure 8). The grout holes and vent holes in the baseplate allow air to escape as the grout fills the cavity. A stiff wire may be used through the grout holes to work the grout and release any air pockets. It is usually best to start at one end and force the air out as the grout proceeds toward the other end of the baseplate. Leveling shims and wedges used to level the baseplate should be left in place after grouting.

When the grout has properly cured, voids have been filled and forms have been removed, the exposed surfaces of the grout and foundation can be properly finished. The foundation anchor/hold-down bolts should be finally torqued to the proper values and the coupling halves can be rechecked for alignment.

The pump and driver alignment must be rechecked thoroughly after the grout has hardened permanently, and at reasonable intervals thereafter.

Doweling of Pump and Driver When the pump handles hot liquids, doweling of both the pump and its driver should be delayed until the unit has been operated. A final recheck of alignment with the coupling bolts removed and with the pump and driver at operating temperature is advisable before doweling.

Large pumps handling hot liquids are usually doweled near the coupling end, allowing the pump to expand from that end out. Sometimes the other end is provided with a key and a keyway in the casing foot and the baseplate.

PIPING

Suction Piping The suction piping should be as direct and short as possible. If a long suction line is required, the pipe size should be increased to reduce frictional losses. (The exception to this recommendation is in the case of boiler-feed pumps, where difficulties may arise during transient conditions of load change if the suction piping volume is excessive. This is a special and complex subject, and the manufacturer should be consulted.) Where the pump must lift the liquid from a lower level, the suction piping should be laid out with a continual rise toward the pump, avoiding high spots in the line to prevent the formation of air pockets. Where a static suction head will exist, the pump suction piping should slope continuously downward to the pump.

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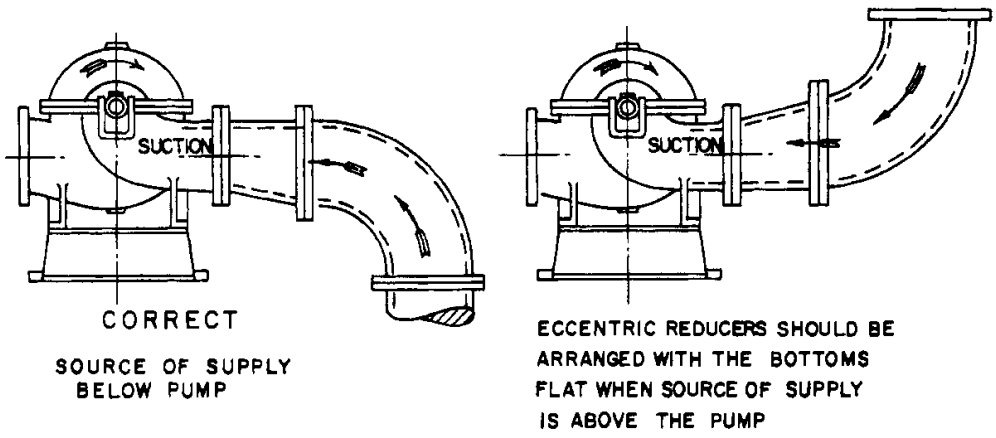


FIGURE 9 Recommended installation of reducers at pump suction

Generally, the suction line is larger than the pump suction nozzle and eccentric reducers should be used. If the source of supply is above the pump centerline, the reducer should be installed straight side up. If the source of supply is above the pump, the straight side of the reducer should be at the bottom (Figure 9). Installing eccentric reducers with a change in diameters greater than 4 in (10 cm) could disturb the suction flow. If such a change is necessary, it is advisable to use properly vented concentric reducers.

Elbows and other fittings next to the pump suction should be carefully arranged, or the flow into the pump impeller will be disturbed. Long-radius elbows are preferred for suction lines because they create less friction and provide a more uniform flow distribution than standard elbows.

It is extremely important to avoid the formation of vortices at the suction of both wet-pit and dry-pit pump installations. For a discussion of this and other suction conditions recommendations, see Sections 13.1 and 13.2.

Discharge Piping Generally both a check valve and a gate valve are installed in the discharge line. The check valve is placed between the pump and the gate valve and protects the pump from reverse flow in the event of unexpected driver failure or from reverse flow from another operating pump. The gate valve is used when priming the pump or when shutting it down for inspection and repairs. Manually operated valves that are difficult to reach should be fitted with a sprocket rim wheel and chain. In many cases, discharge gate valves are motorized and can be operated by remote control.

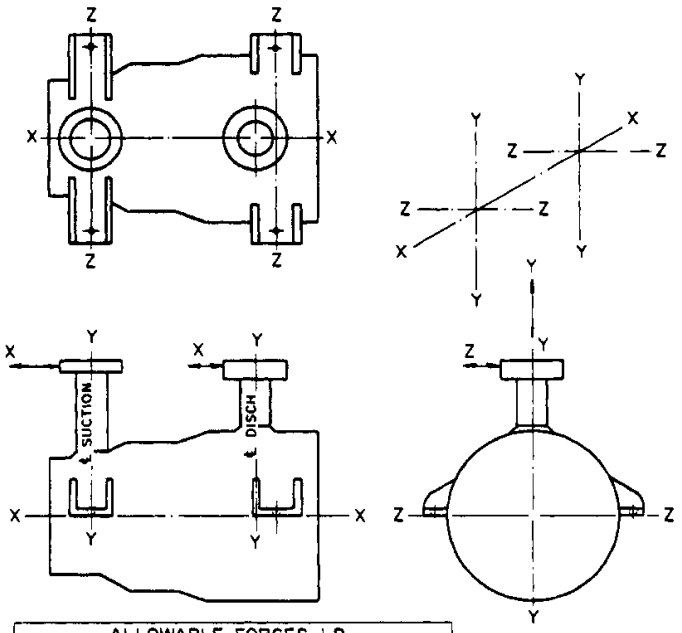
Piping Strains Cast iron pumps are never provided with raised face flanges. If steel suction or discharge piping is used, the pipe flanges should be of the flat-face and not the raised-face type. Full-face gaskets must be used with cast iron pumps.

Piping should not impose excessive forces and moments on the pump to which it is connected because these might spring the pump or pull it out of position. Piping flanges must be brought squarely together before the bolts are tightened. The suction and discharge piping and all associated valves, strainers, and so on should be supported and anchored near to but independent of the pump, so no strain will be transmitted to the pump casing.

There are four factors to be considered in determining the effect of nozzle loads: material stress in pump nozzles resulting from forces and bending moments, distortion of internal moving parts affecting critical clearances, stresses in pump hold-down bolts, and distortion in pump supports and baseplates resulting in driver coupling misalignment. API Standard 610 (Centrifugal Pumps for Refinery, Heavy Duty Chemical, and Gas Industry Services) provides guidelines for limiting the magnitude of nozzle loads and moments on pumps with suction nozzles 16 in (41 cm) and smaller and with casings constructed of steel or alloy steel.

With large pumps or when major temperature changes are expected, the pump manufacturer generally indicates to the user the maximum moments and forces that can be

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ALLOWABLE FORCES, LB			
	X	Y	Z
SUCTION	15,800	15,800	11,200
DISCHARGE	19,300	15,800	11,200
TOTAL	19,300	15,800	11,200

ALLOWABLE MOMENTS, FT-LB			
	X-X AXIS	Y-Y AXIS	Z-Z AXIS
SUCTION	22,900	45,400	22,900
DISCHARGE	21,600	43,200	21,600
TOTAL	22,900	45,400	22,900

FIGURE 10 Diagram of typical permissible pipe stresses and moments for a radially split double-casing multistage pump with top suction and discharge (1 N = 0.225 lb; 1 N · m = 0.737 ft · lb)

imposed on the pump by the piping. A typical diagram is illustrated in Figure 10 for a radially split double-casing multistage pump with top suction and discharge.

Expansion Joints Expansion joints are sometimes used in the discharge and suction piping to avoid transmitting any piping strains caused by misalignment or by expansion when hot liquids are handled. On occasion, expansion joints are formed by looping the pipe. More often, they are of the slip-joint or corrugated-diaphragm type. However, they transmit to the pump a force equal to the area of the expansion joint times the pressure in the pipe. These forces can be of very significant magnitude, and it is impractical to design the pump casings, baseplates, and so on to withstand them. Consequently, when expansion joints are used, a suitable pipe anchor must be installed between them and the pump proper. Alternately, tie rods can be used to prevent the forces from being transmitted to the pump.

Suction Strainers Except for certain special designs, pumps are not intended to handle liquid containing foreign matter. If the particles are sufficiently large, such foreign matter can clog the pump, reduce its capacity, or even render it altogether incapable of pumping. Small particles of foreign matter may cause damage by lodging between close running clearances. Therefore, proper suction strainers may be required in the suction lines of pumps not specially designed to handle foreign matter.

In such an installation, the piping must first be thoroughly cleaned and flushed. The recommended practice is to flush all piping to waste before connecting it to the pump. Then a temporary strainer of appropriate size should be installed in the suction line as close to the pump as possible. This temporary strainer may have a finer mesh than the

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permanent strainer installed after the piping has been thoroughly cleaned of all possible mill scale or other foreign matter. The size of the mesh is generally recommended by the pump manufacturer. For further details on strainers, see Sections 11.1 and 13.1.

Venting and Draining Vent valves are generally installed at one or more high points of the pump casing waterways to provide a means of escape for air or vapor trapped in the casing. These valves are used during the priming of the pump or during operation if the pump should become air- or vapor-bound. In most cases, these valves need not be piped up away from the pump because their use is infrequent, and the vented air or vapors can be allowed to escape into the surrounding atmosphere. On the other hand, vents from pumps handling flammable, toxic, or corrosive fluids must be connected in such a way that they endanger neither the operating personnel nor the installation. The suction vents of pumps taking liquids from closed vessels under vacuum must be piped to the source of the pump suction above the liquid level.

All drain and drip connections should be piped to a point where the leakage can be disposed of or collected for reuse if worth reclaiming.

Warm-Up Piping When it is necessary for a pump to come up to operating temperature before it is started up or to keep it ready to start at rated temperature, provision should be made for a warm-up flow to pass through the pump. There are a number of arrangements used to accomplish this. If the pump operates under positive pressure on the suction, the pumped liquid can be permitted to drain out through the pump casing drain connection to some point at a pressure lower than the suction pressure (Figure 11). Alternately, some liquid can be made to flow back from the discharge header through a jumper line around the check valve into the pump and out into the suction header (Figure 12). An orifice is provided in this jumper line to regulate the amount of warm-up flow. Care must be exercised in such an installation to maintain the suction valve open (unless the warm-up line valve is closed, as when the pump is to be dismantled) lest the entire pump, suction valve, and suction piping be subjected to full discharge pressure.

The manufacturer's recommendations should be sought in all cases as to the best means of providing an adequate warm-up procedure. Care must be taken to ensure that the pump is warmed uniformly. Stratification of the warm-up flow, or inadequate warm-up flow volume, can result in casing distortion or rotor bowing, or both (Heald and Penry, 1988).

RELIEF VALVES Positive displacement pumps, such as rotary and reciprocating pumps, can develop discharge pressures much in excess of their maximum design pressures. To protect these pumps against excessive pressures when the discharge is throttled or shut off, a pressure relief valve might be used. Some pumps are provided with internal integral relief valves, but unless operation against a closed discharge is both infrequent and of very

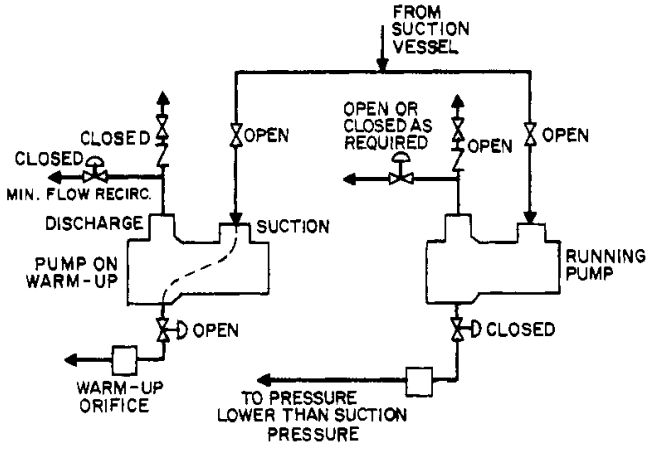


FIGURE 11 Arrangement for warm-up through pump casing drain connection

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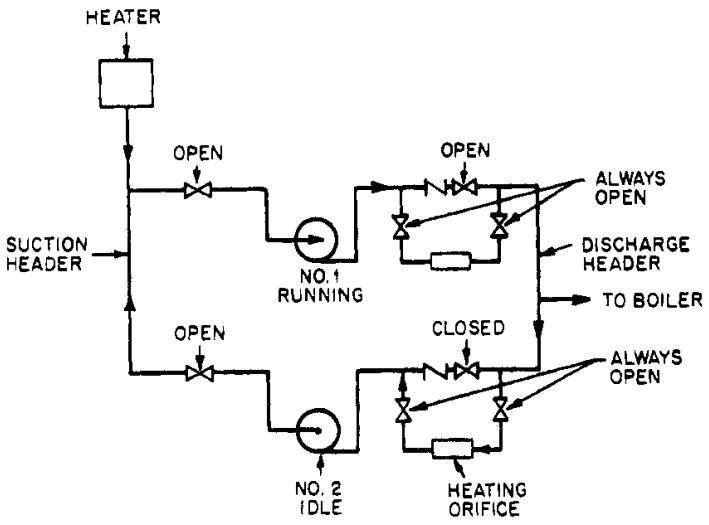


FIGURE 12 Arrangement for warm-up through jumper line around discharge check valve

short duration, a relief valve with an external return connection must be used and the liquid from the relief valve must be piped back to the source of supply.

Surge Chambers Generally, centrifugal pumps do not require surge chambers in their suction or discharge piping. Reciprocating pumps may have a suction and discharge piping layout that does not require compensation for variations in the flow velocity in the piping system. In many cases, however, reciprocating pump installations require surge chambers when the suction or discharge lines are of considerable length, when there is an appreciable static head on the discharge, when the liquid pumped is hot, or when it is desirable to smooth out variations in the discharge flow. The type, size, and arrangement of the surge chamber should be chosen on the basis of the manufacturer's recommendations. For more details, see Section 3.4.

Instrumentation There are a number of instruments that are essential to maintaining a close check on the performance and condition of a pump. A compound pressure gage should be connected to the suction of the pump, and a pressure gage should be connected to its discharge at the pressure taps that may be provided in the suction and discharge flanges. The gages should be mounted in a convenient location so they can be easily observed.

In addition, it is advisable to provide a flow-metering device. Depending upon the importance of the installation, indicating meters may be supplemented by recording attachments.

Whenever pumps incorporate various leakoff arrangements, such as a balancing device or pressure-reducing labyrinths, a check should be maintained on the quantity of these leakoffs by measuring orifices and differential gages installed in the leakoff lines.

Pumps operating in important or complex services or operating completely unattended by remote control may have additional instrumentation, such as speed indicators, vibration monitors, and bearing or casing temperature indicators. For more detail, see Sections 2.1.4 and 3.4.

OPERATION

Pumps are generally selected for a given capacity and total head when operating at rated speed. These characteristics are referred to as "rated conditions of service" and, with few exceptions, represent those conditions at or near which the pump will operate the greatest part of the time. Positive displacement pumps cannot operate at any greater flows than

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rated except by increasing their speed, nor can they operate at lower flows except by reducing their operating speed or bypassing some of the flow back to the source of supply. (See Section 3.5.)

On the other hand, centrifugal pumps can operate over a wide range of capacities, from near zero flow to well beyond the rated capacity. Because a centrifugal pump will always operate at the intersection of its head-capacity and system-head curves, the pump operating capacity may be altered either by throttling the pump discharge (hence altering the system-head curve) or by varying the pump speed (changing the pump head-capacity curve). This makes the centrifugal pump very flexible in a wide range of services and applications that require the pump to operate at capacities and heads differing considerably from the rated conditions. There are, however, some limitations imposed upon such operation by hydraulic, mechanical, or thermodynamic considerations (Section 2.1.3).

Operation of Centrifugal Pumps at Reduced Flows There are certain minimum operating flows that must be imposed on centrifugal pumps for either hydraulic or mechanical reasons. Four limiting factors must be considered: radial thrust, temperature rise, internal recirculation, and shape of the brake horsepower curve.

Radial thrust is discussed in Sections 2.1.3 and 2.2.1. For sustained operation, it is imperative to adhere to the minimum flow limits recommended by the pump manufacturer, which depend on the specific design of the pump casing and impeller.

The thermodynamic problem that arises when a centrifugal pump is operated at extremely reduced flows is caused by the heating up of the liquid handled. The difference between the brake horsepower consumed and the water horsepower developed represents the power losses in the pump, except for a small amount lost in the pump bearings. These power losses are converted to heat and transferred to the liquid passing through the pump.

If the pump were to operate against a completely closed valve, the power losses would be equal to the shutoff brake horsepower, and because there would be no flow through the pump, all this power would go into heating the small quantity of liquid contained in the pump casing. The pump casing would heat up, and a certain amount of heat would be dissipated by radiation and convection to the atmosphere. However, because the temperature rise in the liquid pumped could be quite rapid, it is generally safer to ignore the dissipation of heat through radiation and the absorption of heat by the casing. Calculations for determining the temperature rise in the pumped liquid are given in Section 2.1.3. The maximum permissible temperature rise in a centrifugal pump varies over a wide range, depending on the type of service and installation. For hot-water pumps, as on boiler-feed service, it is generally advisable to limit the temperature rise to about 15°F (8°C). As a general rule, the minimum permissible flow to hold the temperature rise in boiler-feed pumps to this value is 30 gpm for each 100 bhp (9.13 m^3/h per 100 kW) at shutoff. When the pump handles cold water, the temperature rise may be permitted to reach 50 or even 100°F (28 or 56°C). The minimum capacity based on thermodynamic considerations is then established as the capacity at which the temperature rise is the maximum permitted. Means and controls used to provide the necessary minimum flows are described in Section 10.

There are also hydraulic considerations that may affect the minimum flow at which a centrifugal pump can operate. In recent years, correlation has been developed between operation at low flows and the appearance of hydraulic pulsations both in the suction and in the discharge of centrifugal impellers. It has been proved that these pulsations are caused by the development of an internal recirculation at the inlet and discharge of an impeller at certain flows below the best-efficiency capacity. This subject is treated in Sections 2.1.3 and 2.1.4. The pump manufacturer's recommendations on minimum flows dictated by these considerations must be followed.

Priming With very few exceptions, no centrifugal pump should ever be started until it is fully primed; that is, until it has been filled with the liquid pumped and all the air contained in the pump has been allowed to escape. The exceptions involve self-priming pumps

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and some special large-capacity, low-head, and low-speed installations where it is not practical to prime the pump prior to starting; the priming takes place almost simultaneously with the starting in these cases. For further details, see Section 2.2.2.

Reciprocating pumps of the piston or plunger type are in principle self-priming. However, if quick starting is required, priming connections should be piped to a supply above the pump.

Positive displacement pumps of the rotating type, such as rotary or screw pumps, have clearances that allow the liquid in the pump to drain back to the suction. When pumping low-viscosity liquids, the pump may completely dry out when it is idle. In such cases a foot valve may be used to help keep the pump primed. Alternately, a vacuum device may be used to prime the pump. When handling liquids of higher viscosity, foot valves are usually not required because liquid is retained in the clearances and acts as a seal when the pump is restarted. However, before the initial start of a rotating positive displacement pump, some of the liquid to be pumped should be introduced through the discharge side of the pump to wet the rotating element.

The various methods and arrangements used for priming pumps are described in Section 2.2.2.

Final Checks Before Start-Up A few final checks are recommended before a pump is placed into service for its initial start. For pumps with journal bearings, the bearing covers should be removed, and the bearings should be flushed and thoroughly cleaned. They should then be filled with new lubricant in accordance with the manufacturer's recommendations.

With the coupling disconnected, the driver should be tested again for correct direction of rotation. Generally an arrow on the pump casing indicates the correct rotation.

It must be possible to rotate the rotor of a centrifugal pump by hand, and in the case of a pump handling hot liquids, the rotor must be free to rotate with the pump cold or hot. If the rotor is bound or even drags slightly, do not operate the pump until the cause of the trouble is determined or corrected.

Starting and Stopping Procedures The steps necessary to start a centrifugal pump depend upon its type and upon the service on which it is installed. For example, standby pumps are generally held ready for immediate starting. The suction and discharge gate valves are held open, and reverse flow through the pump is prevented by the check valve in the discharge line.

The methods followed in starting are greatly influenced by the shape of the power-capacity curve of the pump. High- and medium-head pumps (low and medium specific speeds) have power curves that rise from zero flow to the normal capacity condition. Such pumps should be started against a closed discharge valve to reduce the starting load on the driver. A check valve is equivalent to a closed valve for this purpose, as long as another pump is already on the line. The check valve will not lift until the pump being started comes up to a speed sufficient to generate a head high enough to lift the check valve from its seat. If a pump is started with a closed discharge valve, the recirculation bypass line must be open to prevent overheating.

Low-head pumps (high specific speed) of the mixed-flow and propeller type have power curves that rise sharply with a reduction in capacity; they should be started with the discharge valve wide open against a check valve, if required, to prevent backflow.

Assuming that the pump in question is motor-driven, that its shutoff power does not exceed the safe motor power, and that it is to be started against a closed gate valve, the starting procedure is as follows:

1. Prime the pump, opening the suction valve, closing the drains, and so on, to prepare the pump for operation.
2. Open the valve in the cooling supply to the bearings, where applicable.
3. Open the valve in the cooling supply if the seal chambers are liquid-cooled.
4. Open the valve in the sealing liquid supply if the pump is so fitted.

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5. Open the warm-up valve of a pump handling hot liquids if the pump is not normally kept at operating temperature. When the pump is warmed up, close the valve.
6. Open the valve in the recirculating line if the pump should not be operated against dead shutoff.
7. Start the motor.
8. Open the discharge valve slowly.
9. For pumps equipped with mechanical seals, check for seal leakage: there should be none.
10. For pump with shelf packing, observe the leakage from the stuffing boxes and adjust the sealing liquid valve for proper flow to ensure the lubrication of the packing. If the packing is new, do not tighten up on the gland immediately, but let the packing run in before reducing the leakage through the stuffing boxes.
11. Check the general mechanical operation of the pump and motor.
12. Close the valve in the recirculating line when there is sufficient flow through the pump to prevent overheating.

If the pump is to be started against a closed check valve with the discharge gate valve open, the steps are the same, except that the discharge gate valve is opened some time before the motor is started.

In certain cases, cooling to the bearings and flush liquid to the mechanical seals or to the packing seal cages is provided by the pump. This, of course, eliminates the need for the steps listed for the cooling and sealing supply.

Just as in starting a pump, the stopping procedure depends upon the type and service of the pump. Generally, the steps followed to stop a pump that can operate against a closed gate valve are

1. Open the valve in the recirculating line.
2. Close the gate valve.
3. Stop the motor.
4. Open the warm-up valve if the pump is to be kept at operating temperature.
5. Close the valve in the cooling supply to the bearings and seal chambers.
6. If the sealing liquid supply is not required while the pump is idle, close the valve in this supply line.
7. Close the suction valve, open the drain valves, and so on, as required by the particular installation or if the pump is to be opened up for inspection.

If the pump is of a type that does not permit operation against a closed gate valve, steps 2 and 3 are reversed.

In general, the starting and stopping of steam-turbine-driven pumps require the same steps and sequence prescribed for a motor-driven pump. As a rule, steam turbines have various drains and seals that must be opened or closed before and after operation. Similarly, many turbines require warming up before starting. Finally, some turbines require turning gear operation if they are kept on the line ready to start up. The operator should therefore follow the steps outlined by the turbine manufacturer in starting and stopping the turbine.

Most of the steps listed for starting and stopping centrifugal pumps are equally applicable to positive displacement pumps. There are, however, two notable exceptions:

1. Never operate a positive displacement pump against a closed discharge. If the gate valve on the discharge must be closed, always start the pump with the recirculation bypass valve open.
2. Always open the steam cylinder drain cocks of a steam reciprocating pump before starting, to allow condensate to escape and to prevent damage to the cylinder heads.

Auxiliary Services on Standby Pumps Standby pumps are frequently started up from a remote location, and several methods of operation are available for the auxiliary services, such as the cooling supply to bearings or seal chambers:

1. A constant flow may be kept through jackets or oil coolers and through stuffing box lantern rings or seal chambers whether the pump is running or on standby service.
2. The service connections may be opened automatically whenever the pump is started up.
3. The service connections may be kept closed while the pump is idle, and the operator may be instructed to open them shortly after the pump has been put on the line automatically.

The choice among these methods must be dictated by the specific circumstances surrounding each case. There are, however, certain cases where sealing liquid supply to the pump seal chambers must be maintained whether the pump is running or not. This is the case when the pump handles a liquid that is corrosive or that may crystallize and deposit on the sealing components. It is also the case when the sealing supply is used to prevent air infiltration into a pump when it is operating under a vacuum.

Restarting Motor-Driven Pumps After Power Failure Assuming that power failure will not cause the pump to go into reverse rotation; that is, that a check valve will protect the pump against reverse flow, there is generally no reason why the pump should not be permitted to restart after current has been re-established. Whether the pump will start again automatically when power is restored will depend on the type of motor control used. (Section 2.1.3 and Section 11.1 give reasons why some pumps should not be started in reverse.)

Because pumps operating on a suction lift may lose their prime during the time that power is off, it is preferable to use starters with low load protection for such installations to prevent an automatic restart. This does not apply, of course, if the pumps are automatically primed or if some protection device is incorporated so the pump cannot run unless it is primed.

MAINTENANCE

Because of the wide variation in pump types, sizes, designs, and materials of construction, these comments on maintenance are restricted to those types of pumps most commonly encountered. The manufacturer's instruction books must be carefully studied before any attempt is made to service a particular pump.

Daily Observation of Pump Operation When operators are on constant duty, hourly and daily inspections should be made and any irregularities in the operation of a pump should be recorded and reported immediately. This applies particularly to changes in sound of a running pump, abrupt changes in bearing temperatures, and seal chamber leakage. A check of pressure gages and of flowmeters, if installed, and vibration should be made routinely during the day. If recording instruments are provided, a daily check should be made to determine whether the current capacity, pressure, power consumption or vibration level indicates that further inspection is required. If these readings are taken electronically, trending charts should be produced to allow observation of changes as a function of time. Certain trends may allow for scheduled outages to address deterioration of specific performance values.

Semiannual Inspection The following should be done at least every 6 months:

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1. For pumps equipped with shaft packing, the free movement of stuffing box glands should be checked, gland bolts should be cleaned and lubricated, and the packing should be inspected to determine whether it requires replacement.
2. The pump and driver alignment should be checked and corrected if necessary.
3. Housings for oil-lubricated bearings should be drained, flushed, and refilled with fresh oil.
4. Grease-lubricated bearings should be checked to see that they contain the correct amount of grease and that it is still of suitable consistency.

Annual Inspection A very thorough inspection should be performed once a year. In addition to the semiannual procedure, the following items should be considered:

1. Vibration trends should be reviewed. If the pump is trending toward unacceptable vibration levels,
   a. The bearings should be removed, cleaned, and examined for flaws and wear.
   b. The bearing housings should be carefully cleaned.
   c. Rolling element bearings should be examined for scratches and wear.
   d. Immediately after cleaning, rolling element bearings that are considered acceptable for reinstallation should be coated with oil or grease. *Note:* If there is any sign of damage, or if the bearings were damaged during removal, they should be replaced with new bearings of the correct size and type per the manufacturer's instruction book.
   e. The assembled rotor—or major rotor components if the rotor is not assembled of shrink-fit components—should be checked for balance prior to reassembly in the pump.
2. For pumps equipped with shaft packing, the packing should be removed and the shaft sleeves—or shaft, if no sleeves are used—should be examined for wear.
3. For pumps equipped with mechanical seals, if the seals were indicating signs of leaking, they should be removed and returned to the seal manufacturer for inspection, possible bench testing, and refurbishment.
4. When coupling halves are disconnected for an alignment check, the vertical shaft movement of a pump with sleeve (journal) bearings should be checked at both ends with packing or seals removed. Any movement exceeding 150% of the original design clearance should be investigated to determine the cause. Endplay allowed by the bearings should also be checked. If it exceeds that recommended by the manufacturer, the cause should be determined and corrected.
5. All auxiliary piping, such as drains, sealing water piping, and cooling water piping, should be checked and flushed, as necessary. Auxiliary coolers should also be flushed and cleaned.
6. Pump equipped with stuffing boxes should be repacked, and the pump and driver should be realigned and reconnected.
7. All instruments and flow-metering devices should be recalibrated, whenever feasible, and—whenever possible—the pump should be tested to determine whether proper performance is being obtained. If internal repairs are made, the pump should again be tested after completion of the repairs.

Complete Overhaul It is difficult to make general rules about the frequency of complete pump overhauls as it depends on the pump service, the pump construction and materials, the liquid handled, and the economic evaluation of overhaul costs versus the cost of power losses resulting from increased clearances or of unscheduled downtime. Some pumps on very severe service may need a complete overhaul monthly, whereas other applications require overhauls only every 2 to 4 years or even less frequently.

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A pump should not be opened for inspection unless either factual or circumstantial evidence indicates that overhaul is necessary. Factual evidence implies that the pump performance has fallen off significantly or that the noise or driver load indicates trouble. Circumstantial evidence refers to past experience with the pump in question or with similar equipment on similar service.

In order to ensure rapid restoration to service in the event of an unexpected overhaul, an adequate store of spare parts should be maintained at all times.

The relative complexity of the repairs, the facilities available at the site, and many other factors enter into the decision whether the necessary repairs will be carried out at the installation site or at the pump manufacturer's plant.

Spare and Repair Parts The severity of the service in which the pump is used will determine, to a great extent, the minimum number of spare parts that should be carried in stock at the installation site. Unless prior experience is available, the pump manufacturer should be consulted on this subject. As an insurance against delays, spare parts should be purchased when the pump is purchased. Depending on the contemplated method of overhaul, certain replacement parts may have to be supplied either oversized or undersized instead of the size used in the original unit.

API Standard 610, 8th edition, shows recommended spare parts as a function of the number of identical pumps installed at a site. It also lists parts usually associated with start-up and with normal maintenance. A form based on the recommendations of API Standard 610 is shown in Table 1.

When ordering spare parts after a pump has been in service, the manufacturer should always be given the pump serial number and size (stamped on the nameplate). This information is essential in identifying the pump exactly and in furnishing repair parts of correct size and material.

Records of Inspections and Repairs The working schedule of the semiannual and annual inspections should be entered into a log that tracks individual pump maintenance history. This log should then contain a complete record of all the items requiring attention. This log, usually electronic, should also contain comments and observations on the conditions of the parts to be repaired or replaced, on the rate and appearance of wear, and on the repair methods followed. In many cases, it is advisable to photograph badly worn parts before they are repaired.

In all cases, complete records of the cost of maintenance and repairs should be kept for each pump, together with a record of its operating hours. A study of these records will generally reveal whether a change in materials or even construction may be the most economical course of action to improve pump performance, reliability, and life.

Diagnosis of Pump Problems Pump operating problems may be either hydraulic or mechanical. In the first category, a pump may fail to deliver liquid, it may deliver an insufficient capacity or develop insufficient pressure, or it may lose its prime after starting. In the second category, it may consume excessive power, or symptoms of mechanical difficulties may develop at the seal chambers or at the bearings, or vibration, noise, or breakage of some pump parts may occur.

There is a definite interdependence between some difficulties of both categories. For example, increased wear at the running clearances must be classified as a mechanical trouble, but it will result in a reduction of the net pump capacity—a hydraulic symptom—without necessarily causing a mechanical breakdown or even excessive vibration. As a result, it is most useful to classify symptoms and causes separately and to list for each symptom a schedule of potential contributory causes. Such a diagnostic analysis is presented in Tables 2 through 5. Additionally, the following parts of this handbook should also be referred to for assistance in diagnosing pump hydraulic and mechanical problems and possible solutions: Sections 2.1.4, 3.4, 5.1, 5.2, and Chap. 10.

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TABLE 1 Recommended spare parts

Part	See Note	Spares Recommended: Start-up			Normal Maintenance			
Number of identical pumps		1–3	4–6	7+	1–3	4–6	7–9	10+
Cartridge	(2) (5)				1	1	1	1
Element	(2) (6)				1	1	1	1
Rotor	(3) (7)				1	1	1	1
Case	(1)							1
Head (case cover and seal chamber)								1
Bearing bracket	(1)							1
Shaft (w/key)					1	1	2	N/3
Impeller					1	1	2	N/3
Wear rings (set)	(8)	1	1	1	1	1	2	N/3
Bearings complete (rolling element, radial)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearings complete (rolling element, thrust)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearings complete (hydrodynamic, radial)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearing pads only (hydrodynamic, radial)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearing complete (hydrodynamic, thrust)	(1) (9)	1	1	2	1	2	N/3	N/3
Bearing pads only (hydrodynamic, thrust)	(1) (9)	1	1	2	1	2	N/3	N/3
Mechanical seal/packing	(4) (8) (9)	1	2	N/3	1	2	N/3	N/3
Shaft sleeve	(8)	1	2	N/3	1	2	N/3	N/3
Gaskets, shims, O-rings (set)	(8)	1	2	N/3	1	2	N/3	N/3
Add for vertical pump								
Bowls							N/3	N/3
Spiders (set)				1	1	1	N/3	N/3
Bearings, bushings (set)		1	1	2	1	1	N/3	N/3
Add for high speed integral gear								
Gear box			1	1	1	1	1	N/3
Diffuser and cover		1	1	1	1	1	1	N/3
Splined shaft		1	1	1	1	1	1	N/3
Gear box housing					1	1	1	N/3
Oil pump, internal			1	1	1	1	1	N/3
Oil pump, external			1	1	1	1	1	N/3
Oil filter		1	2	N/3	1	2	3	N/3

Notes:
N = Number of identical pumps
(1) Horizontal pumps only
(2) Vital service pumps are generally unspared, partially spared, or multistage. When a vital machine is down, production loss or violation of environmental permits results.
(3) Essential service pumps are required for operation and have an installed spare. A production loss will occur only if main and spare fail simultaneously.
(4) Cartridge type mechanical seals shall include sleeve and gland.
(5) Cartridge consists of assembled element plus discharge head, seal(s), and bearing housing(s).
(6) Element consists of assembled rotor plus stationary hydraulic parts (diffuser(s) or volute(s)).
(7) Rotor consists of all rotating parts attached to the shaft.
(8) Normal wear parts
(9) Per pump set

Source: (American Petroleum Institute, 2004)

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TABLE 2A Check chart for centrifugal pump problems

Symptoms	Possible cause of trouble (each number is defined in Table 2B)
1. Pump does not deliver liquid	1, 2, 3, 5, 10, 12, 13, 14, 16, 21, 22, 25, 30, 32, 38, 40
2. Insufficient capacity delivered	2, 3, 4, 5, 6, 7, 7a, 10, 11, 12, 13, 14, 15, 16, 17, 18, 21, 22, 23, 24, 25, 31, 32, 40, 41, 44, 63, 64
3. Insufficient pressure developed	4, 6, 7, 7a, 10, 11, 12, 13, 14, 15, 16, 18, 21, 22, 23, 24, 25, 34, 39, 40, 41, 63, 64
4. Pump loses prime after starting	2, 4, 6, 7, 7a, 8, 9, 10, 11
5. Pump requires excessive power	20, 22, 23, 24, 26, 32, 33, 34, 35, 39, 40, 41, 44, 45, 61, 69, 70, 71
6. Pump vibrates or is noisy at all flows	2, 16, 37, 43, 44, 45, 46, 47, 48, 49, 50, 51, 52, 53, 54, 55, 56, 57, 58, 59, 60, 61, 67, 78, 79, 80, 81, 82, 83, 84, 85
7. Pump vibrates or is noisy at low flows	2, 3, 17, 19, 27, 28, 29, 35, 38, 77
8. Pump vibrates or is noisy at high flows	2, 3, 10, 11, 12, 13, 14, 15, 16, 17, 18, 33, 34, 41
9. Shaft oscillates axially	17, 18, 19, 27, 29, 35, 38
10. Impeller vanes are eroded on visible side	3, 12, 13, 14, 15, 17, 41
11. Impeller vanes are eroded on invisible side	12, 17, 19, 29
12. Impeller vanes are eroded at discharge near center	37
13. Impeller vanes are eroded at discharge near shrouds or at shroud/vane fillets	27, 29
14. Impeller shrouds bowed out or fractured	27, 29
15. Pump overheats and seizes	1, 3, 12, 28, 29, 38, 42, 43, 45, 50, 51, 52, 53, 54, 55, 57, 58, 59, 60, 61, 62, 77, 78, 82
16. Internal parts are corroded prematurely	66
17. Internal clearances wear too rapidly	3, 28, 29, 45, 50, 51, 52, 53, 54, 55, 57, 59, 61, 62, 66, 77
18. Axially-split casing is cut through wire-drawing	63, 64, 65
19. Internal stationary joints are cut through wire-drawing	53, 63, 64, 65
20. Packed box leaks excessively or packing has short life	8, 9, 45, 55, 57, 68, 69, 70, 71, 72, 73, 74
21. Packed box: sleeve scored	8, 9
22. Mechanical seal leaks excessively	45, 54, 55, 57, 58, 62, 75, 76
23. Mechanical seal: damaged faces, sleeve, bellows	45, 54, 55, 57, 58, 62, 75, 76
24. Bearings have short life	3, 29, 41, 42, 45, 50, 51, 54, 55, 58, 77, 78, 79, 80, 81, 82, 83, 84, 85
25. Coupling fails	45, 50, 51, 54, 67

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TABLE 2B Possible causes of problems

Suction Problems

1. Pump not primed
2. Pump suction pipe not completely filled with liquid
3. Insufficient available NPSH
4. Excessive amount of air or gas in liquid
5. Air pocket in suction line
6. Air leaks into suction line
7. Air leaks into pump through stuffing boxes or through mechanical seal

7a. Air in source of sealing liquid

8. Water seal pipe plugged
9. Seal cage improperly mounted in stuffing box
10. Inlet of suction pipe insufficiently submerged
11. Vortex formation at suction
12. Pump operated with closed or partially closed suction valve
13. Clogged suction strainer
14. Obstruction in suction line
15. Excessive friction losses in suction line
16. Clogged impeller
17. Suction elbow in plane parallel to the shaft (for double-suction pumps)
18. Two elbows in suction piping at 90° to each other, creating swirl and prerotation
19. Selection of pump with too high a suction specific speed

Other Hydraulic Problems

20. Speed of pump too high
21. Speed of pump too low
22. Wrong direction of rotation
23. Reverse mounting of double-suction impeller
24. Uncalibrated instruments
25. Impeller diameter smaller than specified
26. Impeller diameter larger than specified
27. Impeller selection with abnormally high head coefficient
28. Running the pump against a closed discharge valve without opening a by-pass
29. Operating pump below recommended minimum flow
30. Static head higher than shut-off head

Other Hydraulic Problems (continued)

31. Friction losses in discharge higher than calculated
32. Total head of system higher than design of pump
33. Total head of system lower than design of pump
34. Running pump at too high a flow (for low specific speed pumps)
35. Running pump at too low a flow (for high specific speed pumps)
36. Leak of stuck check valve
37. Too close a gap between impeller vanes and volute tongue or diffuser vanes
38. Parallel operation of pumps unsuitable for the purpose
39. Specific gravity of liquid differs from design conditions
40. Viscosity of liquid differs from design conditions
41. Excessive wear at internal running clearances
42. Obstruction in balancing device leak-off line
43. Transients at suction source (imbalance between pressure at surface of liquid and vapor pressure at suction flange)

Mechanical Problems—general

44. Foreign matter in impellers
45. Misalignment
46. Foundation insufficiently rigid
47. Loose foundation bolts
48. Loose pump or motor bolts
49. Inadequate grouting of baseplate
50. Excessive piping forces and moments on pump nozzles
51. Improperly mounted expansion joints
52. Starting the pump without proper warm-up
53. Mounting surfaces of internal fits (at wearing rings, impellers, shaft sleeves, shaft nuts, bearing housings, and so on) not perpendicular to shaft axis
54. Bent shaft
55. Rotor out of balance
56. Parts loose on the shaft
57. Shaft running off-center because of worn bearings

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TABLE 2B Possible causes of problems (*Continued*)

Mechanical Problems—general (continued)

58. Pump running at or near critical speed
59. Too long a shaft span or too small a shaft diameter
60. Resonance between operating speed and natural frequency of foundation, baseplate, or piping
61. Rotating part rubbing on stationary part
62. Incursion of hard solid particles into running clearances
63. Improper casing gasket material
64. Inadequate installation of gasket
65. Inadequate tightening of casing bolts
66. Pump materials not suitable for liquid handled
67. Certain couplings lack lubrication

Mechanical Problems—sealing area

68. Shaft or shaft sleeves worn or scored at packing
69. Incorrect type of packing for operating conditions
70. Packing improperly installed
71. Gland too tight, prevents flow of liquid to lubricate packing
72. Excessive clearance at bottom of stuffing box allows packing to be forced into pump interior

Mechanical Problems—sealing area (continued)

73. Dirt or grit in sealing liquid
74. Failure to provide adequate cooling liquid to water-cooled stuffing boxes
75. Incorrect type of mechanical seal for prevailing conditions
76. Mechanical seal improperly installed

Mechanical Problems—bearings

77. Excessive radial thrust in single-volute pumps
78. Excessive axial thrust caused by excessive wear at internal clearances or, if used, failure or excessive wear of balancing drive
79. Wrong grade of grease or oil
80. Excessive grease or oil in rolling element bearing housings
81. Lack of lubrication
82. Improper installation of rolling element bearings such as damage during installation, incorrect assembly of stacked bearings, use of unmatched bearings as a pair, and so on
83. Dirt getting into bearings
84. Moisture contaminating lubricant
85. Excessive cooling of water-cooled bearings

TABLE 2C Diagnosis from appearance of stuffing box packing in centrifugal pumps

Symptom	Cause
Wear on one or two rings next to packing gland; other rings OK	Improper packing installation
Wear on O.D. of packing rings	Packing rings rotating with shaft sleeve or leakage between rings and I.D. of box. Wrong packing size or incorrectly cut rings
Charring or glazing of inner circumference of rings	Excessive heating. Insufficient leakage to lubricate packing or unsuitable packing
I.D. of rings excessively increased or heavily worn on part of inner circumference	Rotation eccentric

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TABLE 2D Vibration symptoms and causes in centrifugal pumps

Vibration frequency	Cause
Several times pump RPM	Bad rolling element bearings
Twice pump RPM	Loose parts on rotor, axial misalignment of coupling, influence of twin-volute when gap is insufficient
Running Speed	Imbalance of rotor, clogged impeller, coupling misaligned or loose
Running speed times number of impeller vanes	Vane passing syndrome—insufficient gap between impeller vanes and collector vanes. This is also sometimes seen during operating with suction recirculation.
One-half running speed	Oil whirl in bearing
Random low frequency	Internal circulation in impeller or cavitation
Random high frequency	Usually resonance
Subsynchronous frequency at 70% to 90% of running speed	Hydraulic excitation of resonance

TABLE 3 Check chart for rotary pump problems

Symptom	Possible cause of trouble (each number is defined in the list below)
Pump fails to discharge	1, 2, 3, 4, 5, 6, 8, 9, 16
Pump is noisy	6, 10, 11, 17, 18, 19
Pump wears rapidly	11, 12, 13, 20, 24
Pump not up to capacity	3, 5, 6, 7, 9, 16, 21, 22
Pump starts, then loses suction	1, 2, 6, 7, 10
Pump takes excessive power	14, 15, 17, 20, 23

Suction problems

1. Pump not properly primed
2. Suction pipe not submerged
3. Strainer clogged
4. Foot valve leaking
5. Suction lift too high
6. Air leaking into suction
7. Suction pipe too small

System problems

8. Wrong direction of rotation
9. Low speed
10. Insufficient liquid supply
11. Excessive pressure
12. Grit or dirt in liquid

System problems (continued)

13. Pump runs dry
14. Viscosity higher than specified
15. Obstruction in discharge line

Mechanical troubles

16. Pump worn
17. Drive shaft bent
18. Coupling out of balance or alignment
19. Relief valve chatter
20. Pipe strain on pump casing
21. Air leak at packing or seal
22. Relief valve improperly seated
23. Packing too tight
24. Corrosion

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TABLE 4 Check chart for reciprocating pump problems

Symptom	Possible cause of problem (each number is defined in the list below)
Liquid end noise	1, 2, 7, 8, 9, 10, 14, 15, 16
Power end noise	17, 18, 19, 20
Overheated power end	10, 19, 21, 22, 23, 24
Water in crankcase	25
Oil leak from crankcase	26, 27
Rapid packing or plunger wear	11, 12, 28, 29
Pitted valves or seats	3, 11, 30
Valves hanging up	31, 32
Leak at cylinder valve hole plugs	10, 13, 33, 34
Loss of prime	1, 4, 5, 6

Suction problems

1. Insufficient suction pressure
2. Partial loss of prime
3. Cavitation
4. Lift too high
5. Leaking suction at foot valve
6. Acceleration head requirement too high

System problems

7. System shocks
8. Poorly supported piping, abrupt turns in piping, pipe too small, piping misaligned
9. Air in liquid
10. Overpressure or overspeed
11. Dirty liquid
12. Dirty environment
13. Water hammer

Mechanical problems

14. Valves broken or badly worn
15. Packing worn
16. Obstruction under valve
17. Main bearings loose
18. Bearings worn
19. Oil level low
20. Plunger loose
21. Main bearings tight
22. Ventilation inadequate
23. Belts too tight
24. Driver misaligned
25. Condensation
26. Seals worn
27. Oil level too high
28. Pump not level and rigid
29. Packing loose
30. Corrosion
31. Valve binding
32. Valve spring broken
33. Cylinder plug loose
34. O-ring seal damaged

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TABLE 5 Check chart for steam-pump problems

Symptoms	Possible cause of problem (each number is defined in the list below)
Pump does not develop rated pressure	4, 5, 7, 8
Pump loses capacity after starting	1, 2, 6
Pump vibrates	9, 10, 11, 14
Pump has short strokes	12, 13, 14
Pump operation is erratic	1, 2, 3, 6

Suction problems
1. Suction line leaks
2. Suction lift too high
3. Cavitation

System problems
4. Low steam pressure
5. High exhaust pressure
6. Entrained air or vapors in liquid

Mechanical problems
7. Worn piston rings in steam end
8. Binding piston rings in liquid end
9. Misalignment
10. Foundation not rigid
11. Piping not supported
12. Excessive steam cushioning
13. Steam valves out of adjustment
14. Liquid piston packing too tight

FURTHER READING

American Petroleum Institute. "Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries", API Standard 610, 10th ed., 1220 L Street, Washington, DC, 20005, 2004, www.api.org.

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C • H • A • P • T • E • R • 16

PUMP TESTING

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Since the first time a device was used to pump or lift water, pump testing of one sort or another has occurred. Each improvement in pumping devices was accepted only after being tested, which was the proof of its worthiness. As pumping equipment has become more refined, so has the art of pump testing, both in the shop or laboratory and in the field. For very large pumps, model testing is being used to develop the optimum refinement in prototype design.

Every pump, regardless of size or classification, should be tested in some way before final acceptance by the purchaser. If not, the user does not have any way of knowing that all requirements have been fulfilled. What tests to run and what methods to use depend on the ultimate purpose of the tests, which normally have one of two objectives:

1. To check improvement in design or operation
2. To determine if contractual commitments have been met, thus making possible the comparison of specified, predicted, and actual performance

In most cases, the manufacturer supplies a test report and certifies the characteristics of the pump being furnished. Even these can be given a cursory check by the customer from time to time to give a record of performance or an indication of the need for replacement or overhaul. If at all possible, the pump should be tested as installed, with repeat tests from time to time to check operation.

The main object of this chapter is to present a set of procedures and rules for conducting, computing, and reporting on tests of pumping units and for obtaining the head, capacity, power, efficiency, and suction requirements of a pump.

CLASSIFICATION OF TESTS

Pump tests should be classified as follows:

- *Shop tests* are also called laboratory, manufacturer's, or factory acceptance tests. They are conducted in the pump manufacturer's plant under geometrically similar, ideal, and controlled conditions and are usually assumed to be the most accurate tests.

- *Field tests* are made with the pumping unit installed in its exact environment and operating under existing field or ultimate conditions. The accuracy and reliability of field testing depend on the instrumentation used, installation, and advance planning during the design stages of the installation. By mutual agreement, field tests can be used as acceptance tests.

- *Index tests* are a form of field testing usually made to serve as a standard of comparison for wear, changing conditions, or overhaul evaluation. Index tests should always be run using the same procedures, instruments, and personnel where possible, and a very accurate record and log of events should be kept to give as complete and comparable a history of the results as possible.

- *Model tests* precede the design of the prototype and are usually quite accurate. They supplement or complement field tests of the prototype for which the model was made. The role of the model test must be clearly established as early in the design as possible, preferably in the specification or invitation to bid. Model tests may be used when very large units are involved, when the performances of several models must be compared, and when an advance indication of prototype design is required.

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DEFINITIONS, SYMBOLS, AND UNITS

For a detailed discussion of letter symbols, definitions, description of terms, and table of letter symbols in general use, the user is referred to ASME Power Test Code[1] and to the Technical Data section of this handbook.

Standard Units Used in Pump Testing The following definitions and quantities from the Hydraulic Institute standards[2] are used throughout the industry in pump testing.

VOLUME The standard units of volume are the U.S. gallon and the cubic foot (cubic meter). The standard U.S. gallon contains 231.0 in^3 (0.00379 m^3), and 1 ft^3 = 7.4805 gal (0.028 m^3). Rate of flow is expressed in gallons per minute (cubic meters per hour), cubic feet per second, or million gallons per 24-h day. The specific weight (mass) w of pure water at 68°F (20°C), at sea level and 40° latitude, is 62.315 lb/ft^3 (0.998 kg/liter). For other temperatures or locations, proper specific weight corrections should be made. See Table 1 and the appropriate ASME power test codes.

TABLE 1 Specific weight of water in air

	Temperature, °F (°C)							
Latitude	32 (0)	40 (4.4)	50 (10)	60 (16)	70 (21)	80 (27)	90 (32)	100 (38)
				At sea level				
0°	62.1741	62.1823	62.1654	62.1227	62.0578	61.9729	61.8701	61.7514
10	62.1840	62.1921	62.1753	62.1325	62.0677	61.9828	61.8800	61.7612
20	62.2125	62.2206	62.2038	62.1610	62.0961	62.0112	61.9083	61.7895
30	62.2562	62.2643	62.2475	62.2046	62.1397	62.0547	61.9518	61.8328
40	62.3098	62.3179	62.3011	62.2582	62.1932	62.1082	62.0051	61.8861
50	62.3670	62.3751	62.3582	62.3153	62.2503	62.1652	62.0620	61.9429
60	62.4208	62.4289	62.4120	62.3691	62.3040	62.2188	62.1156	61.9963
70	62.4647	62.4728	62.4559	62.4130	62.3478	62.2626	62.1593	62.0399
				At 6000 ft				
0°	62.1508	62.1589	62.1421	62.0993	62.0345	61.9497	61.8469	61.7282
10	62.1607	62.1688	62.1520	62.1092	62.0444	61.9595	61.8568	61.7380
20	62.1891	62.1972	62.1804	62.1377	62.0728	61.9879	61.8851	61.7663
30	62.2328	62.2409	62.2241	62.1813	62.1164	62.0315	61.9286	61.8097
40	62.2864	62.2946	62.2777	62.2349	62.1699	62.0849	61.9819	61.8630
50	62.3436	62.3517	62.3349	62.2920	62.2270	62.1419	62.0388	61.9198
60	62.3974	62.4055	62.3887	62.3458	62.2807	62.1955	62.0924	61.9732
70	62.4413	62.4495	62.4326	62.3896	62.3245	62.2393	62.1361	61.0168

Note: All values are in pounds per cubic foot, 1 lb/ft^3 = 16 kg/m^3; 1 ft = 0.048 m.
Density of water from Source 1, p. 296
Gravity formula from Source 4, p. 488
$G_\phi = 980.616\ (1 - 0.0026373 \cos 2\phi + 0.0000059 \cos^2 2\phi)$
Altitude correction = 0.0003086 × altitude in meters.
Standard gravity = 980.665 cm/s^2
Density of air from Source 2:
Density = $0.001225(1 - 0.0065H \mid 288.16)^{4.2561}$
where H is in meters
Sources:
1. *Smithsonian Physical Tables,* 9th rev. ed.
2. National Advisory Committee for Aeronautics, TN 3182.
3. American Society of Mechanical Engineers, PTC 2-1971.
4. *Smithsonian Meteorological Tables,* 6th rev. ed.

HEAD The unit for measuring head is the foot (meter). The relation between a pressure expressed in pounds per square inch (kilopascals) and one expressed in feet (meters) of head is

in USCS units $$\text{head, ft} = \text{lb/in}^2 \times \frac{144}{w}$$

in SI units $$\text{Head, m} = \text{kPa} \times \frac{0.102}{w}$$

where w = specific weight of liquid being pumped under pumping conditions, lb/ft^3 (kg/l)

All pressure readings must be converted to feet (meters) of the liquid being pumped, referenced to a datum elevation, which is defined as follows. For a horizontal shaft unit, the datum elevation is the centerline of the pump shaft (Figure 1). For vertical-shaft single-suction pumps, it is the entrance eye to the first-stage impeller (Figure 2). For vertical-shaft double-suction pumps, it is the impeller-discharge horizontal centerline (Figure 3).

VELOCITY HEAD The velocity head h_v is computed from the average velocity V, obtained by dividing the flow by the pipe cross-sectional area. Velocity head is determined at the point of gage connection and is expressed by the formula

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

where V = velocity in the pipe, ft/s (m/s)

g = acceleration due to gravity = 32.17 ft/s^2 (9.81 m/s^2) (Table 2)

FLOODED SUCTION Flooded suction implies that the liquid must flow from an atmospherically vented source to the pump without the average or minimum pressure at the pump datum dropping below atmospheric pressure with the pump operating at specified capacity.

TOTAL SUCTION LIFT Suction lift exists where the total suction head is below atmospheric pressure. Total suction lift h_s is the reading of a liquid manometer or pressure gage at the

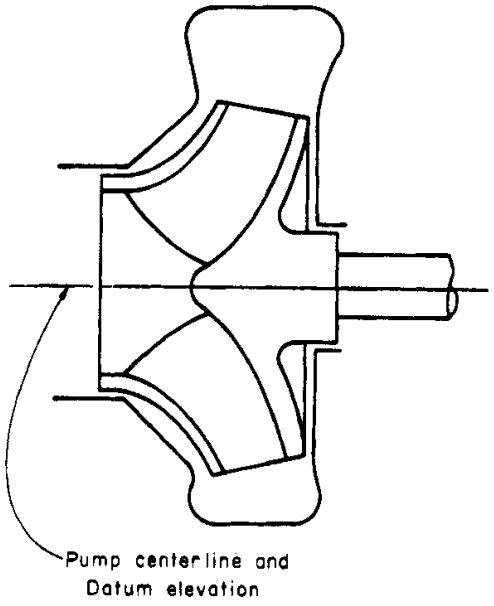


FIGURE 1 Horizontal pump

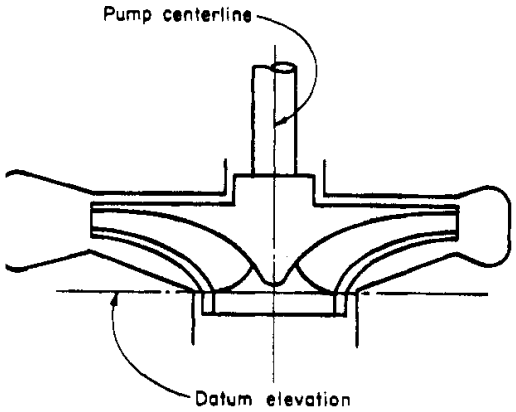


FIGURE 2 Vertical single-suction pump

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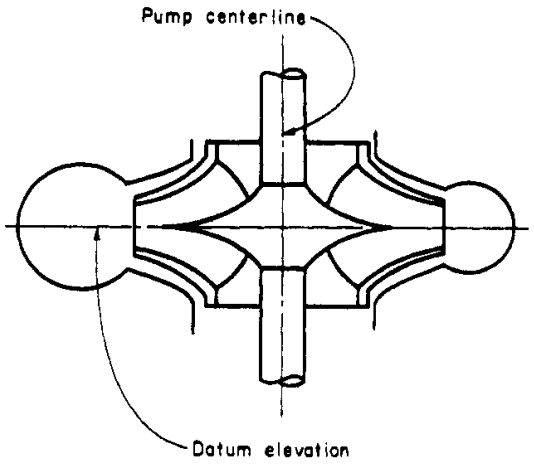


FIGURE 3 Vertical double-suction pump

TABLE 2 Variation of acceleration of gravity with latitude and altitude

	Altitude above mean sea level, ft						
Latitude	0	2000	4000	6000	8000	10,000	12,000
0°	32.0878	32.0816	32.0754	32.0693	32.0631	32.0569	32.0508
10	32.0929	32.0867	32.0805	32.0744	32.0682	32.0620	32.0558
20	32.1076	32.1014	32.0952	32.0890	32.0829	32.0767	32.0705
30	32.1301	32.1239	32.1177	32.1115	32.1054	32.0992	32.0930
40	32.1577	32.1515	32.1454	32.1392	32.1330	32.1269	32.1207
50	32.1872	32.1810	32.1748	32.1687	32.1625	32.1563	32.1501
60	32.2149	32.2087	32.2026	32.1964	32.1902	32.1841	32.1779
70	32.2375	32.2314	32.2252	32.2190	32.2129	32.2067	32.2005

Note: All values are in feet per second per second; 1 ft/s^2 = 0.3048 m/s^2; 1 ft = 0.3048 m

Gravity = 980.616 (1 − 0.0026373 cos 2∅ + 0.0000059 cos^2 20)(1.0/30.48)

Correction for altitude = −0.003086 ft/s^2/1,000 ft

The international standard value of gravity adopted by the International Commission on Weights and Measures is 980.665 cm/s^2 (32.17405 ft/s^2) at sea level and approximately latitude 45°.

Sample computation

Given:

Altitude = 12,000 ft (3657.60 m)

Latitude = 70°

Water temperature = 40°F (4.4°C)

Altitude correction for gravity:

0.0003086 × 3657.60 = 1.128785 cm/s^2

Gravity corrected for latitude and altitude:

Gravity = 980.616 (1 − 0.0026373 cos 140° − 0.0000059 cos^2 140°) − 1.128735 = 981.471784 cm/s^2

Density corrected for gravity:

Density = (0.9999988 × 981.471784/980.6650 × 1.000028) = 1.0007930 g/cm^3

Correcting for buoyancy of air:

Density = 1.0007930 − 0.0008491 = 0.9999438 g/cm^3

Density in USCS units:

Density = 0.9999438 × 62.4279606 = 62.4244543 lb/ft^3

Sources:

1. "Smithsonian Physical Tables," 9th rev. ed.

2. American Society of Mechanical Engineers, PTC 2—1971.

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suction nozzle of the pump, converted to feet (meters) of liquid, and referred to datum minus the velocity head at the point of gage attachment.

TOTAL SUCTION HEAD Suction head exists when the total suction head h_s is above atmospheric pressure. Total suction head is the reading of a gage at the suction of the pump converted to feet (meters) of liquid and referred to datum plus the velocity head at the point of gage attachment.

TOTAL DISCHARGE HEAD Total discharge head h_d is the reading of a pressure gage at the discharge of the pump, converted to feet (meters) of liquid and referred to datum plus the velocity head at the point of gage attachment.

TOTAL HEAD Total head H is the measure of the work increase per pound (kilogram) of liquid, imparted to the liquid by the pump, and is therefore the algebraic difference between the total discharge head and the total suction head. Total head, as determined on test where suction lift exists, is the sum of the total discharge head and total suction lift. Where positive suction head exists, the total head is the total discharge head minus the total suction head.

NET POSITIVE SUCTION HEAD The net positive suction head (NPSH) h_{sv} is the total suction head in feet (meters) of liquid absolute determined at the suction nozzle and referred to datum less the vapor pressure of the liquid in feet (meters) absolute.

DRIVER INPUT The driver input ehp is the input to the driver expressed in horsepower (kilowatts). Usually this is electric input horsepower (kilowatts).

PUMP INPUT Pump input bhp is the power delivered to the pump shaft and is designated as brake horsepower (brake kilowatts).

LIQUID OR WATER POWER Water power whp (wkW) is the useful work delivered by the pump and is usually expressed by the formula

in USCS units $$\text{whp} = \frac{(\text{sp. gr.})QH}{3960}$$

in SI units $$\text{wkW} = 9.8QH(\text{sp. gr.})$$

where sp. gr. = specific gravity of liquid, referred to water at 68°F (20°C)
Q = flow rate, gpm(m^3/h)
H = total head, ft (m)

This formula is derived on p. 16.36.

EFFICIENCY Pump efficiency E_p is the ratio of the power delivered by the pump to the power supplied to the pump shaft; that is, the ratio of the liquid power (also known as water power) to the brake power expressed in percent:

In USCS units $$E_p = \frac{\text{whp}}{\text{bhp}} \times 100$$

In SI units $$E_p = \frac{\text{wkW}}{\text{kW}} \times 100$$

Overall efficiency E_o is the ratio of the power delivered by the pump to the power supplied to the input side of the pump driver; that is, the ratio of the output power to the input power to the driver.

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In USCS units $$E_o = \frac{\text{whp}}{\text{ehp}} \times 100$$

In SI units $$E_o = \frac{\text{wkW}}{\text{kW}} \times 100$$

PRIME MOVER RATINGS The prime movers for driving pumps are rated according to established standards. For example, for electric motor drivers, see the standards of the latest edition of the *National Electrical Manufacturers Association*.

ACCURACY AND TOLERANCES

Accuracy The accuracy to which tests can be made depends on the instruments used, their proper installation, the skill of the test engineer, and the shop tests for the simulation of field conditions. The test engineer must have sufficient knowledge of the characteristics and limitations of the test instruments to obtain maximum accuracy when using them, along with a thorough understanding of the pumps, prime movers, controls, and installation peculiarities to interpret the results. For shop testing, the acceptable deviations and fluctuations of the instrumented test readings are given in Table 1.11 of the *ASME Performance Test Code for Centrifugal Pumps* (ASME PTC 8.2-1990). These deviations are not to be misconstrued as tolerances, which must be spelled out in the specifications. The limits of accuracy of pump test measuring devices for use in field testing are shown in Figure 4. Using these limits, the combined accuracy of the efficiency is the square root of the quantity [square of the head accuracy plus square of the flow rate accuracy plus square of the power input accuracy]:

$$A_c = \sqrt{(\pm H^2) + (\pm Q^2) + (\pm \text{ehp}^2)} \text{ (percent)}$$

Pump speed and voltage are not required for efficiency computations, and so the values for these are not included in this formula.

Instrumentation All instruments should be calibrated before the tests, and all calibration and correction data or curves should be prepared in advance. Where required, a certified calibration curve showing the calibration of the instrument, including any procedures for establishing a coefficient, should be furnished before testing begins. The specifications should be explicit in regard to a waiving of these calibration requirements. After testing, all instruments should be recalibrated. Any differences between before and after calibration values must be resolved either by retest or by acceptable variations being spelled out in the specifications.

Tolerances The tolerances in pump performance permitted are usually given in the specifications. The user can and should make these requirements known before the order for pumping apparatus is placed. The test tolerances permitted by the many organizations are commonly used.

TEST REQUISITES

Operating Conditions The primary factors affecting the operation of a pump are the inlet (suction), outlet (discharge or total head), and speed. The secondary factors are physical and climatic variables, such as the temperature, viscosity, specific weight, and turbidity of the liquid being pumped and the elevation of the pumping system above sea level. In some installations, it is impossible to measure discharge or even head accurately. In these instances, good shop tests are essential. It follows then that, in order for the shop test to predict the field performance of a pump, the field operating, installation, and suction conditions should be simulated.

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Quantity to be measured	Measuring device	Calibrated limit of accuracy plus or minus, %
Capacity	Venturi meter	1/2
	Nozzle	1
	Pitot tube	1 1/2
	Orifice	1 1/4
	Disk	2
	Magnetic flowmeter	1/2
	Piston	1/4
	Volume or weight-tank	1
	Propeller meter	4
Head	Electric sounding line	1/4
	Air line	1/2
	Instrument grade pressure transducer:	
	1/4–1/2 scale	1 %
	1/2–3/4 scale	1/2 %
	3/4–1 scale	1/4 %
	Liquid manometer, 3- to 5-in (75- to 127-mm) deflections	3/4
	Liquid manometer, over 5-in (127-mm) deflections	1/2
	Bourdon gage—5 in (127-mm) min dial:	
	1/4–1/2 full scale	1
	1/2–3/4 full scale	3/4
	Over 3/4 scale	1/2
Power input	Watt-hour meter and stopwatch	1 1/2
	Portable recording wattmeter	1 1/2
	Test precision wattmeter:	
	1/4–1/2 scale	3/4
	1/2–3/4 scale	1/2
	Over 3/4 scale	1/4
	Clamp on ammeter	4
Speed	Revolution counter and stopwatch	1 1/4
	Handheld tachometer	1 1/4
	Stroboscope	1 1/2
	Automatic counter and stopwatch	1/2
	Digital tachometer	1/8
Voltage	Test meter	
	1/4–1/2	1
	1/2–3/4	%
	3/4-full	%
	Rectifier voltmeter	5

Note: Instruments with accuracy significantly different than shown are available. Refer to the specific instruction manual.

Source: ANSI B-58.1 (AWWAE 101–61).

FIGURE 4 Typical Limits of accuracy of pump test measuring devices in field use

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The inlet passages are critical, and the sump where used on the suction lift must be duplicated as closely as possible. During the shop tests, no total suction head less than specified should be permitted, nor should the suction head exceed the specified amount in cases where cavitation or possibly operating "in the break" could occur.

For field installations above sea level, the difference in elevation between the shop test site and the field installation must be taken into account by reducing to the barometric pressure at the specified elevation. This is especially true if a suction lift or negative suction head is involved. Standard tables of barometric pressures are available for use in computing the data, and the tables to be used should be acceptable to all interested parties.

Flow Control The most common means of adjusting pump flow during a test is by using a valve to provide the required change in system head. The valve should be selected so that the minimum and maximum head conditions may be attained. Since valve cavitation is a common problem in tests using butterfly, gate, or globe valves, special noncavitating valves may be needed for high power tests, or tests requiring a significant head breakdown.

Cavitation Tests Cavitation tests should be run if required by the specifications (provided such tests are needed and have not been previously conducted on similar pumps and certified by the manufacturer) or if needed to assume a successful pump installation.

The suction requirements that must be met by the pump are usually defined by the cavitation coefficient σ. Plant σ is defined as NPSHA (net positive suction head available) divided by total pump head per stage:

$$\sigma = \frac{NPSHA}{H}$$

Three typical arrangements for determining the cavitation characteristics of pumps are illustrated in Figure 5. In Figure 5A, the suction is taken from a sump with a constant-level surface. The liquid is drawn first through a valve (throttle) and then through a section of pipe containing screens and straightening devices, such as vanes and baffles. This setup will dissipate the turbulence created by the suction valve and will also straighten the flow so the pump suction flow will be relatively free from undue turbulence. In Figure 5B, the suction is taken from a relatively deep sump or well in which the water surface can be varied over a fairly large range to provide the designed variation in suction lift. In Figure 5C, the suction is taken from a closed vessel in a closed loop in which the pressure level can be varied by a gas pressure over the liquid, by temperature of the liquid, or by a combination of these.

By using one of these cavitation test arrangements, the critical value of σ (that is, the value at which cavitation will begin) can be found by one of the following two methods:

1. Constant speed and capacity vary the suction lift. Run the pump at constant speed and capacity with the suction lift varied to produce cavitation conditions. Plots of the head, efficiency, and power input against σ as shown in Figure 6.

   When the values of σ are held high, the values of head, efficiency, and power should remain relatively constant. As σ is reduced, a point is reached when the curves break from the normal, indicating an unstable condition. This breakaway condition may and usually does impair the operation of the pump. The extent of impairment depends on the size, specific speed, and service of the pump and on the characteristics of the pumped fluid. A variation of this method is to plot results using capacities both greater and less than normal, as shown in Figure 7.
2. Constant speed and suction lift vary the capacity. Run the pump at constant speed and suction lift and vary the capacity. For a given suction lift, the pumping head is plotted against capacity. A series of such tests will result in a family of curves, as shown in Figure 8.

Where the plotted curve for any suction condition breaks away from the normal, cavitation has occurred. The value of σ may be calculated at the breakaway points by dividing NPSHA by total head H at the point under consideration.

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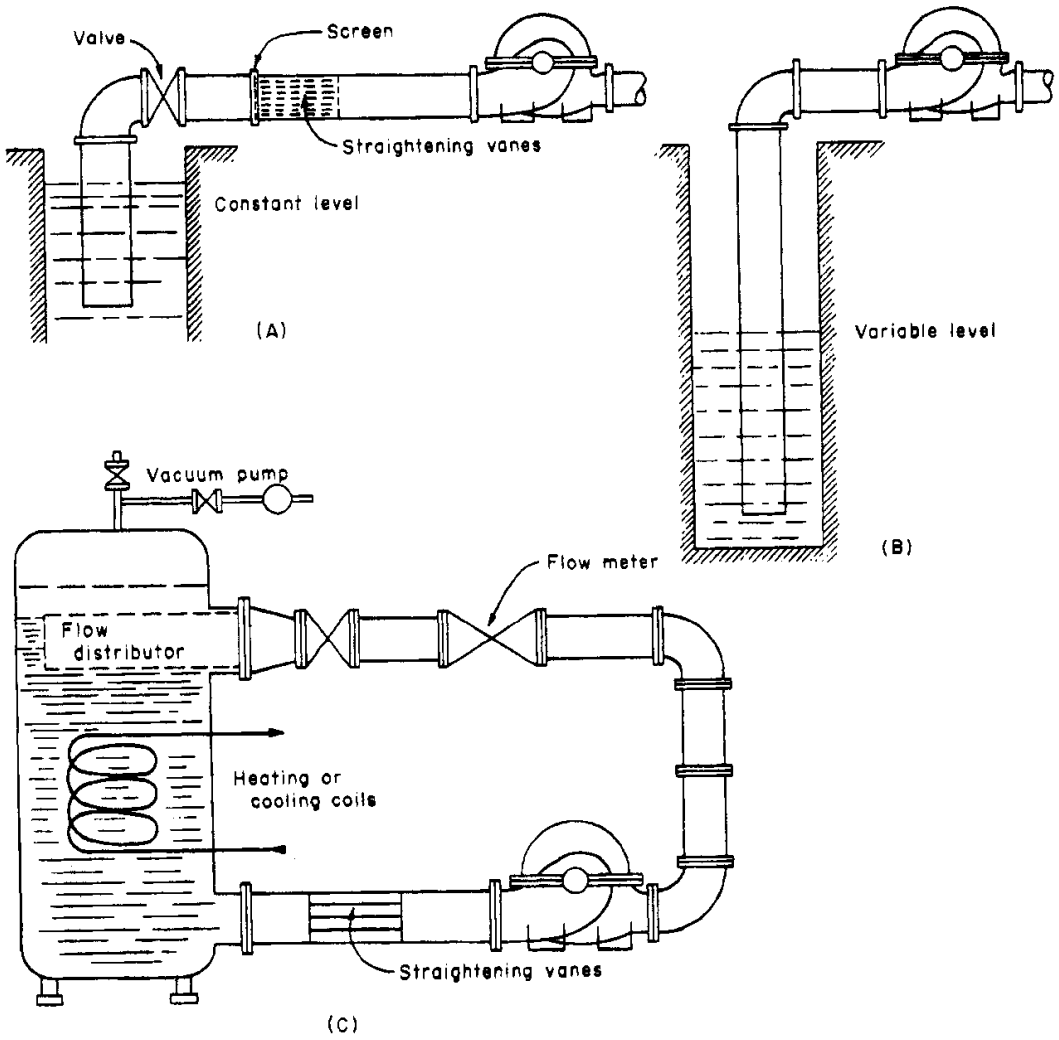


FIGURE 5A through C Typical arrangement for determining cavitation characteristics (Hydraulic Institute Standards, 13th Edition—out of print[2])

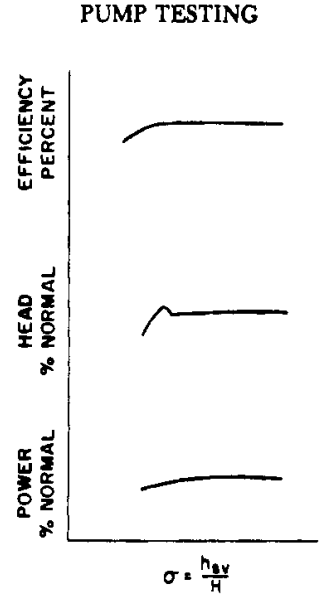


FIGURE 6 Functions of sigma at constant capacity and speed; suction pressure varied

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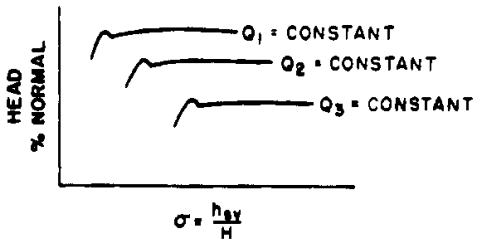


FIGURE 7 Sigma capacities above and below normal; suction pressure varied

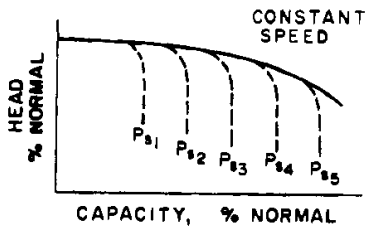


FIGURE 8 Typical cavitation curves at constant speed and suction pressure

Testing at Zero Flow In some cases, pumps can be damaged by testing at zero flow (shut-off). As pump power increases, and casing water volume decreases, this problem becomes acute. Prior to test, all parties should agree on the minimum flow to be demonstrated.

TEST PROCEDURE

Agreements The specifications and contract should be very clear on any special points that must be covered by the pump testing. All interested parties shall be represented and given equal rights in regard to test date, setup, conditions, instrumentation, calibration of instruments, examination of pump and test setup, and accuracy of results and computations. Any controversial points or methods not provided for in the specifications should be resolved to the satisfaction of all interested parties before testing is begun.

In some special instances and by agreement between parties, an independent test expert may be engaged to take over full responsibility of the test stand and apparatus. This person will make all decisions after consultation with the interested parties and should be used only where an impasse is reached.

Normally, the manufacturer will establish the time and date for the pump tests. In some cases, the specifications will cover such items as duration, limiting date, and notification time. Reasonable notice must be given to all official witnesses or representatives. A 30-day notice is preferred, and 1 week should be considered a bare minimum.

Any time limitation regarding correction of mechanical defects or equipment malfunctions that arise during testing must be resolved by mutual agreement.

Observers and Witnesses Representatives from each party to the contract shall have equal opportunity to attend the testing. Where more than one representative from one of the parties is present, their function as observers, official witnesses, or representatives must be made clear before the start of testing. The number of representatives present from any one party shall not be a deciding factor when disagreements are being resolved. Any comments or constructive criticism from the observers and witnesses should be duly considered.

Inspection and Preliminary Operation All interested parties shall make as complete an inspection as possible before, during, and after the test to determine compliance with specifications and correct connection of all instrumentation. The following items should be inspected before or during the test:

- Impeller and casing passages
- Pump and driver alignment
- Piezometer openings
- Electrical connections
- Lubricating devices and system
- Wearing ring and other clearances
- Stuffing box or mechanical seal adjustment and leakage

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This is not a complete list of items and should be taken as only a guide for the interested parties. Instruments installed on the pump to obtain the necessary test information shall not affect the pump operation or performance. If a question arises as to the effect an instrument has on the operation, it should be resolved by all parties. Where necessary, comparative preliminary tests can be conducted with the disputed equipment removed and then reinstalled. The dimensions at the piezometer connections on both the suction and discharge sides must be accurately determined to permit accurate determination of the velocity head correction.

On satisfactory completion of the preliminary inspection, the pump may be started. The pump and all instrumentation should then be checked for proper operation, scale readings, or evidence of malfunction. When all equipment and apparatus are functioning properly, a preliminary test run should be made. If possible, this run should be made at or near the rated condition. The correct procedures for observing and recording the data should be established during this run. Also, the time it takes to obtain steady test conditions is determined for use in the pump test runs. In test systems with relatively long circuits (like pipelines), the time needed to reach steady-state conditions increases dramatically. The acceptable deviations and fluctuations for test readings are given in the section named "Accuracy."

Suggested Test Procedure

TIME AND DATE After tests have been decided upon, it is to the best interests of all parties to conduct them with the least delay. The contract normally will not give a time or date for the test, but will specify a completion date for submission or approval of a final test report.

PERSONNEL Pump testing, regardless of classification, should be carried out by personnel specially trained in the operation of the test equipment used. Representatives from each party to the contract shall be given equal opportunity to witness the test or tests and shall also have equal voice in commenting on the conduct of the tests or on compliance with specifications or code requirements where applicable.

SCHEDULE A schedule should be agreed upon by all parties in advance of the test. The schedule should be as complete a program as possible and give some particulars on the range of test heads, discharge rates, and speed to be used. This schedule should be flexible and subject to change, especially after the preliminary runs have been made.

INSPECTION The pump and test setup should be thoroughly inspected both before and after the tests. Special attention should be given to the hydraulic passages and pressure taps near the suction and discharge sections. Also, the discharge measuring device should be inspected.

CALIBRATION OF INSTRUMENTS While the setup is being inspected, all measuring devices should be calibrated and adjusted as explained under the section named "Instrumentation."

PRELIMINARY TESTS After it has been determined that the test setup complies with the installation and specification requirements and that the instrumentation is properly installed, the pump is started. A sufficient number of preliminary test runs should be made to check the functioning of the test stand and all control and measuring devices. These preliminary tests also give the test personnel and representatives an opportunity to check and adjust the entire setup and serve as a basis for agreements on accuracy and compliance. Each test point is held until satisfactory stable conditions exist. The acceptable fluctuations in test readings is covered under "Accuracy." It is suggested that preliminary computations be made, plotted, and analyzed prior to actual test runs. (*Note:* Most pump manufacturers will conduct a complete preliminary shop test to be sure of specifications and contractual compliance before inviting the purchaser's representatives to witness the official test.)

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OFFICIAL TEST RUNS The test points for the official test runs must be sufficient in number to establish the head-discharge curve over the specified range and to provide any other data needed to compute or plot the information required by the specifications. It is suggested that one test run be as near the rated condition as possible and that at least three runs be in the specified operating range of the pump.

LOGGING OF EVENTS In most instances, the official pump test is conducted by only two, three, or four test engineers, and no record other than test data is needed. In more complicated or important tests, it is sometimes very desirable to assign someone the task of recording and logging events as they occur. This is most important if reruns are required. Complete records, including any notes or comments on inspection and calibration, shall be kept of all data, readings, observations, and information relevant to the test. A suggested form for shop and field tests is shown in Figure 28.

PRELIMINARY COMPUTATIONS Sufficient preliminary computations should be made to determine whether all specification requirements have been met and whether reruns will be necessary.

RERUNS When the preliminary computations indicate that reruns are necessary, they should be run immediately or as soon as possible after the official test runs and with the same personnel, instruments, and devices. Sometimes mechanical or electrical faults will necessitate a rerun. If after correction of these faults several reruns indicate a change, a complete retesting may be required. Any official representatives shall have the right to ask for a rerun or be shown to their satisfaction that a rerun is not required.

COMPUTATIONS, PLOTTING REPORTS These topics are discussed later in the text.

TEST MEASUREMENTS

DISCHARGE The choice of which method of discharge measurement to use should be made by an agreement between all parties concerned. Some test codes and procedures in regular use permit or even recommend certain methods for model or shop testing, but restrict their use in field or index testing. Some methods are more adaptable to the site conditions than others, and so the test engineers and interested parties should be completely familiar with the several methods applicable before settling on the one to be used.

The most commonly used methods of discharge measurement are those that use quantity meters and those that use rate-of-flow meters. Both meters are usually classified as liquid meters, and their functions are listed in Table 3.

QUANTITY METERS The term *quantity* is here used to designate those meters in which the fluid passes through the primary element in successive and more or less completely isolated

TABLE 3 Liquid meters and their functions

Quantity meters	Rate-of-flow meters
Weighing meters	Differential pressure meters
Weighing tank	Venturi
Tilting trap	Nozzle
Volumetric meters	Orifice plate
Tank	Pitot tube
Reciprocating piston	Head area meters
Rotary piston	Weir
Nutating disk	Flume
	Current meters
	Velocity meters (magnetic, ultrasonic)

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quantities, either weights or volumes, by alternately filling and emptying containers of known capacities. The secondary element of a quantity meter consists of a counter with suitably graduated dials for registering the total quantity that has passed through. Quantity meters are classified into two groups: weighing meters and volumetric meters.

WEIGHING METERS There are two types of weighing meters: weighing tank and tilting trap. In the tilting trap meter, the equilibrium of a container is upset by a rise of the center of gravity as the container is filled. Weighing tank meters employ a container suspended from a counterbalanced scale beam. The weighing tank and the tilting trap are affected slightly by the temperature of the liquid but not enough to cause concern in normal testing.

VOLUMETRIC METERS Volumetric meters measure volumes instead of weights. There are four types: tank, reciprocating piston, rotary piston, and nutating disk.

Tank meters are a very elementary form of meter of limited commercial importance. As the name implies, they consist of one or more tanks that are alternately filled and emptied. The height to which they are filled can be regulated manually or automatically. In some cases, the rising liquid operates a float that controls the inflow and outflow; in others, it starts a siphon. Occasionally, some tank meters have been erroneously classified as weighing meters.

Reciprocating piston meters use one or more members that have a reciprocating motion and operate in one or more fixed chambers. The quantity per cycle can be adjusted either by varying the magnitude of movement of one or more of the reciprocating members or by varying the relation between the primary and secondary elements.

Rotary (or oscillating) piston meters have one or more vanes that serve as pistons or movable partitions for separating the fluid segments. These vanes may be either flat or cylindrical and rotate within a cylindrical metering chamber. The axis of rotation of the vanes may or may not coincide with that of the chamber. The portion of the chamber in which the fluid is measured usually includes about 270°. In the remaining 90°, the vanes are returned to the starting position for closing off another segment of fluid. This may be accomplished by the use of an idle rotor or gear, a cam, or a radial partition. The vanes must make almost a wiping contact with the walls of the measuring chamber. The rotation of the vanes operates the counter.

Nutating disk meters have the disk mounted in a circular chamber with a conical roof and either a flat or conical floor. When in operation, the motion of the disk is such that the shaft on which it is mounted generates a cone with the apex down. However, the disk does not rotate about its own axis; this is prevented by a radial slot that fits about a radial partition extending in from the chamber sidewall nearly to the center. The peculiar motion of the disk is called *nutating*. The inlet and outlet openings are in the sidewall of the chamber on either side of the partition. These meters are usually adjusted by changing the relation between the primary and secondary elements.

RATE-OF-FLOW METERS The term *rate of flow* is applied to all meters through which the fluid passes not in isolated quantities but in a continuous stream. The movement of this fluid stream through the primary element is directly or indirectly utilized to actuate the secondary element. The quantity of flow per unit time is derived from the interactions of the stream and the primary element, using physical laws supplemented by empirical relations.

In rate-of-flow meters, the functioning of the primary element depends upon some property of the fluid other than, or in addition to, volume or mass. This property may be kinetic energy (head meters), inertia (gate meters), specific heat (thermal meters), or the like. The secondary element senses a change in the property concerned and usually embodies some device that draws the necessary inferences automatically, so the observer can read the rate of flow from a dial or chart. In some cases, the secondary element records pressures, such as static and differential, from which the rate of flow and time-quantity flow must be computed. In others, the secondary element not only indicates the rate of flow but also integrates it with respect to time and records the total quantity that has passed through the meter. In some cases, the indications of the secondary element are transmitted to a point some distance from the primary element.

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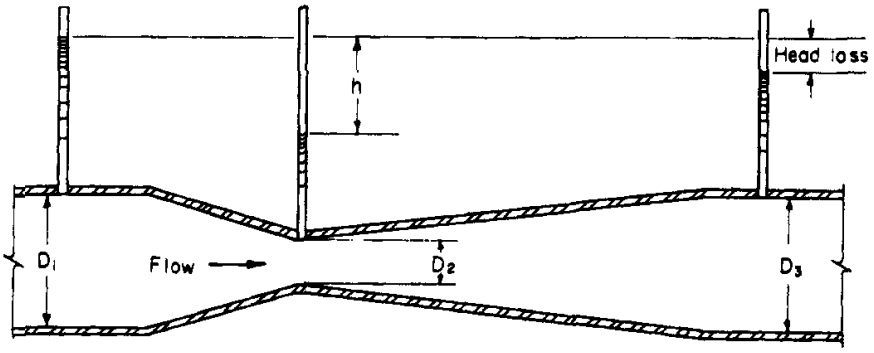


FIGURE 9 Diagram of venturi meter

DIFFERENTIAL PRESSURE METERS With this group of meters the stream of fluid creates a pressure difference as it flows through the primary element. The magnitude of this pressure difference depends upon the speed and density of the fluid and features of the primary element.[3]

Flow in a pipeline, or closed pressure conduit, can be measured by a wide variety of methods, and the choice of method for a particular installation will depend upon prevailing conditions. The accuracy of flow measurements in pressure conduits made with properly selected, installed, and maintained measuring equipment, such as venturi meters, flow nozzles, orifice meters, and pitot tubes, can be very high.

The venturi meter (Figure 9) is perhaps the most accurate flow measuring device that can be used in a water supply system. It contains no moving parts, requires very little maintenance, and causes very little head loss. Venturi meters operate on the principle that flow in a closed conduit system is faster through areas of small cross section (D_2 in Figure 9) than through areas of large cross action (D_1). The total energy in the flow, consisting primarily of velocity head and pressure head, is essentially the same at D_1 and D_2. Thus the pressure must decrease in the constricted throat D_2, where the velocity is higher, and conversely must increase at D_1 upstream from the throat, where the velocity is lower. This reduction in pressure from the meter entrance to the meter throat is directly related to the rate of flow through the meter and is the measurement used to determine flow rate.

The coefficient of discharge for the venturi meter ranges from 0.935 for small-throat velocities and diameters to 0.99 for large-throat velocities and diameters. Equations for the venturi meter are

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

$$Q' = 3.118\,\frac{CA_2'\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

where Q = rate of flow, ft³/s (m³/s)
C = coefficient of discharge for meter
A_2 = area of throat section, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)
h = differential head of liquid between meter inlet and throat, ft (m)
β = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A'_2 = area of throat section, in²

Flow nozzles operate on the same basic principle as venturi meters. In effect, the flow nozzle is a venturi meter that has been simplified and shortened by omission of the long

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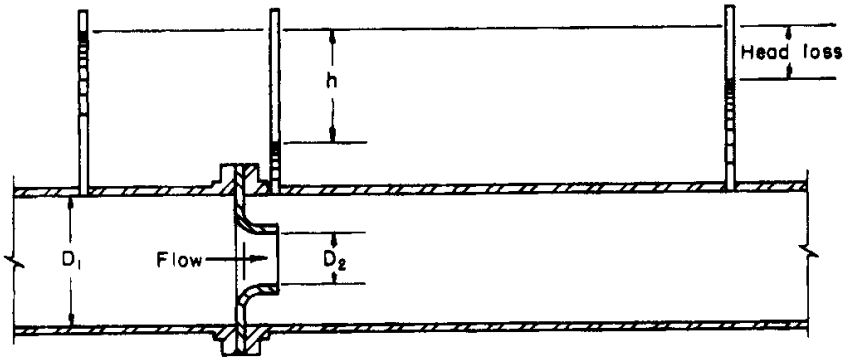


FIGURE 10 Diagram of flow nozzle

diffuser on the outlet side (Figure 10). The streamlined entrance of the nozzle provides a straight cylindrical jet without contraction, so the coefficient of discharge is almost the same as that for the venturi meter. In the flow nozzle, the jet is allowed to expand of its own accord, and the high degree of turbulence created downstream from the nozzle causes a greater loss of head than occurs in the venturi meter, where the diffuser suppresses turbulence. The relationship of flow rate to head and flow nozzle dimensions is

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

$$Q' = \frac{3.118CA_2'\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

where Q = rate of flow, ft³/s (m³/s)
C = coefficient of discharge for nozzle
A_2 = area of nozzle throat, ft² (m²)
g = acceleration of gravity, 32.17 ft/s² (9.81 m/s²)
h = head at or across the nozzle, ft (m)
β = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A_2' = area of nozzle throat, in²

In an orifice meter, a thin plate orifice inserted across a pipeline is used for measuring flow in much the same manner as a flow nozzle (Figure 11). The upstream pressure connection is often located about one pipe diameter upstream from the orifice plate. The

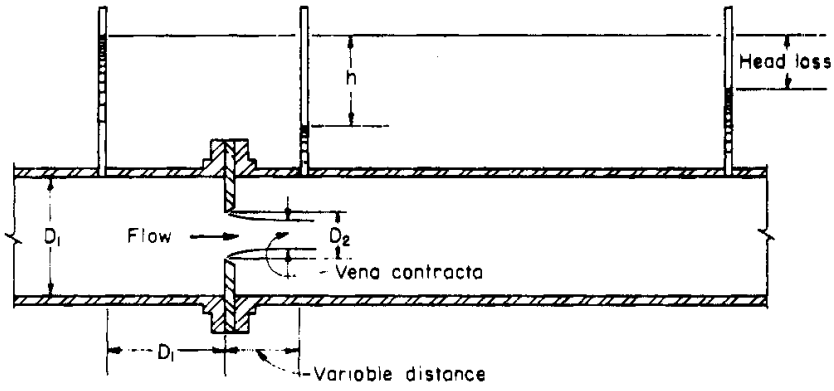


FIGURE 11 Diagram of orifice meter

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pressure of the jet ranges from a minimum at the *vena contracta* (the smallest cross section of the jet) to a maximum at about four or five conduit diameters downstream from the orifice plate. The downstream pressure connection (the center connection in Figure 11) is usually made at the *vena contracta* to obtain a large pressure differential across the orifice.

The pressure tap openings should be free from burrs and flush with the interior surfaces of the pipe. Equations for the orifice plate are

$$Q = \frac{CA_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

$$Q' = \frac{3.118CA'_2\sqrt{2gh}}{\sqrt{1-\beta^4}}$$

where Q = rate of flow, ft^3/s (m^3/s)
C = coefficient of discharge for orifice plate
A_2 = area of orifice, ft^2 (m^2)
g = acceleration of gravity, 32.17 ft/s^2 (9.81 m/s^2)
h = head across the orifice plate, ft (m)
β = ratio of throat to inlet diameter (D_2/D_1)
Q' = rate of flow, gpm
A'_2 = area of orifice, in^2

The principal disadvantage of orifice meters, compared with venturi meters and flow nozzles, is their greater loss of head. On the other hand, they are inexpensive and capable of producing accurate flow measurements.

It should be noted that the relationship of flow rate to head and dimensions of the metering section is identical for the venturi meter, flow nozzle, and orifice meter except that the coefficients of discharge vary.

Where it is impossible to employ one of the methods previously described, the pitot tube is often used. A pitot tube in its simplest form consists of a tube with a right-angle bend which, when partly immersed with the bent part under water and pointed directly into the flow, indicates flow velocity by the distance water rises in the vertical stem. The pitot tube makes use of the difference between the static and total pressures at a single point.

The height of rise h of the water column above the water surface, expressed in feet (meters) and tenths of feet (millimeters), equals the velocity head $v^2/2g$. The velocity of flow v in feet (meters) per second may thus be determined from the relation $v = \sqrt{2gh}$.

In a more complete form known as the pitot static tube, the instrument consists of two separate, essentially parallel parts, one for indicating the sum of the pressure and velocity heads (total head) and the other for indicating only the pressure head. Manometers are commonly used to measure these heads, and the velocity head is obtained by subtracting the static head from the total head. A pressure transducer may be used instead of the manometer to measure the differential head. Oscillographic or digital recording of the electric signal from the transducer provides a continuous record of the changes in head.

The simple form of the pitot tube has little practical value for measuring discharges in open channels handling low-velocity flows because the distance the water in the manometer tube rises is difficult to measure. This limitation is overcome to a large extent by using a pressure transducer for the measurement and precise electronic equipment for the data readings.

The pitot static tube, on the other hand, works very well for this purpose if the tube is used with a differential manometer of the suction lift type (Figure 12). In this manometer, the two legs are joined at the top by a T that connects to a third line, in which a partial vacuum can be created. After the pitot tube has been bled to remove all air, water flows up

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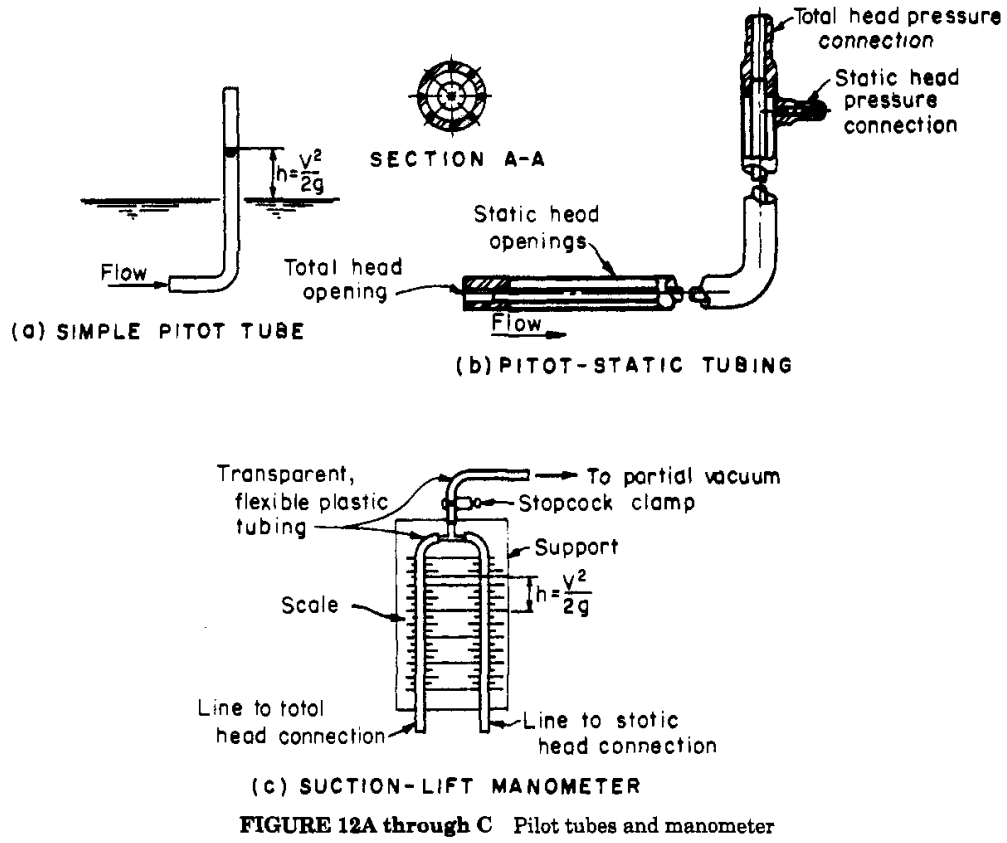


FIGURE 12A through C Pilot tubes and manometer

through it into the manometer to the height desired for easy reading. Then the stopcock or clamp on the vacuum line is closed. The partial vacuum acts equally on the two legs and does not change the differential head. The velocity head h is then the difference between the total head reading and the static head reading. If desired, a pressure transducer can also be used for the head measurement.

Pitot tubes can be used to measure relatively high velocities in canals, and it is often possible to make satisfactory discharge measurements at drops, chutes, overfall crests, or other stations where the water flows rapidly and fairly large velocity heads occur. At low velocities, values of h become quite small. The pitot tube head error for a low velocity will lead to a much larger inaccuracy in discharge computation than the same error when the velocity is high. The velocity traverse with a pitot tube may be made in the same manner as with a current meter (discussed later).

HEAD AREA METERS The instruments used to measure flow in open conduits are normally classified as head area meters, the most common of which are weirs and flumes.

A weir is an overflow structure built across an open channel. Weirs are one of the oldest, simplest, and most reliable structures for measuring the flow of water in canals and ditches. These structures can be easily inspected, and any improper operations can be quickly detected and corrected.

The discharge rates are determined by measuring the vertical distance from the crest of the overflow portion of the weir to the water surface in the pool upstream from the crest and referring to computation tables that apply to the size and shape of the weir. For standard tables to apply, the weir must have a regular shape and definite dimensions and must be placed in a bulkhead and pool of adequate size so the system performs in a standard manner.

Weirs may be termed rectangular, trapezoidal, or triangular, depending upon the shape of the opening. In rectangular and trapezoidal weirs, the bottom edge of the opening is the crest and the side edges are called *sides* or *weir ends* (Figures 13 and 14). The sheet of

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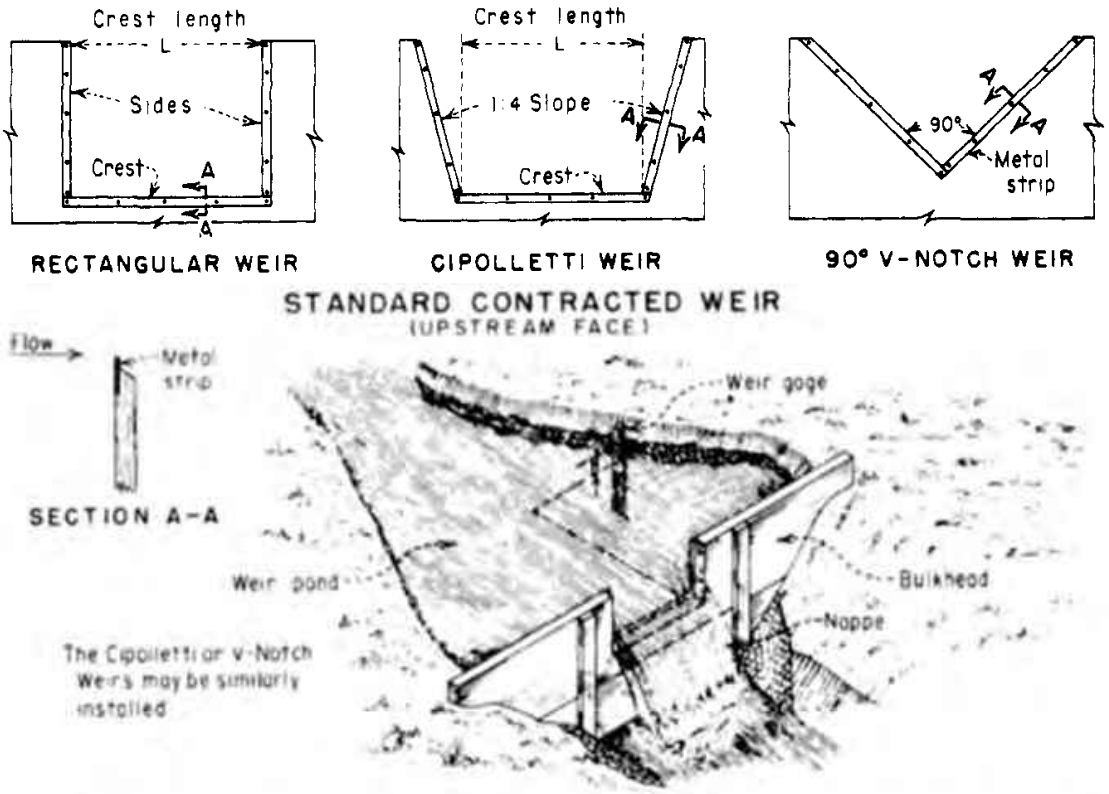


FIGURE 13 Standard contracted weirs and temporary bulkhead with contracted rectangular weir discharging at free flow

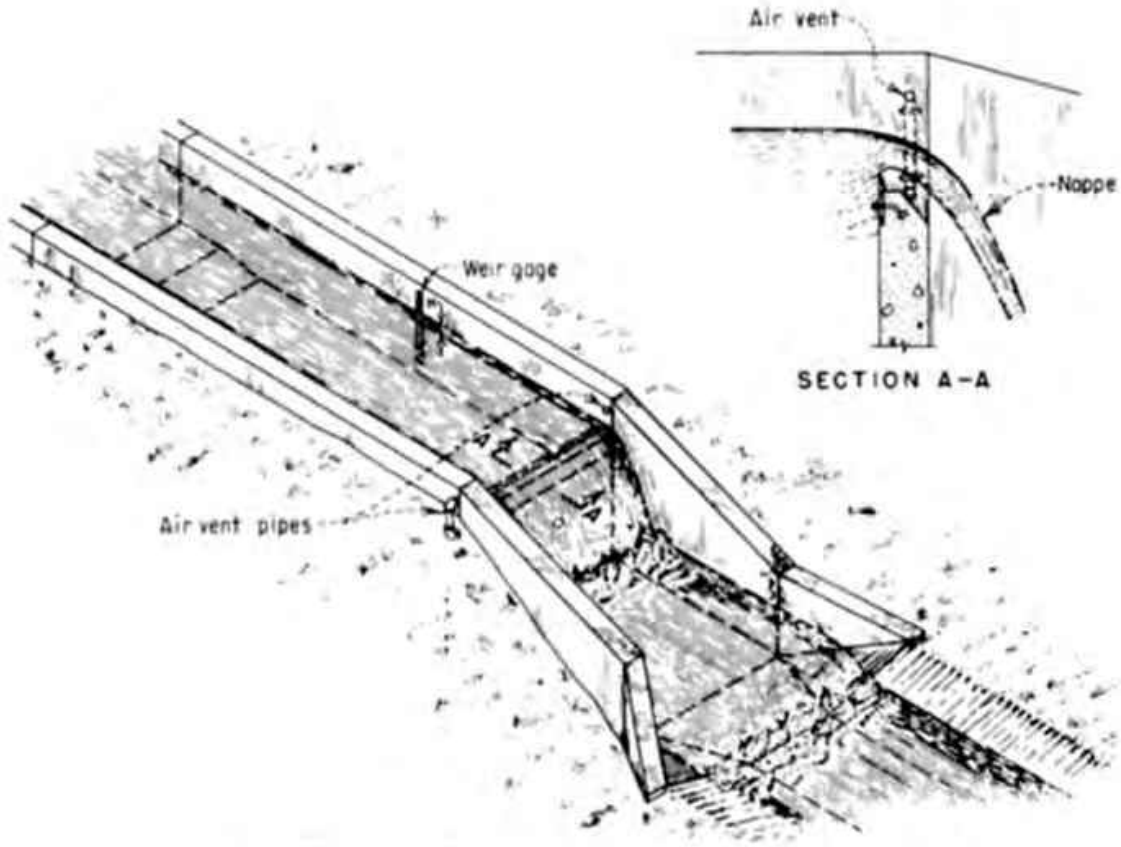


FIGURE 14 Typical suppressed weir in a flume drop

water leaving the weir crest is called the *nappe*. In certain submerged conditions, the under-nappe airspace must be ventilated to maintain near-atmospheric pressure.

The types of weirs most commonly used to measure water are

- Sharp-crested and sharp-sided Cipolletti weirs
- Sharp-sided 90° V-notch weirs
- Sharp-crested contracted rectangular weirs
- Sharp-crested suppressed rectangular weirs

For measuring water flow, the type of weir used has characteristics that make it suitable for a particular operating condition. In general, for best accuracy, a rectangular suppressed weir or a 90° V-notch weir should be used.

The discharge in second-feet (cubic meters per second) over the crest of a contracted rectangular weir, a suppressed rectangular weir, or a Cipolletti weir is determined by the head H in feet (meters) and the crest length L in feet (meters). The discharge of the standard 90° V-notch weir is determined directly by the head on the bottom of the V notch.

As the stream passes over the weir, the top surface curves downward. This curved surface, or drawdown, extends upstream a short distance from the weir notch. The head H must be measured at a point on the water surface in the weir pond beyond the effect of the drawdown. This distance should be at least four times the maximum head on the weir, and the same gage point should be used for lesser discharges. A staff gage (Figure 15) having a graduated scele with the zero placed at the same elevation as the weir crest is usually provided for the head measurements.

Two widely used sets of formulas for computing discharge over standard contracted rectangular weirs are those of Smith[4] and Francis.[5] The formulas proposed by Smith require the use of coefficients of discharge that vary with the head of water on the weir and with the length of the weir. Consequently, the Smith formulas are somewhat inconvenient to use, although they are accurate for the ranges of coefficients usually given. For this type of weir operating under favorable conditions as prescribed in preceding paragraphs, the Francis formula when velocity of approach is neglected is

in USCS units $$Q = 3.33H^{3/2}(L - 0.2H) \tag{1}$$

in SI units $$Q = 1.837H^{3/2}(L - 0.2H)$$

and the formula when velocity of approach is included is

in USCS units $$Q' = 3.33[(H + h)^{3/2} - h^{3/2}](L - 0.2H)$$

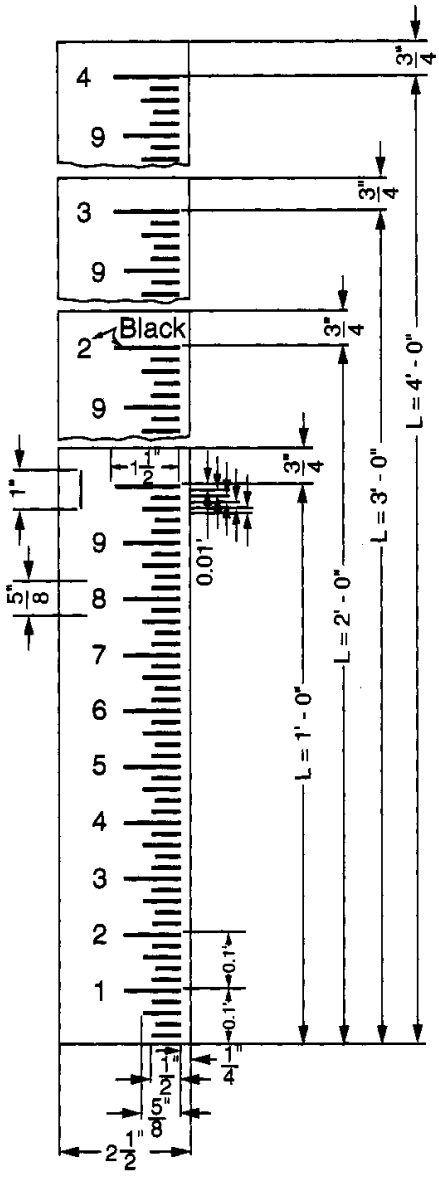
in SI units $$Q' = 1.837[(H + h)^{3/2} - h^{3/2})(L - 0.2H) \tag{2}$$

where Q = discharge neglecting velocity of approach, ft^3/s (m^3/s)
Q' = discharge considering velocity of approach, ft^3/s (m^3/s)
H = head on weir, ft (m)
L = length of weir, ft (m)
h = head due to velocity of approach ($v^2/2g$), ft (m)

Note that the Francis formulas contain constant discharge coefficients that allow computation without the use of tables. A table of $\frac{3}{2}$ powers that provides values of $H^{3/2}$, $h^{3/2}$, and $(H + h)^{3/2}$ for convenience in computing discharge with the Francis formulas may be found in hydraulic handbooks.

The principal formulas used for computing the discharge of the standard suppressed rectangular weir were also proposed by Smith and Francis. In the Smith formulas for suppressed weirs, as for contracted weirs, coefficients of discharge vary with weir head and length; therefore, these formulas are not convenient for use in computations without tables or coefficients.

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NOTES

Material of 18 gage (U.S. Standard) metal coated with substantial thickness of porcelain enamel. Face of gage is white. Numerals and graduations are black. Graduations are sharp and accurate to dimensions shown.

Length "L" represents gate limits.

Gages may be made in any length desired using similar details.

FIGURE 15 Standard weir, or staff, gage

The Francis formula for the standard suppressed rectangular weir neglecting velocity of approach in

in USCS units $$Q = 3.33LH^{3/2}$$

in SI units $$Q = 1.857LH^{3/2} \quad (3)$$

and that including velocity of approach is

in USCS units $$Q' = 3.33L[(H + h)^{3/2} - h^{3/2}]$$

in SI units $$Q' = 1.837L[(H + h)^{3/2} - h^{3/2}] \quad (4)$$

In these formulas, the letters have the same significance as in the formulas for contracted rectangular weirs. The coefficient of discharge was obtained by Francis from the same general set of experiments as those used for the contracted rectangular weir. No tests have been made to determine the applicability of these formulas to weirs less than 4 ft (1.2 m) in length.

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The Cipolletti weir is a contracted weir and must be installed as such to obtain reasonably correct and consistent discharge measurements. However, Cipolletti has compensated for the reduction in discharge due to end contractions by sloping the sides of the weir sufficiently to overcome the effect of contraction. The Cipolletti formula, in which the Francis coefficient is increased by about 1% and velocity of approach is neglected, is

in USCS units $$Q = 3.367LH^{3/2} \quad (5)$$

in SI units $$Q = 1.858LH^{3/2}$$

The formula including velocity of approach is

in USCS units $$Q' = 3.367L(H + 1.5h)^{3/2} \quad (6)$$

in SI units $$Q' = 1.858L(H + 1.5h)^{3/2}$$

where Q = discharge neglecting velocity of approach, ft³/s (m³/s)
Q' = discharge considering velocity of approach, ft³/s (m³/s)
H = head on weir, ft (m)
L = length of weir, ft (m)
h = head due to velocity of approach ($v^2/2g$), ft (m)

The accuracy of measurements obtained with Cipolletti weirs and these formulas is inherently not as great as that obtained with suppressed rectangular or V-notch weirs.[6] It is, however, acceptable where no great precision is required.

There are several well-known formulas used to compute the discharge over 90° V-notch weirs. The most commonly used in the field of irrigation are the Cone formula and the Thomson formula. The Cone formula, considered by authorities to be more reliable for small weirs and for conditions generally encountered in measuring water for open channels, is

in USCS units $$Q = 2.49H^{2.48}$$

in SI units $$Q = 1.34H^{2.48} \quad (7)$$

where Q = discharge over weir, ft³/s (m³/s)
H = head on weir, ft (m)

Ordinarily V-notch weirs are not appreciably affected by velocity of approach. If the weir is installed with complete contraction, the velocity of approach will be low.

Flumes have a measuring section that is produced by contraction of the channel sidewalls or by raising of the bottom to form a hump, or by both. The Parshall flume[7] is the most common and best known measuring flume, especially in irrigation canals. It is a specially shaped open-channel flow section that may be installed in a canal, lateral, or ditch to measure the rate of flow of water. The flume has four significant advantages: (1) it can operate with relatively small head loss, (2) it is relatively insensitive to velocity of approach, (3) it has the capability of making good measurements with no submergence, moderate submergence, or even with considerable submergence downstream, and (4) its velocity of flow is sufficiently high to virtually eliminate sediment deposition in the structure during operation.

Discharge through a Parshall flume can occur for two conditions of flow. The first, free flow, occurs when there is insufficient backwater depth to reduce the discharge rate. The second, submerged flow, occurs when the water surface downstream from the flume is far enough above the elevation of the flume crest to reduce the discharge. For free flow, only the flume head H_a at the upstream gage location is needed to determine the discharge from a standard table. The free-flow range includes some of the range that might ordinarily be considered submerged flow because Parshall flumes tolerate 50 to 80% submergence before the free-flow rate is measurably reduced. For submerged flows (when submergence is greater than 50 to 80%, depending upon flume size), both the upstream and downstream heads H_a and H_b are needed to determine the discharge (Figure 16).

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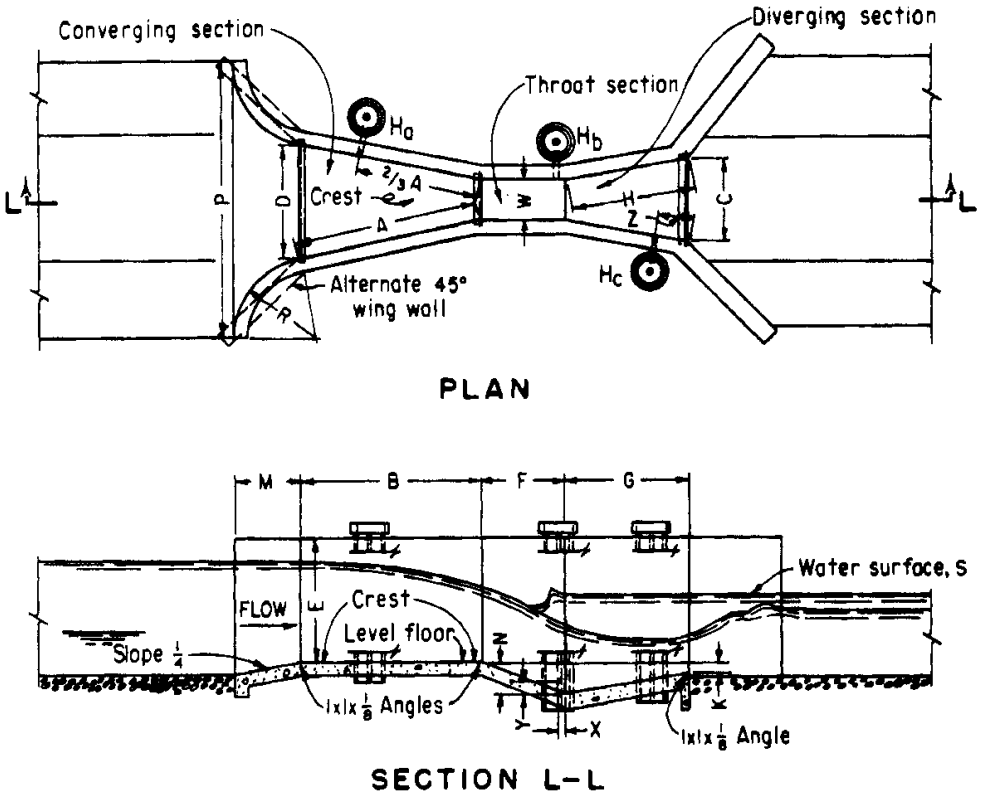


FIGURE 16A Parshall flume dimensions (sheet 1 of 2)

A distinct advantage of the Parshall flume is its ability to function as a flowmeter over a wide operating range with minimum head loss while requiring but a single head measurement for each discharge. The head loss is only about one-fourth of that needed to operate a weir having the same crest length. Another advantage is that the velocity of approach is automatically controlled if the correct size of flume is chosen and if the flume is used as it should be, that is, as an in-line structure.

Flumes are widely used because there is no easy way to alter the dimensions of flumes that have been constructed or to change the device or channel to obtain an unfair proportion of water.

The main disadvantages of Parshall flumes are (1) they cannot be used in close-coupled combination structures consisting of turnout, control, and measuring device; (2) they are usually more expensive than weirs or submerged orifices; (3) they require a solid, watertight foundation; and (4) they require accurate workmanship for satisfactory construction and performance.

Parshall flume sizes are designated by the throat width W, and sizes range from 1 in (25.4 mm) for discharges as small as 0.01 ft^3/s (2.83×10^{-2} m^3/s) up to 50 ft (15 m) for discharges as large as 3000 ft^3/s (85 m^3/s).[8] Flumes may be built of wood, concrete, galvanized sheet metal, or other desired materials. Large flumes are usually constructed on the site, but smaller flumes may be purchased as prefabricated structures to be installed in one piece. Some flumes are available as lightweight shells, which are made rigid and immobile by placing concrete outside the walls and beneath the bottom. Larger flumes are used in rivers and large canals and streams; smaller ones are used for measuring farm deliveries or for row requirements in the farmer's field.

Flumes can operate in two modes: free flow and submerged flow. In free flow, the discharge depends solely upon the width of the throat W and the depth of water H_a at the gage point in the converging section (Figures 16 and 17). Free-flow conditions in the flume are similar to those that occur at a weir or spillway crest in that water passing over the crest is not slowed by downstream conditions.

In submerged flow, other factors are operative. In most installations, when the discharge is increased above a critical value, the resistance to flow in the downstream

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	W		A		2/3 A		B		C		D		E		F		G		H		K		M		N		P		R		X		Y		Z		FREE-FLOW CAPACITY	
																																					MINIMUM	MAXIMUM
	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	FT.	IN.	SEC - FT.	SEC. - FT.
[2]	0	1 [1]	1	$2\frac{9}{32}$	0	$9\frac{17}{32}$	1	2	0	$3\frac{21}{32}$	0	$6\frac{19}{32}$	0	6 TO 9	0	3	0	8	0	$8\frac{1}{8}$	0	$\frac{3}{4}$	–	–	0	$1\frac{1}{8}$		–	–		0	$\frac{5}{16}$	0	$\frac{1}{2}$	0	$\frac{1}{8}$	.01	0.19
		2 [1]	1	$4\frac{5}{16}$		$10\frac{7}{8}$	1	4		$5\frac{5}{16}$		$8\frac{13}{32}$		6 TO 10		$4\frac{1}{2}$		10		$10\frac{1}{8}$		$\frac{7}{8}$		–		$1\frac{11}{16}$		–	–			$\frac{5}{8}$		1		$\frac{1}{4}$	.02	.47
		3 [1]	1	$6\frac{3}{8}$	1	$\frac{1}{4}$	1	6		7		$10\frac{3}{16}$	1 TO 1 1/2			6	1	0	1	$\frac{5}{32}$		1	–			$2\frac{1}{4}$		–	–			1		$1\frac{1}{2}$		$\frac{1}{2}$	.03	1.13
[3]	0	6	2	$\frac{7}{16}$	1	$4\frac{5}{16}$	2	0	1	$3\frac{1}{2}$	1	$3\frac{5}{8}$	2	0	1	0	2	0	–		0	3	1	0	0	$4\frac{1}{2}$	2	$11\frac{1}{2}$	1	4	0	2	0	3	–		.05	3.9
		9	2	$10\frac{5}{8}$	1	$11\frac{1}{8}$	2	10	1	3	1	$10\frac{5}{8}$	2	6	1	0	1	6	–			3	1	0		$4\frac{1}{2}$	3	$6\frac{1}{2}$	1	4		2		3	–		.09	8.9
	1	0	4	6	3	0	4	$4\frac{7}{8}$	2	0	2	$9\frac{1}{4}$	3	0	2	0	3	0	–			3	1	3		9	4	$10\frac{3}{4}$	1	8		2		3	–		.11	16.1
	1	6	4	9	3	2	4	$7\frac{7}{8}$	2	6	3	$4\frac{3}{8}$	3	0	2	0	3	0	–			3	1	3		9	5	6	1	8		2		3	–		.15	24.6
	2	0	5	0	3	4	4	$10\frac{7}{8}$	3	0	3	$11\frac{1}{2}$	3	0	2	0	3	0	–			3	1	3		9	6	1	1	8		2		3	–		.42	33.1
[3]	3	0	5	6	3	8	5	$4\frac{3}{4}$	4	0	5	$1\frac{7}{8}$	3	0	2	0	3	0	–			3	1	3		9	7	$3\frac{1}{2}$	1	8		2		3	–		.61	50.4
	4	0	6	0	4	0	5	$10\frac{5}{8}$	5	0	6	$4\frac{1}{4}$	3	0	2	0	3	0	–			3	1	6		9	8	$10\frac{3}{4}$	2	0		2		3	–		1.3	67.9
	5	0	6	6	4	4	6	$4\frac{1}{2}$	6	0	7	$6\frac{5}{8}$	3	0	2	0	3	0	–			3	1	6		9	10	$1\frac{1}{4}$	2	0		2		3	–		1.6	85.6
	6	0	7	0	4	8	6	$10\frac{3}{8}$	7	0	8	9	3	0	2	0	3	0	–			3	1	6		9	11	$3\frac{1}{2}$	2	0		2		3	–		2.6	103.5
	7	0	7	6	5	0	7	$4\frac{1}{4}$	8	0	9	$11\frac{3}{8}$	3	0	2	0	3	0	–			3	1	6		9	12	6	2	0		2		3	–		3.0	121.4
	8	0	8	0	5	4	7	$10\frac{1}{8}$	9	0	11	$1\frac{3}{4}$	3	0	2	0	3	0	–			3	1	6		9	13	$8\frac{1}{4}$	2	0		2		3	–		3.5	139.5
[4]	10	0	–		6	0	14	0	12	0	15	$7\frac{1}{4}$	4	0	3	0	6	0	–		0	6	–		1	$1\frac{1}{2}$	–		–		0	9	1	0	–		6	200
	12	0	–		6	8	16	0	14	8	18	$4\frac{3}{4}$	5	0	3	0	8	0	–			6	–		1	$1\frac{1}{2}$	–		–			9	1	0	–		8	350
	15	0	–		7	8	25	0	18	4	25	0	6	0	4	0	10	0	–			9	–		1	6	–		–			9	1	0	–		8	600
	20	0	–		9	4	25	0	24	0	30	0	7	0	6	0	12	0	–		1	0	–		2	3	–		–			9	1	0	–		10	1000
	25	0	–		11	0	25	0	29	4	35	0	7	0	6	0	13	0	–		1	0	–		2	3	–		–			9	1	0	–		15	1200
	30	0	–		12	8	26	0	34	8	40	$4\frac{3}{4}$	7	0	6	0	14	0	–		1	0	–		2	3	–		–			9	1	0	–		15	1500
	40	0	–		16	0	27	0	45	4	50	$9\frac{1}{2}$	7	0	6	0	16	0	–		1	0	–		2	3	–		–			9	1	0	–		20	2000
	50	0	–		19	4	27	0	56	8	60	$9\frac{1}{2}$	7	0	6	0	20	0	–		1	0	–		2	3	–		–			9	1	0	–		25	3000

[1] Tolerance on throat width (w) $\pm \frac{1}{64}$ inch; tolerance on other dimensions $\pm \frac{1}{32}$ inch. Sidewalls of throat must be parallel and vertical.

[2] From Colorado State University Technical Bulletin No. 61.

[3] From U.S. Department of Agriculture Soil Conservation Circular No. 843.

[4] From Colorado State University Bulletin No. 426-A.

FIGURE 16B Parshall flume dimensions (sheet 2 of 2.) (1 ft = 0.3048 m; ft^3/s = 0.0253 m^3/s)

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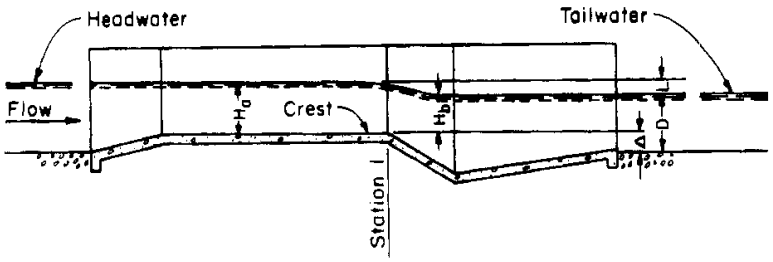


FIGURE 17 Relationships of flow depth to flume crest elevation

channel becomes sufficient to reduce the velocity, increase the flow depth, and cause a backwater effect at the Parshall flume. It might be expected that the discharge would begin to be reduced as soon as the backwater level H_b exceeds the elevation of the flume crest; however, this is not the case. Calibration tests show that the discharge is not reduced until the submergence ratio H_b/H_a (expressed in percent) exceeds the following values:

50% for flumes 1, 2, and 3 in (25, 50, and 75 mm) wide
60% for flumes 6 and 9 in (152 and 229 mm) wide
70% for flumes 1 to 8 ft (0.3 to 2.4 m) wide
80% for flumes 8 to 50 ft (2.4 to 15.2 m) wide

The discharge equations for free flow over flumes are as follows. The equation which expresses the relationship between upstream head H_a and discharge Q for widths W from 1 to 8 ft (0.3 to 2.4 m) is

in USCS units $$Q = 4WH_a^{1.522W^{0.026}}$$

in SI units $$Q = 0.371W\left(\frac{Ha}{0.305}\right)^{1.57W^{0.026}}$$

If this equation is used to compute discharges through flumes ranging from 10 to 50 ft (3 to 15.7 m) wide, the computed discharges are always larger than actual discharges. Therefore, a more accurate equation was developed for the large flumes:

In USCS units $$Q = (3.6875W + 2.5)H_a^{1.6}$$

In SI units $$Q = (2.293W + 0.474)H_a^{1.6}$$

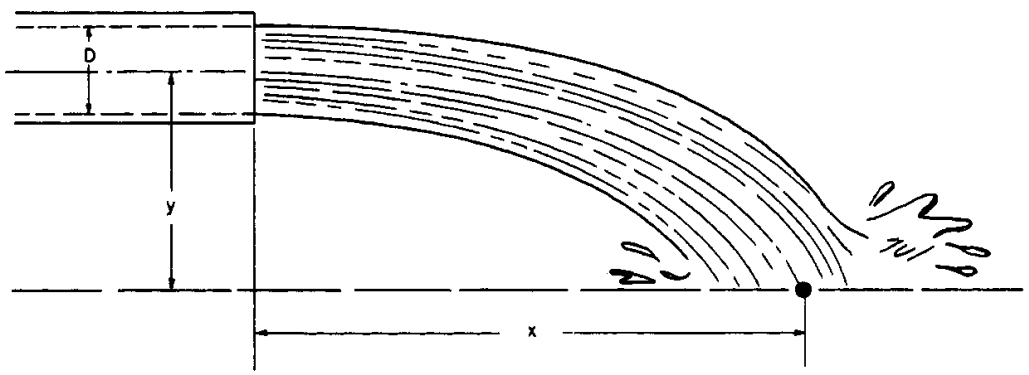
This difference in computed discharges obtained by using the two equations for an 8-ft (2.4-m) flume is normally less than 1%; however, the difference becomes greater as the flume size increases. Because of the difficulties in regularly using these equations, discharge tables have been prepared for use with flumes 1 to 50 ft (0.3 to 15.2 m) wide.

CURRENT METERS The essential features of a conventional current meter are a wheel that rotates when immersed in flowing water and a device for determining the number of revolutions of the wheel. For open-channel flow measurement, a type generally used is the Price meter with five or six conical cups. The relationship between the velocity of the water and the number of revolutions of the wheel per unit time is determined experimentally for each instrument for various velocities. Also, the operator must have considerable skill to obtain consistent satisfactory results.

A detailed explanation of the use of current meters is contained in Chap. 5 of Reference 9.

OTHER METHODS The discharge measurement methods previously given are the ones in common use; however, a number of other methods, some newer and more sophisticated,

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$$\text{CAPACITY, GPM} = \frac{2.45\,D^2\,x}{\sqrt{\frac{2y}{32.16}}} \qquad \text{CAPACITY, m}^3\text{/h} = \frac{0.00283\,D^2\,x}{\sqrt{\frac{2y}{9.81}}}$$

Where as
D = Pipe Dia., Inches (mm)
x = Hor. Dist., Feet (m)
y = Vert. Dist., Feet (m)

FIGURE 18 Approximating flow from a horizontal pipe

are well established. The use of these special methods is acceptable provided their limitations are recognized and all parties to the testing program are in agreement on their use. A few of these methods, not in any particular order, are

Salt velocity	Deflection meters	Radioisotope	Color dilution
Salt dilution	Propeller meters	Gates and sluices	Slope area
Color velocity	Float movement	Acoustic flowmeters	Magnetic meters

FIELD APPROXIMATING Often a field approximation of water flow from a pump discharge becomes necessary, especially if no other methods are practical or readily available. One of the accepted methods is by trajectory. The discharge from the pipe may be either vertical or horizontal, the principal difficulty being in measuring the coordinates of the flowing stream accurately. The pipes must be flowing full, and the accuracy of this method varies from 85 to 100%. Figure 18 illustrates the approximation from a horizontal pipe. This method can be further simplified by measuring to the top of the flowing stream and always measuring so y equals 12 in (300 mm) and measuring the horizontal distance X in inches (millimeters), as illustrated in Figure 19.

Figure 20 illustrates a method of measuring discharge from a vertical pipe.

Head Measurements Head is the quantity used to express the energy content of a liquid per unit weight of the liquid referred to any arbitrary datum. In terms of foot-pounds of energy per pound of liquid, all head quantities have the dimensions of feet of liquid. The unit for measuring head is the foot (meter). The relation between a pressure expressed in pounds per square inch (kilopascals) and one expressed in feet (meters) of head is

in USCS units $$\text{Head, ft} = \text{lb/in}^2 \times \frac{2.31 \times 62.3}{W} = \text{lb/in}^2 \times \frac{2.31}{\text{sp. gr.}}$$

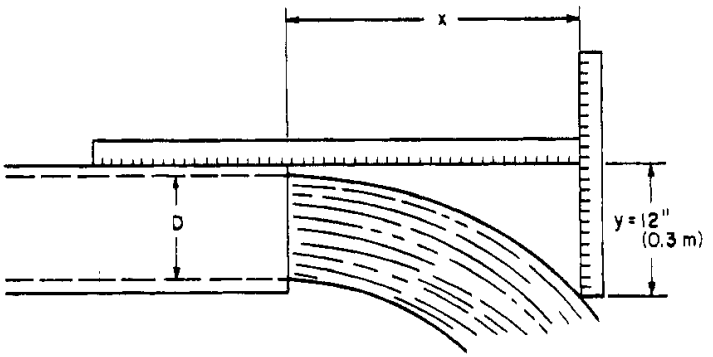
in SI units $$\text{Head, m} = \frac{0.102 \times \text{kPa}}{W}$$

where = specific weight (mass), lb/ft^3 (kg/l)

sp. gr. = specific gravity of the liquid

The following exerpt from the *Hydraulic Institute Standard* is used by the Bureau of Reclamation throughout its test program:

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CAPACITY, GPM $= 0.818\ D^2X$

CAPACITY, $m^3/h = 1.1336 \times 10^{-6} D^2X$

D = in (mm)

X = in (mm)

APPROXIMATE CAPACITY, GPM,
FOR FULL FLOWING HORIZONTAL PIPES

STD. WT. STEEL PIPE, INSIDE DIA., IN.		DISTANCE X, IN., WHEN Y = 12"										
NOMINAL	ACTUAL	12	14	16	18	20	22	24	26	28	30	32
2	2.067	42	49	56	63	70	77	84	91	98	105	112
2½	2.469	60	70	80	90	100	110	120	130	140	150	160
3	3.068	93	108	123	139	154	169	185	200	216	231	246
4	4.026	159	186	212	239	266	292	318	345	372	398	425
5	5.047	250	292	334	376	417	459	501	543	585	627	868
6	6.065	362	422	482	542	602	662	722	782	842	902	962
8	7.981	627	732	837	942	1047	1150	1255	1360	1465	1570	1675
10	10.020	980	1145	1310	1475	1635	1800	1965	2130	2290	2455	2620
12	12.000	1415	1650	1890	2125	2360	2595	2830	3065	3300	3540	3775

FIGURE 19 Simplified method for approximating flow from a horizontal pipe (1 in = 25.4 mm; 1 gpm = 0.227 m^3/h)

It is important that steady flow conditions exist at the point of instrument connection. For this reason, it is necessary that pressure or head measurement be taken on a section of pipe where the cross section is constant and straight. Five to ten diameters of straight pipe of unvarying cross section following any elbow or curved member, valve, or other obstruction, are necessary to insure steady flow conditions.

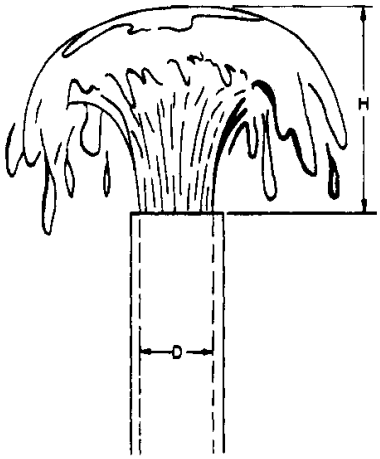
The following precautions should be taken in forming orifices for pressure-measuring instruments and for making connections. The orifice in the pipe should be flush with and normal to the wall of the water passage. The wall of the water passage should be smooth and of unvarying cross section. For a distance of at least 12 in (0.8 m) preceding the orifice, all tubercles and roughness should be removed with a file or emery cloth, if necessary. The orifice should be from $\frac{1}{8}$ to $\frac{1}{4}$ in (3–6 mm) in diameter and two diameters long.

The edges of the orifice should be provided with a suitable radius tangential to the wall of the water passage and should be free from burrs or irregularities. Two pressure tap arrangements shown in Figure 27 indicate taps in conformity with the above. Where more than one tap is required at a given measuring section, separate properly valved connections should be made. As an alternative, separate instruments should be provided.

Multiple orifices should not be connected to one instrument except on those metering devices, such as venturi meters, where proper calibrations have been made.

All leads from the orifice should be tight and as short and direct as possible. For dry-tube leads, suitable drain pots should be provided and a loop of sufficient height to keep

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Capacity, Gpm = $5.68\ KD^2\ H1/2$
Capacity, $m^3/h = 7.872 \times 10^{-5} KD^2\ H1/2$
D = I.D. of Pipe, in. (mm)
H = Vertical Height of water jets, in. (mm)
K = a constant, varying from .87 to .97 for pipes 2 to 6 in. (51 to 152 mm)
H = 6 to 24 in. (152 to 610 mm)

APPROXIMATE CAPACITY, GPM, FOR FLOW FROM VERTICAL PIPES

NOMINAL	VERTICAL HEIGHT, H, OF WATER JET, IN.										
I.D. PIPE, IN.	3	3.5	4	4.5	5	5.5	6	7	8	10	12
2	38	41	44	47	50	53	56	61	65	74	82
3	81	89	96	103	109	114	120	132	141	160	177
4	137	151	163	174	185	195	205	222	240	269	299
6	318	349	378	405	430	455	480	520	560	635	700
8	567	623	684	730	776	821	868	945	1020	1150	1270
10	950	1055	1115	1200	1280	1350	1415	1530	1640	1840	2010

FIGURE 20 Approximating flow from a vertical pipe (1 in = 25.4 mm; 1 gpm = 0.227 m^3/h)

the pumped liquid from entering the leads should be formed. For wet-tube leads, vent cocks for flushing should be provided at any high point or loop crest to assure that tubes do not become airbound. All instrument hose, piping, and fittings should be checked under pressure prior to test to assure that there are no leaks. Suitable damping devices may be used in the leads.

If these conditions cannot be satisfied at the point of measurement, it is recommended that four separate pressure taps be installed, equally spaced about the pipe, and that the pressure at that section be taken as the average of the four separate values. If the separate readings show a difference of static pressure that might affect the head beyond the contract tolerances, the installation should be corrected or an acceptable tolerance determined.

Figures 21 to 27 show suitable arrangements for various types of instruments and formulas for translating instrument readings into feet (meters) of liquid pumped, for expressing instrument head as elevation over a common datum, and for correcting these formulas for the velocity head existing in the suction and discharge pipes.

The datum is taken as the centerline of the pump for horizontal-shaft pumps and as the entrance eye of the impeller for vertical-shaft pumps.

The instruments, when practicable, are water columns or manometers for normal pressures and mercury manometers, bourdon gages, electric pressure transducers, or deadweight gage testers for high pressure. When water columns are used, care should be taken to avoid errors due to the difference between the temperature of the water in the gage and that of the water in the pump.

DEFINITIONS AND SYMBOLS The symbols used throughout this section for expressing and computing head are those used by the ASME Power Test Code[1] for pumps and the

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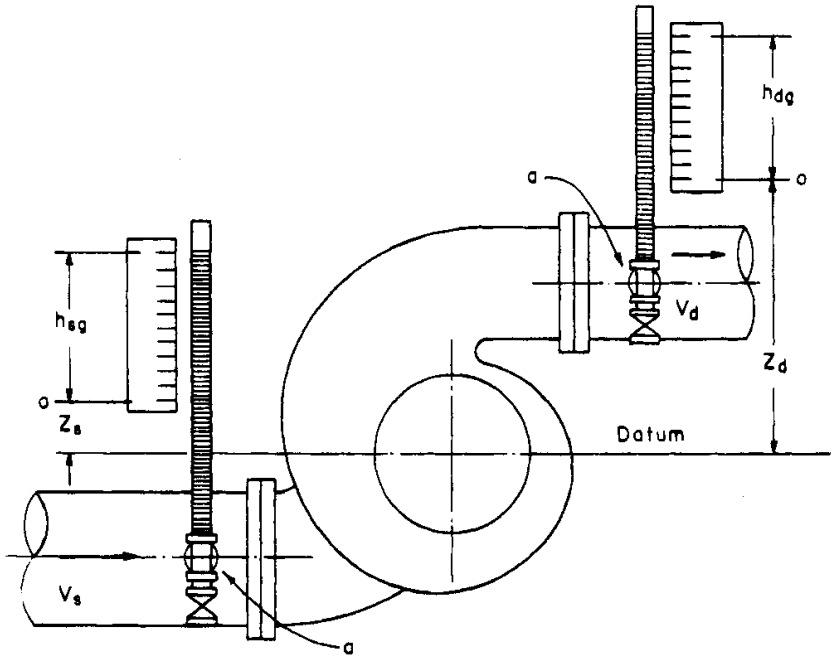


FIGURE 21 Head measurement

Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 13). The following symbols apply to Figures 21 to 27 where temperature effects are negligible:

H = total head or dynamic head, ft (m) = the energy increase per pound (kilogram) of liquid imparted to the liquid by the pump and therefore the algebraic difference between total discharge head and total suction head ($H = h_d - h_s$); h_d and h_s are negative if the corresponding values at the datum elevation are below atmospheric pressure

h_{gd} = discharge gage reading, ft (m) H_2O

h_{gs} = suction gage reading, ft (m) H_2O

$$h_d = +h_{gd} + Z_d + \frac{V_d^2}{2g}$$

$$h_s = +h_{gs} + Z_s + \frac{V_s^2}{2g}$$

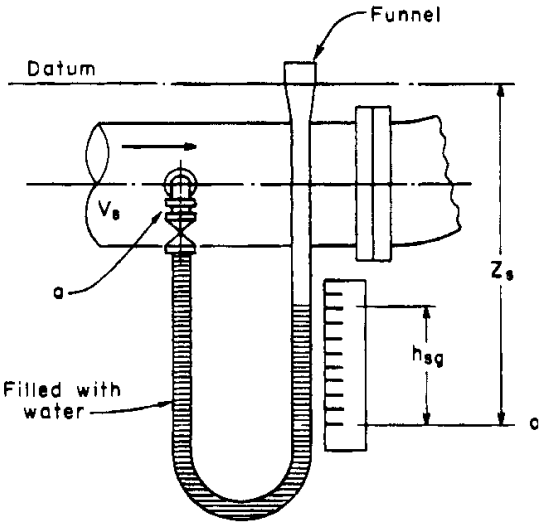


FIGURE 22 Head measurement

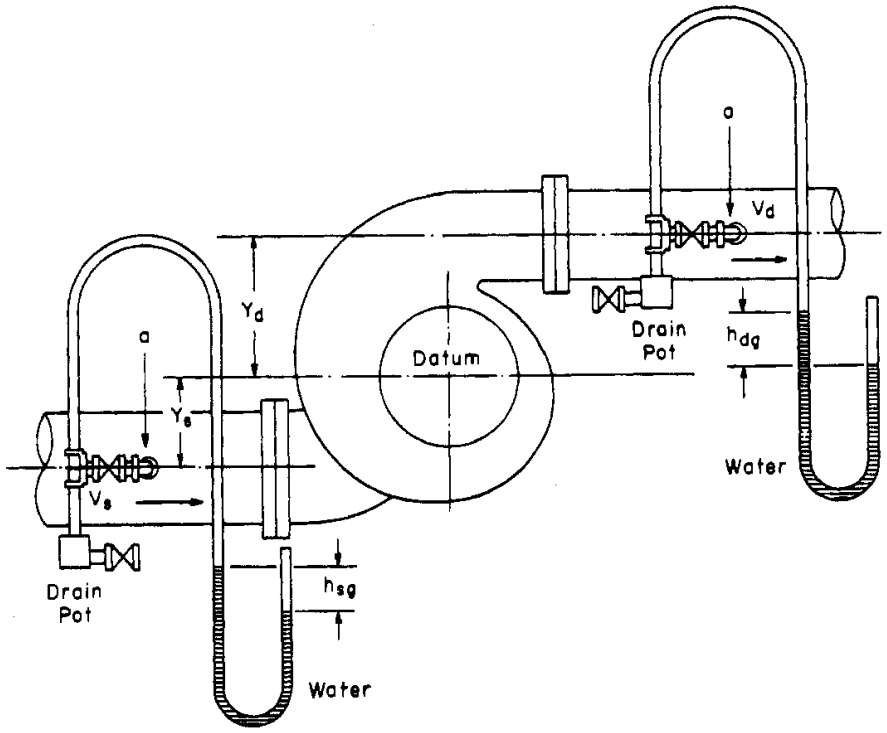


FIGURE 23 Head measurement

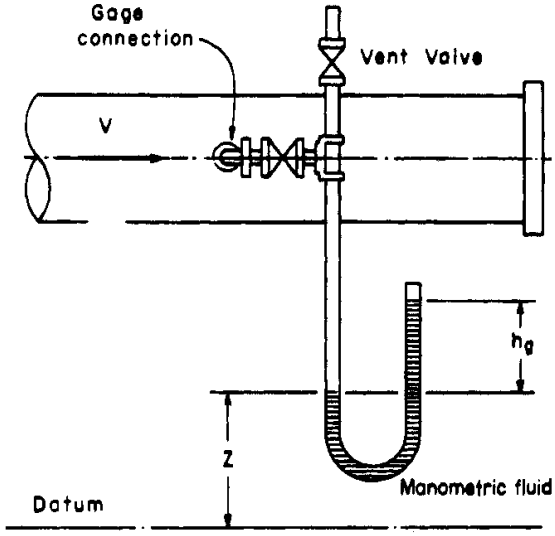


FIGURE 24 Head measurement

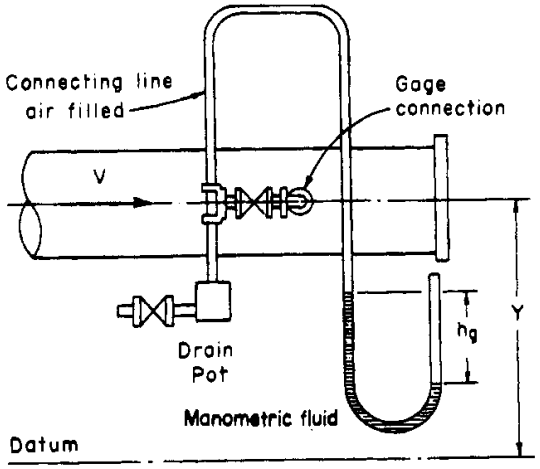


FIGURE 25 Head measurement

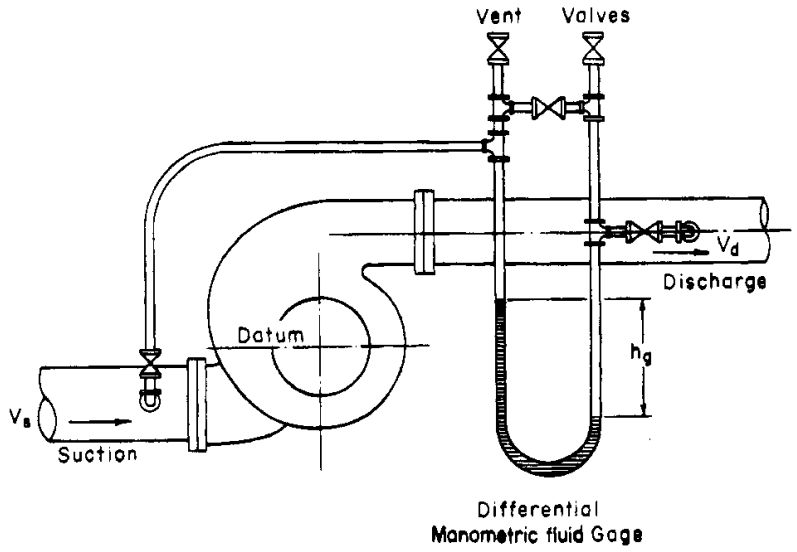


FIGURE 26 Head measurement

Both the discharge gage and the suction gage can be direct-reading water manometers, converted mercury manometers, or calibrated bourdon pressure gages.

Z_d = elevation of discharge gage, zero above datum elevation, ft (m)

Z_s = elevation of suction gage, zero above datum elevation, ft (m)

The quantities Z_d and Z_s are negative if the gage zero is below the datum elevation.

Y_d = elevation of discharge gage connection to discharge pipe above datum elevation, ft (m)

Y_s = elevation of suction gage connection to suction pipe above datum elevation, ft (m)

The quantities Y_d and Y_s are negative if the gage connection to the pipe lies below the datum elevation.

V_d = average water velocity in discharge pipe at discharge gage connection, ft/s (m/s)

V_s = average water velocity in suction pipe at suction gage connection, ft/s (m/s)

h_d = total discharge head above atmospheric pressure at datum elevation, ft (m)

h_s = total suction head above atmospheric pressure at datum elevation, ft (m)

h_{vs} = velocity head in suction pipe ($V_s^2/2g$), ft (m)

h_{vd} = velocity head in discharge pipe ($V_d^2/2g$), ft (m)

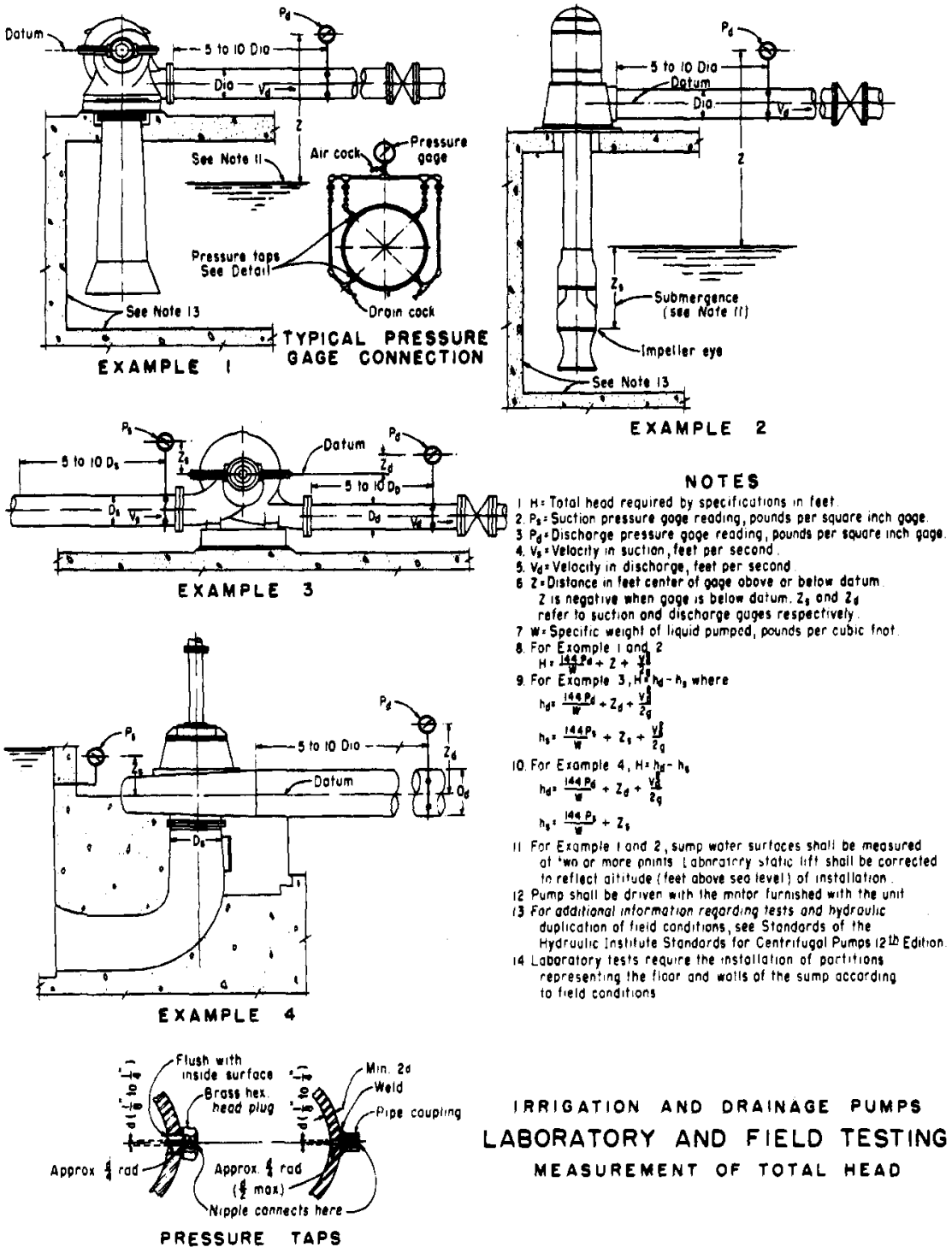
NPSHA = net positive suction head available = total suction head in feet (meters) of liquid absolute, determined at suction nozzle and referred to datum less absolute vapor pressure of the liquid in feet (meters) of liquid pumped ($NPSHA = h_a - H_{vpa} + h_s$)

h_{sa} = total suction head absolute ($h_a + h_s$)

H_{vpa} = vapor pressure of liquid, ft (m) abs

h_a = atmosphere pressure, ft (m) abs

MEASURING HEAD WITH WATER GAGES The following examples illustrate how to calculate the head in a centrifugal pump arrangement with gages either above or below atmospheric pressure.



NOTES

1. H = Total head required by specifications in feet.
2. P_s = Suction pressure gage reading, pounds per square inch gage.
3. P_d = Discharge pressure gage reading, pounds per square inch gage.
4. V_s = Velocity in suction, feet per second.
5. V_d = Velocity in discharge, feet per second.
6. Z = Distance in feet center of gage above or below datum. Z is negative when gage is below datum. Z_s and Z_d refer to suction and discharge gages respectively.
7. W = Specific weight of liquid pumped, pounds per cubic foot.
8. For Example 1 and 2 $H = \frac{144 P_d}{W} + Z + \frac{V_d^2}{2g}$
9. For Example 3, $H = h_{gd} - h_s$ where
   $h_{gd} = \frac{144 P_d}{W} + Z_d + \frac{V_d^2}{2g}$
   $h_s = \frac{144 P_s}{W} + Z_s + \frac{V_s^2}{2g}$
10. For Example 4, $H = h_{gd} - h_s$
    $h_{gd} = \frac{144 P_d}{W} + Z_d + \frac{V_d^2}{2g}$
    $h_s = \frac{144 P_s}{W} + Z_s$
11. For Example 1 and 2, sump water surfaces shall be measured at two or more points. Laboratory static lift shall be corrected to reflect altitude (feet above sea level) of installation.
12. Pump shall be driven with the motor furnished with the unit.
13. For additional information regarding tests and hydraulic duplication of field conditions, see Standards of the Hydraulic Institute Standards for Centrifugal Pumps 12th Edition.
14. Laboratory tests require the installation of partitions representing the floor and walls of the sump according to field conditions.

IRRIGATION AND DRAINAGE PUMPS
LABORATORY AND FIELD TESTING
MEASUREMENT OF TOTAL HEAD

FIGURE 27 Head measurement. In SI units, $hd = [(0.102\,P_d/W) + Z_d + (V_d^2/2g)]$; $h_s = [(0.102\,P_s/W) + Z_s + (V_s^2/2g)]$. (ft × 0.3048 = m; psi × 6.894 = kPa; pcf × .016 = kg/liter)

EXAMPLE The pressure at gage connection a in Figure 21 is above atmospheric pressure, and the line between either the discharge or suction pipe and the corresponding gage is filled completely with water. The following equations apply:

$$h_d = +h_{gd} + Z_d + \frac{V_d^2}{2g}$$

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$$h_s = +h_{gs} + Z_s + \frac{V_s^2}{2g}$$

Note: The word *water* is used to represent the liquid being pumped. The provisions are applicable to the pumping of other liquids, provided the gages and connecting lines contain the liquid being pumped.

EXAMPLE The pressure at gage connection a in Figure 22 is below atmospheric pressure. The following equation applies:

$$h_s = h_{gs} - Z_s + \frac{V_s^2}{2g}$$

The negative sign of Z_s indicates that the gage zero is located below the datum.

EXAMPLE The pressure at gage connection a in Figure 23 is below atmospheric pressure, and the line between either the discharge or suction pipe and the corresponding gage is filled completely with air. The following equations apply:

$$h_d = -h_{gd} + Y_d + \frac{V_d^2}{2g}$$

$$h_s = -h_{gs} - Y_s + \frac{V_s^2}{2g}$$

Note: If a connecting pipe is air-filled, it must be drained before a reading is made. Water cannot be used in the U tube if either h_{dg} or h_{ds} exceeds the height of the rising loop.

MEASURING HEAD WITH MANOMETRIC FLUID GAGES The following examples illustrate the use of manometric fluid gages for measuring head in a centrifugal pump arrangement.

EXAMPLE In Figure 24, the gage pressure is above atmospheric pressure and the connecting line is completely filled with water. The following equation applies:

$$h = \frac{W_m}{W} h_g + Z + \frac{V^2}{2g}$$

where W_m = specific weight (mass) of manometric fluid, lb/ft^3 (kg/liter)
W = specific weight (mass) of liquid pumped, lb/ft^3 (kg/liter)
h_g = suction or discharge gage reading, ft (m) Hg

The quantities h, Z, Y, and V without subscripts apply equally to suction and discharge head measurements.

EXAMPLE The gage pressure in Figure 25 is below atmospheric pressure, and the connecting line is completely filled with air, with a rising loop to prevent water from passing to the manometric fluid column. The following equation applies

$$h = \frac{W_m}{W} h_g = Y + \frac{V^2}{2g}$$

MEASURING HEAD WITH DIFFERENTIAL MANOMETRIC FLUID GAGES Figure 26 indicates a centrifugal pump arrangement in which a differential manometric fluid gage is used to measure head. When this type of gage is used and the connecting lines are completely filled with water, the correct equation is

$$H = \left(\frac{W_m}{W} - 1\right) h_g + \frac{V_d^2}{2g} - \frac{V_s^2}{2g}$$

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In addition to the differential gage, a separate suction gage can be used, as shown in Figures 22 and 25. The equation in this case is

$$h_s = \frac{W_m}{W} h_{gs} - Z + \frac{V_s^2}{2g}$$

MEASURING HEAD WITH BOURDON GAGES AND PRESSURE TRANSDUCERS An example of a centrifugal pump arrangement that uses calibrated bourdon gages for head measurement is shown in Example 3 of Figure 27, with the gage pressure above atmospheric pressure. The distances Z_s and Z_d are measured to the center of the gage and are negative if the center of the gage lies below the datum line.

MEASURING HEAD ON VERTICAL SUCTION PUMPS IN SUMPS AND CHANNELS In vertical-shaft pumps drawing water from large open sumps and having inlet passages whose length does not exceed about three inlet opening diameters, such inlet pieces having been furnished as part of the pump, the total head should be the reading of the discharge connection in feet (meters) plus the vertical distance from the gage centerline to the free water level in the sump in feet (meters) (Example 2 of Figure 27).

Power Measurement The pump input power may be determined with a calibrated motor, a transmission dynamometer, or a torsion dynamometer. The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 13) are generally used as the basis for most power measurement procedures.

CALIBRATED MOTORS When pump input power is to be determined with a calibrated motor, the power input should be measured at the terminals of the motor to exclude any line losses that may occur between the switchboard and the driver. Certified calibration curves of the motor must be obtained. The calibration should be conducted on the motor in question and not on a similar machine. Such calibrations must indicate the true input-output value of motor efficiency and not some conventional method of determining an arbitrary efficiency. Calibrated laboratory-type electric meters and transformers should be used to measure power input to all motors.

TRANSMISSION DYNAMOMETERS The transmission, or torque-reaction, dynamometer consists of a cradled electric motor with its frame and field windings on one set of bearings and the rotating element on another set, so the frame is free to rotate but is restrained by means of some weighting or measuring device.

When pump input power is to be determined with a transmission dynamometer, the unloaded and unlocked dynamometer must be properly balanced prior to the test at the same speed at which the test is to be run. The balance should be checked against standard weights. After the test the balance must be rechecked to assure that no change has taken place. In the event of an appreciable change, the test should be rerun. An accurate measurement of speed is essential and should not vary from the pump rated speed by more than 1%. Power input is calculated as shown later in this section under "Computations."

TORSION DYNAMOMETERS The torsion dynamometer consists of a length of shafting whose torsional strain when rotating at a given speed and delivering a given torque is measured by some standard method. When pump input power is to be determined with a torsion dynamometer, the unloaded dynamometer should be statically calibrated prior to the test. This is done by restraining one end of the dynamometer shaft and applying a known force at a known radius. Loads should be applied in intervals of 10% of the dynamometer's full scale rating from zero to 100% of full scale. If the dynamometer is to be used in an overload condition, the calibration range should be increased to include the overload force. When the force is removed, the dynamometer must return to within 0.25% of zero. The no-load (zero) output of the dynamometer should be recorded before and after the test. The temperature of the torsion dynamometer during the test runs must be within 10°F (6°C) of the temperature when the dynamic calibrations were made unless the dynamometer is temperature compensated. All torsion dynamometer calibrations should be witnessed and

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approved by all parties to the test. In the event of a variation greater than allowed, a rerun of the test must be made. Power input calculations are shown in the following text.

Speed Measurement The speed of the pump under test is determined by one of the following methods:

Revolution counter (manual or automatic)
Tachometer
Stroboscopic device
Electronic counter

In all cases, the instruments used must be carefully calibrated before the test to demonstrate that they will produce the required speed readout to within the desired accuracy. Accepted accuracy is usually ±0.1%. Should cyclic speed change result in power fluctuations, at least five equally spaced, timed readings should be taken to give a satisfactory mean speed.

COMPUTATIONS

Pump Power

POWER OUTPUT The water power, or useful work, done by the pump is found by the formula

in USCS units $$\text{whp} = \frac{\text{lb of liquid pumped/min} \times \text{total head in ft of liquid}}{33{,}000}$$

in SI units $$\text{wkW} = \frac{\text{kg of liquid pumped/min} \times \text{total head in m of liquid}}{6131}$$

If the liquid has a specific gravity of 1 and the specific weight of the liquid is 62.3 lb/ft^3 (1.0 kg/liter) at 68°F (20°C), the formula is

in USCS units $$\text{whp} = \frac{\text{gpm} \times \text{head in ft}}{3960}$$

in SI units $$\text{wkW} = 9.8\ \text{m}^3/\text{h} \times \text{head in m}$$

POWER INPUT The brake power required to drive the pump is found by the formula

in USCS units $$\text{bhp} = \frac{\text{gpm} \times \text{total head in ft}}{3960 \times \text{pump efficiency}}$$

in SI units $$\text{bkW} = \frac{9.8\ \text{m}^3/\text{h} \times \text{total head in m}}{\text{pump efficiency}}$$

where pump efficiency is obtained from the formula

$$\text{Pump efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{whp}}{\text{bhp}} = \frac{\text{wkW}}{\text{bkW}}$$

(used as a decimal)

The electric power input to the motor is

in USCS units $$\text{ehp} = \frac{\text{bhp}}{\text{motor efficiency}} = \frac{\text{gpm} \times \text{head in ft}}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

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in SI units $$\text{kW} = \frac{9.8\ \text{m}^3/\text{h} \times \text{head in m}}{\text{pump efficiency} \times \text{motor efficiency}}$$

The kilowatt input to the motor is

$$\text{kW input} = \frac{\text{bhp} \times 0.746}{\text{motor efficiency}}$$

$$= \frac{\text{gpm} \times \text{head} \times 0.746}{3960 \times \text{pump efficiency} \times \text{motor efficiency}}$$

Pump Efficiency The pump efficiency is

$$\text{Pump efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{whp}}{\text{bhp}} = \frac{\text{wkW}}{\text{bkW}}$$

For an electric-motor-driven pumping unit, the overall efficiency is

$$\text{Overall efficiency} = \text{pump efficiency} \times \text{motor efficiency}$$

In many specifications, it is required that the actual job motor be used to drive the pump during shop or field testing. Using this test setup, the overall efficiency then becomes what is commonly called "wire-to-water" efficiency, which is expressed by the formula

$$\text{Overall efficiency} = \frac{\text{water power}}{\text{electric power input}} = \frac{\text{whp}}{\text{ehp}} = \frac{\text{wkW}}{\text{kW}}$$

Speed Adjustments The best and standard practice in testing pump speed is to use the actual job motor to drive the pump under test. However, for purposes of plotting test results, it becomes necessary to correct the test values at test speed to rated pump speed.

To adjust pump flow rate, head, power, and NPSH from the values recorded during test to another speed, the following formulas[10] should be used:

Capacity: $$Q_2 = \frac{N_2}{N_1} \times Q_1$$

where Q_1 = flow rate at test speed, gpm (m^3/h)
Q_2 = flow rate at rated speed, gpm (m^3/h)
N_1 = test speed, rpm
N_5 = rated speed, rpm

Head: $$H_2 = \left(\frac{N_2}{N_1}\right)^2 \times H_1$$

where H_1 = head at test speed, ft (m)
H_2 = head at rated speed, ft (m)

Power:

in USCS units $$\text{hp}_2 = \left(\frac{N_2}{N_1}\right)^3 \times \text{hp}_1$$

in SI units $$\text{kW}_2 = \left(\frac{N_2}{N_1}\right)^3 \times \text{kW}_1$$

where hp_1 = power at test speed, hp
hp_2 = power at rated speed, hp

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kW_1 = power at test speed, kW
kW_2 = power at rated speed, kW

Net positive suction head: $$NPSH_2 = \left(\frac{N_2}{N_1}\right)^2 \times NPSH_1$$

where $NPSH_1$ = net positive suction head at test speed, ft (m)
$NPSH_2$ = net positive suction head at rated speed, ft (m)

Shop or field testing at either reduced or increased speed should be permitted only when absolutely no alternatives are available. It is recommended, if reduced or increased speed tests are used as official performance tests, that the specifications state the test head and speed and that the performance warranties be based on the specified head and speed conditions.

If reduced or increased speed tests are considered, certain affinity laws must be observed to maintain hydraulic similarity between actual and test conditions. These affinity law relationships can be expressed by

$$\frac{Q_1}{Q} = \frac{N_1}{N} = \left(\frac{H_1}{H}\right)^{1/2}$$

where test = Q_1 = flow rate and H_1 = head at N_1 = rpm
actual = Q = flow rate and H = head at N = rpm

RECORDS

Data There probably are as many test data forms as there are test laboratories. Each may or may not have an advantage for its particular application. A form for recording pump performance data is shown in Figure 28.

The manufacturer's serial number, type, and size or other means of identification of each pump and driver involved in the test should be carefully recorded in order that mistakes in identity may be avoided. The dimensions and physical conditions not only of the machine tested but of all associated parts of the plant which have any important bearing on the outcome of a test, should be noted.

Normal practice suggests that one test run be either at or as near as possible to the rated condition and that at least three runs be within the specified range of heads.

Automatic Data Collection Computerized data collection is possible and desirable as long as the required test accuracy can be demonstrated. Multiple sets of data should be recorded at each point to prove that steady state conditions exist. Before and after test, zero (speed) conditions should also be recorded.

Plotting Test Results In plotting curves from the test results, it should be kept in mind that any one point may be in error but that all the points should establish a trend.

Unless some external factor is introduced to cause an abrupt change, a smooth curve can be drawn for the points plotted, not necessarily through each and every one. Figure 29 is a graphic representation showing the determination of pump performance with the total head, power input, and efficiency in percent, all plotted on the same graph with the capacity as the abscissa.

Reports In some instances, a preliminary report may be issued as part of a contract agreement. However, normally a final report is all that is required.

On shop tests of relatively small pumps, the test log and plotted results constitute the entire report. Reports get progressively more involved as pumps become larger. Where model

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RECORD OF PUMP PERFORMANCE TEST

DATE OF TEST_____ TEST NO._____ CUSTOMER_____ CUST. ORDER NO._____
MANUFACTURER'S ORDER NO._____ PLANT_____ UNIT NO._____
PROJECT_____
RATED CONDITIONS:
CAPACITY, G.P.M. (m^3/h)_____ TOTAL HEAD, FEET (m)_____ R.P.M._____
OVERALL EFFICIENCY PERCENT_____ RANGE OF HEAD_____
DRIVER:
TYPE_____ HORSEPOWER (kW)_____
MANUFACTURER_____ SERIAL NO._____ TEST VOLTAGE_____
TEST EQUIPMENT:
DISCHARGE MEASUREMENT METHOD_____ CONVERSION FACTOR_____
DISCHARGE GAGE_____ CORRECTION_____ SUCTION GAGE_____ CORRECTION_____
DIFFERENTIAL BETWEEN GAGES_____ INSIDE DIAMETER SUCTION_____ INSIDE DIAMETER DISCHARGE_____
PUMP DATA:
TYPE OF PUMP_____ SIZE_____ NO. STAGES_____
MANUFACTURER_____ SERIAL NO._____ SUCTION SIZE_____ DISCHARGE SIZE_____

	RUN NO.	1	2	3	4	5	6	7	8	9	10
HEAD	PRESSURE, P.S.I. (kPa)										
	HEAD, FEET (m)										
	GAGE ℄ TO WATER LEVEL, FEET (m)										
	VELOCITY HEAD, FEET (m)										
	TOTAL HEAD, FEET (m)										
CAPACITY	READING										
	CONVERSION										
	FLOW, G.P.M. (m^3/h)										
POWER DATA	MOTOR VOLTAGE										
	AMPERES										
	KILOWATTS										
	TOTAL HORSEPOWER INPUT										
	MOTOR EFFICIENCY, %										
	SPEED, R.P.M.										
	DYNAMOMETER										
	BRAKE HORSEPOWER (kW)										
	WATER HORSEPOWER (kW)										
	PUMP EFFICIENCY, %										
	OVERALL EFFICIENCY, %										

TESTED BY_____ WITNESSED BY_____
TYPE OF TEST_____
(FIELD OR SHOP)
REMARKS:

FIGURE 28 Record of pump performance test

testing is used, the final report is a complete record of the agreements, inspections, personnel, calibration data, sample computations, tabulations, descriptions, discussions, etc.

MODEL TESTING

Models are tested for one or more of the following purposes:

1. To develop new ideas and new designs
2. To give the manufacturer a range of warranties for performance and efficiency

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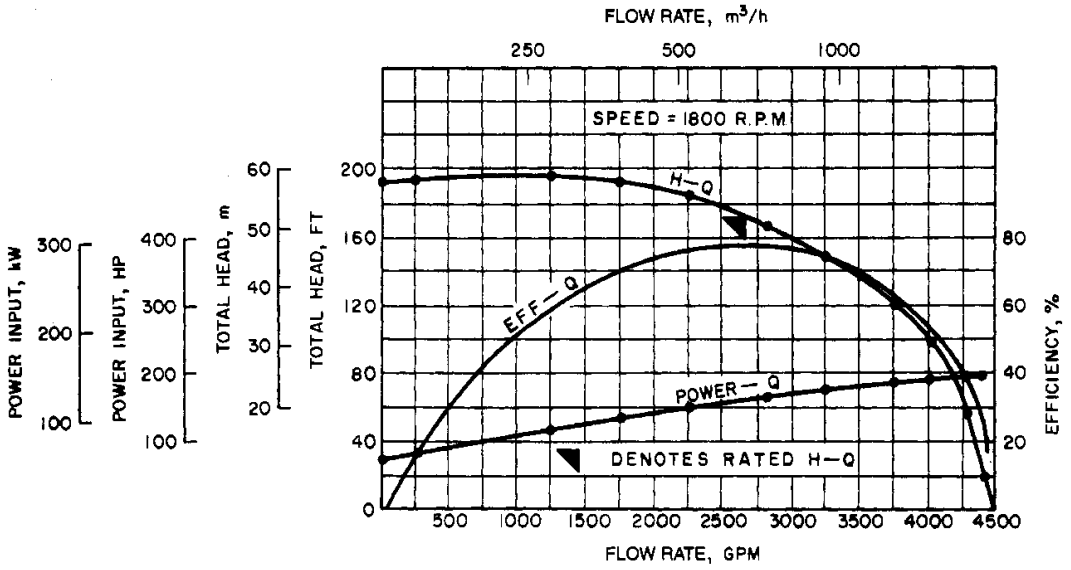


FIGURE 29 Plotted pump performance

3. To give the buyer some assurance that requirements are being met
4. To replace or supplement field testing of a prototype
5. To compare performances of several models

Model testing in advance of final design and installation of a large unit not only provides advance assurance of satisfactory performance but allows for alterations in time for incorporation into the prototype.

Testing Procedures The model should have complete geometric similarity to the prototype in all wetted parts between the intake and discharge sections of the pump. The model should be tested in the same horizontal- or vertical-shaft position as the prototype. The speed of the model should be such that, at the test head, the specific speed for each test run is the same as that of the installed unit or prototype. Unless otherwise specified, the suction head or suction lift should give the same (cavitation factor) value.

If model and prototype diameters are D_1 and D, respectively, then the model speed N_1 and capacity Q_1 under the test head H_1 must agree with the relations

$$\frac{N_1}{N} = \frac{D}{D_1}\sqrt{\frac{H_1}{H}} \quad \text{and} \quad \frac{Q_1}{Q} = \left(\frac{D_1}{D}\right)^2\sqrt{\frac{H_1}{H}}$$

In testing a model of reduced size under the previous conditions, complete hydraulic similarity will not be secured unless the relative roughness of the impeller and pump casing surfaces are the same. With the same surface texture in model and prototype, the model efficiency will be lower than that of the prototype, and greater relative clearances and shaft friction in the model will also reduce its efficiency.

The efficiency of a pump model can conveniently be stepped up to match the prototype efficiency by applying a formula of the same general form as the Moody formula used for hydraulic turbines:

$$\frac{1 - e_1}{1 - e} = \left(\frac{D}{D_1}\right)^n$$

The exponent n should be determined for a given laboratory and given type of pump on the basis of an adequate number of comparisons of the efficiencies of models and prototypes, with consistent surface finish in model and prototype. The *Hydraulic Institute Standards*[2] states that n has been found to vary from zero (when the surface roughness

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and clearances of the model and prototype are proportional to their size) to 0.26 (when the absolute roughness is the same in both).

When model tests are to serve as acceptance tests, it is generally recommended that the efficiency guarantees be stated in terms of model performance rather than in terms of calculated prototype performance. In the absence of such a provision, the efficiency stepup formula and the numerical value of its exponent should be clearly specified or agreed upon in advance of tests.

The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (Reference 10) give an example of model testing as follows:

EXAMPLE A single-stage pump to deliver 200 ft³/s (5.66 m³/s) against a head of 400 ft (122 m) at 450 rpm and with a positive suction head, including velocity head, of 10 ft (3 m) has an impeller diameter of 6.8 ft (2.1m). The pump being too large for a shop or laboratory test, a model with an 18-in (0.46-m) impeller is to be tested at a reduced head at 320 ft (97.5 m). At what speed, capacity, and suction head should the test be run?

Applying the above relations:

in USCS units $$N_1 = N\frac{D}{D_1}\sqrt{\frac{H_1}{H}} = 450\left(\frac{6.8}{1.5}\right)\sqrt{\frac{320}{400}} = 1825 \text{ rpm}$$

$$Q_1 = Q\left(\frac{D_1}{D}\right)^2\sqrt{\frac{H_1}{H}} = 200\left(\frac{1.5}{6.8}\right)^2\sqrt{\frac{320}{400}} = 8.73 \text{ ft}^3/\text{s} = 3920 \text{ gpm}$$

in SI units $$N_1 = 450\left(\frac{2.1}{0.46}\right)^2\sqrt{\frac{97.5}{122}} = 1825 \text{ rpm}$$

$$Q_1 = 5.66\left(\frac{0.46}{22.1}\right)^2\sqrt{\frac{97.5}{122}} = 0.247 \text{ m}^3/\text{s}$$

To check these results, the specific speed of the prototype is

in USCS units $$N_s = N\frac{\sqrt{Q}}{H^{3/4}} = 450\frac{\sqrt{200}}{400^{3/4}} = 71.2 \text{ in the ft}^3/\text{s system}$$

in SI units $$N_s = \frac{450\sqrt{5.66}}{122^{3/4}} = 39 \text{ in the m}^3/\text{s system}$$

and that of the model is

in USCS units $$N_{s1} = 1.825\frac{\sqrt{8.73}}{320^{3/4}} = 71.2 \text{ (or 1,510 in the gpm system)}$$

in SI units $$N_{s1} = 1825\frac{\sqrt{0.247}}{97.5^{314}} = 29$$

The cavitation factor σ for the field installation, assuming a water temperature of 80°F (27°C) as a maximum probable value and H_b = 32.8 ft (10 m) as in the first example, will be

in USCS units $$\sigma = \frac{H_b - H_s}{H} = \frac{32.8 - 10}{400} = 0.057$$

in SI units $$\sigma = \frac{10 - 3}{122} = 0.057$$

where $H_b = h_{sa} - H_{vpa}$ (absolute atmospheric pressure minus absolute vapor pressure)

H_s = distance from datum to suction level

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which should be the same in the test. With the water temperature approximately the same,

$$\sigma = \frac{H_b - H_{s1}}{H_1}$$

in USCS units $H_{s1} = H_b - \sigma H = 32.8 - (0.057)(320) = 14.6$ ft

in SI units $H_{s1} = 10 - (0.057)(97.5) = 4.4$ m

Hence the model should be tested with a positive suction head of 14.6 ft (4.4 m) to reproduce the field conditions.

Normally, one of the requirements when using model tests as acceptance tests is to make sure true geometric similarity exists between model and installed prototype. True values of all required and specified dimension should be determined. The actual parts, areas, shape, clearances, and positions should be clearly understood by all parties. Also, the amount of permissible geometric deviation between prototype and model should be agreed to, in writing, before the test is begun.

OTHER OBSERVATIONS

Testing Noncentrifugal Pumps The next largest class of pumps after centrifugal are displacement pumps. This classification includes reciprocating, rotary, screw, and other miscellaneous displacement pumps. Testing of these closely parallels the centrifugal procedures. Normally the capacities are smaller and the heads higher, but the objectives, methods, and measurements are all about the same. The Hydraulic Institute ANSI/HI 2000 Edition Pump Standards (References 11 and 12) thoroughly cover testing of rotary and reciprocating pumps.

The testing of pumps not falling into the two broad classifications of centrifugal and displacement is usually very special, and each case must be treated separately. The test procedures are normally spelled out in the specifications; otherwise an agreement between all parties must be made before testing is started. The testing of eduction or jet pumps falls under this special category of testing.

Other Test Phenomena When testing pumps, other phenomena of interest should also be checked and noted on the test record. The two phenomena normally reported on are vibration and noise. The acceptable limits of these plus instrumentation for measuring them are special and normally covered in the contract. If the pumps are to be installed in a special environment, this should also be taken into consideration during testing.

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A • P • P • E • N • D • I • X

TECHNICAL DATA

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NOTE: For more units and conversions and wealth of general technical information, the following resource is recommended: www.nist.gov

TABLE 1A Base units of the International System of Units (SI) and their definitions

Symbol	Unit	Definition
m	meter[a]	Unit of length equal to 1,650,763.73 wavelengths in vacuum of the radiation corresponding to the transition between the $2p^{10}$ and $5d^5$ levels of the krypton-86 atom
kg	kilogram	Unit of mass equal to that of the international prototype of the kilogram
s	second	Unit of time equal to the duration of 9,192,631,770 periods of the radiation corresponding to the transition between two hyperfine levels of the ground state of the cesium-133 atom
A	ampere	Unit of electric current equal to that which, if maintained in two straight parallel conductors of infinite length and negligible cross-section and placed 1 meter (m) apart in vacuum, produces between those conductors a force equal to 2×10^{-7} newtons per meter (N/m) of length
K	kelvin	Unit of thermodynamic temperature equal to the fraction 1/273.16 of the thermodynamic temperature of the triple point of water
cd	candela	Unit of luminous intensity equal to that (in the perpendicular direction) of a surface of 1/600,000 m^2 of a blackbody at the temperature of freezing platinum under a pressure of 101,325 newtons per square meter (N/m^2)
mol	mole	Unit of substance equal to the moment of substance of a system that contains as many elementary entities[b] as there are atoms in 0.012 kg of carbon-12

[a]Spelled *metre* in countries other than the United States.

[b]When the mole is used, the elementary entities must be specified and may be atoms, molecules, ions, electrons, other particles, or specified groups of each particle.

TABLE 1B Supplementary SI units and their definitions

Symbol	Unit	Definition
rad	radian	Unit of measure of a planar angle with its vertex at the center of a circle subtended by an arc equal in length to the radius
sr	steradian	Unit of measure of a solid angle with its vertex at the center of a sphere and enclosing an area of the spherical surface equal to that of a square with sides equal in length to the radius

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TABLE 2 Derived units of the SI system

Quantity	Unit	Symbol	Formula
Acceleration	meters per second per second	m/s^2	—
Angular acceleration	radians per second per second	rad/s^2	—
Angular velocity	radians per second	rad/s	—
Area	square meters	m^2	—
Density	kilograms per cubic meter	kg/m^3	—
Energy	joules	J	$N \cdot m$
Force	newtons	N	$kg \cdot m/s^2$
Frequency	hertz	Hz	cycle/s
Power	watts	W	J/s
Pressure	pascals	Pa	N/m^2
Stress	newtons per square meter	N/m^2	—
Velocity	meters per second	m/s	—
Viscosity, dynamic	newton-seconds per square meter	$N \cdot s/m^2$	—
Viscosity, kinematic	square meters per second	m^2/s	—
Volume	cubic meters	m^3	—
Work	joules	J	$N \cdot m$

TABLE 3 Prefixes for SI multiple and submultiple units

Prefix	SI symbol	Multiplication factor
exa	E	10^{18}
peta	P	10^{15}
tera	T	10^{12}
giga	G	10^{9}
mega	M	10^{6}
kilo	k	10^{3}
hecto	h	10^{2}
deka	da	10
deci	d	10^{-1}
centi	c	10^{-2}
milli	m	10^{-3}
micro	μ	10^{-6}
nano	n	10^{-9}
pico	p	10^{-12}
femto	f	10^{-15}
atto	a	10^{-18}

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TABLE 4 Velocity and frictional head loss in old cast iron piping based on Hazen-Williams $C = 100$

gpm[a]	*v*[b]	*f*[c]	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	*v*	*f*	gpm
							3-in ID pipe[d]		4-in ID pipe										
30							1.36	0.534	0.77	0.131	5-in ID pipe								30
40							1.81	0.910	1.02	0.224									40
50							2.27	1.38	1.28	0.338	0.82	0.114	6-in ID pipe						50
60							2.72	1.92	1.53	0.475	0.98	0.160							60
70							3.18	2.56	1.79	0.631	1.14	0.213	0.79	0.088					70
80							3.63	3.28	2.04	0.808	1.31	0.273	0.91	0.112					80
90							4.08	4.08	2.30	1.01	1.47	0.339	1.02	0.139					90
100							4.54	4.96	2.55	1.22	1.63	0.412	1.14	0.170	8-in ID pipe				100
125							5.68	7.50	3.19	1.85	2.04	0.623	1.42	0.256					125
150							6.81	10.5	3.83	2.59	2.47	0.874	1.70	0.360	0.96	0.89			150
175							7.95	14.0	4.47	3.44	2.86	1.16	1.99	0.478	1.12	0.118			175
200							9.08	17.9	5.10	4.41	3.27	1.49	2.27	0.613	1.28	0.151	10-in ID pipe		200
225							10.2	22.3	5.74	5.48	3.68	1.85	2.55	0.762	1.44	0.188			225
250							11.3	27.1	6.38	6.67	4.08	2.25	2.84	0.926	1.60	0.228	1.02	0.077	250
275							12.5	32.3	7.02	7.96	4.50	2.68	3.12	1.11	1.76	0.272	1.12	0.092	275
	12-in ID pipe																		
300							13.6	37.9	7.65	9.34	4.90	3.13	3.41	1.30	1.91	0.320	1.23	0.108	300
350	0.99	0.059	14-in ID pipe				15.9	50.4	8.93	12.4	5.72	4.20	3.97	1.73	2.23	0.425	1.43	0.144	350
400	1.13	0.076					18.2	64.6	10.2	15.9	6.54	5.38	4.54	2.21	2.55	0.545	1.63	0.184	400
450	1.28	0.094	0.94	0.044					11.5	19.8	7.36	6.68	5.10	2.75	2.87	0.678	1.84	0.228	450
500	1.42	0.114	1.04	0.054				12.8	24.1	8.18	8.12	5.68	3.34	3.19	0.823	2.04	0.278		500
					16-in ID pipe														
550	1.56	0.136	1.15	0.064					14.0	28.7	8.99	9.69	6.24	3.99	3.51	0.982	2.24	0.331	550
600	1.70	0.160	1.25	0.076	0.96	0.039			15.3	33.7	9.81	11.4	6.81	4.68	3.82	1.15	2.45	0.389	600
650	1.84	0.186	1.36	0.088	1.04	0.046	18-in ID pipe		16.6	39.1	10.6	13.2	7.38	5.43	4.15	1.34	2.65	0.452	650
700	1.99	0.214	1.46	0.100	1.12	0.052			17.9	44.9	11.4	15.1	7.94	6.23	4.47	1.53	2.86	0.518	700
750	2.13	0.242	1.56	0.114	1.20	0.060	0.95	0.034			12.3	17.2	8.51	7.08	4.78	1.74	3.06	0.589	750
800	2.27	0.273	1.67	0.129	1.28	0.067	1.01	0.038	20-in ID pipe		13.1	19.4	9.08	7.98	5.10	1.97	3.26	0.666	800
900	2.55	0.339	1.88	0.160	1.44	0.084	1.13	0.047			14.7	24.1	10.2	9.92	5.74	2.44	3.67	0.825	900
1,000	2.83	0.412	2.08	0.195	1.60	0.102	1.26	0.057	1.02	0.034	16.3	29.3	11.4	12.1	6.38	2.97	4.08	1.00	1,000
1,100	3.12	0.492	2.29	0.232	1.76	0.121	1.39	0.068	1.12	0.041	18.0	35.0	12.5	14.4	7.02	3.55	4.50	1.20	1,100
1,200	3.40	0.578	2.50	0.273	1.92	0.143	1.51	0.080	1.23	0.048			13.6	16.9	7.66	4.17	4.90	1.41	1,200

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1,300	3.69	0.671	2.71	0.316	2.08	0.165	1.64	0.093	1.33	0.056	24-in ID pipe		14.8	19.6	8.30	4.83	5.31	1.63	1,300
1,400	3.97	0.770	2.92	0.363	2.24	0.190	1.76	0.107	1.43	0.064			15.9	22.5	8.93	5.54	5.72	1.87	1,400
1,500	4.25	0.875	3.12	0.413	2.40	0.215	1.89	0.121	1.53	0.073	1.06	0.030	17.0	25.5	9.55	6.30	6.12	2.13	1,500
1,600	4.54	0.985	3.33	0.465	2.55	0.243	2.02	0.137	1.63	0.082	1.13	0.034	18.2	28.8	10.2	7.10	6.53	2.39	1,600
1,800	5.11	1.22	3.75	0.578	2.87	0.302	2.27	0.170	1.84	0.102	1.28	0.042			11.5	8.83	7.35	2.98	1,800
2,000	5.67	1.49	4.17	0.703	3.19	0.367	2.52	0.207	2.04	0.124	1.42	0.051			12.8	10.7	8.17	3.62	2,000
2,500	7.09	2.25	5.21	1.06	3.99	0.555	3.15	0.312	2.55	0.187	1.77	0.077			16.0	16.2	10.2	5.48	2,500
3,000	8.51	3.16	6.25	1.49	4.78	0.778	3.78	0.438	3.06	0.262	2.13	0.108			19.1	22.8	12.3	7.67	3,000
3,500	9.93	4.20	7.29	1.98	5.59	1.04	4.41	0.583	3.57	0.349	2.48	0.143					14.3	10.2	3,500
4,000	11.3	5.38	8.33	2.54	6.39	1.33	5.04	0.746	4.08	0.447	2.83	0.184					16.3	13.1	4,000
4,500	12.8	6.68	9.38	3.15	7.18	1.65	5.67	0.928	4.59	0.555	3.19	0.228					18.4	16.3	4,500
5,000	14.2	8.13	10.4	3.83	7.98	2.00	6.30	1.13	5.10	0.675	3.54	0.278							5,000
5,500	15.6	9.70	11.5	4.58	8.78	2.39	6.93	1.35	5.61	0.806	3.90	0.332							5,500
6,000	17.0	11.4	12.5	5.38	9.68	2.81	7.56	1.58	6.12	0.947	4.25	0.390							6,000
6,500	18.4	13.2	13.6	6.24	10.4	3.26	8.19	1.83	6.73	1.10	4.61	0.452							6,500
7,000	19.9	15.2	14.6	7.16	11.2	3.74	8.82	2.11	7.15	1.26	4.96	0.518							7,000
7,500			15.6	8.13	12.0	4.24	9.45	2.39	7.66	1.43	5.32	0.589							7,000
8,000			16.7	9.16	12.8	4.79	10.1	2.69	8.17	1.61	5.66	0.664							8,000
9,000			18.8	11.4	14.4	5.95	11.3	3.39	9.18	2.01	6.38	0.825							9,000
10,000					16.0	7.24	12.6	4.07	10.2	2.44	7.09	1.00							10,000
11,000					17.6	8.63	13.9	4.86	11.2	2.91	7.80	1.20							11,000
12,000					19.2	10.1	15.1	5.71	12.3	3.42	8.51	1.41							12,000
13,000							16.4	6.62	13.3	3.96	9.12	1.63							13,000
14,000							17.6	7.59	14.3	4.54	9.93	1.87							14,000
15,000							18.9	8.63	15.3	5.27	10.6	2.13							15,000
16,000									16.3	5.82	11.3	2.40							16,000
18,000									18.4	7.24	12.8	2.98							18,000
20,000											14.2	3.62							20,000
25,000											17.7	5.48							25,000

[a]1 gpm = 6.31 × 10−5 m^3/s = 0.227 m^3/h = 0.0631 liters/s

[b]Velocity, in feet per second. 1 ft/s = 0.305 m/s

[c]Frictional head loss, in feet of water per 100 feet of pipe. 1 ft = 0.305 m. Friction values apply to cast iron pipes after 15 years of service handling average water at 60°F (15.6°C). Based on Hazen-Williams formula with C = 100. See Figure 1 for other values of C.

[d]1 in = 25.4 mm

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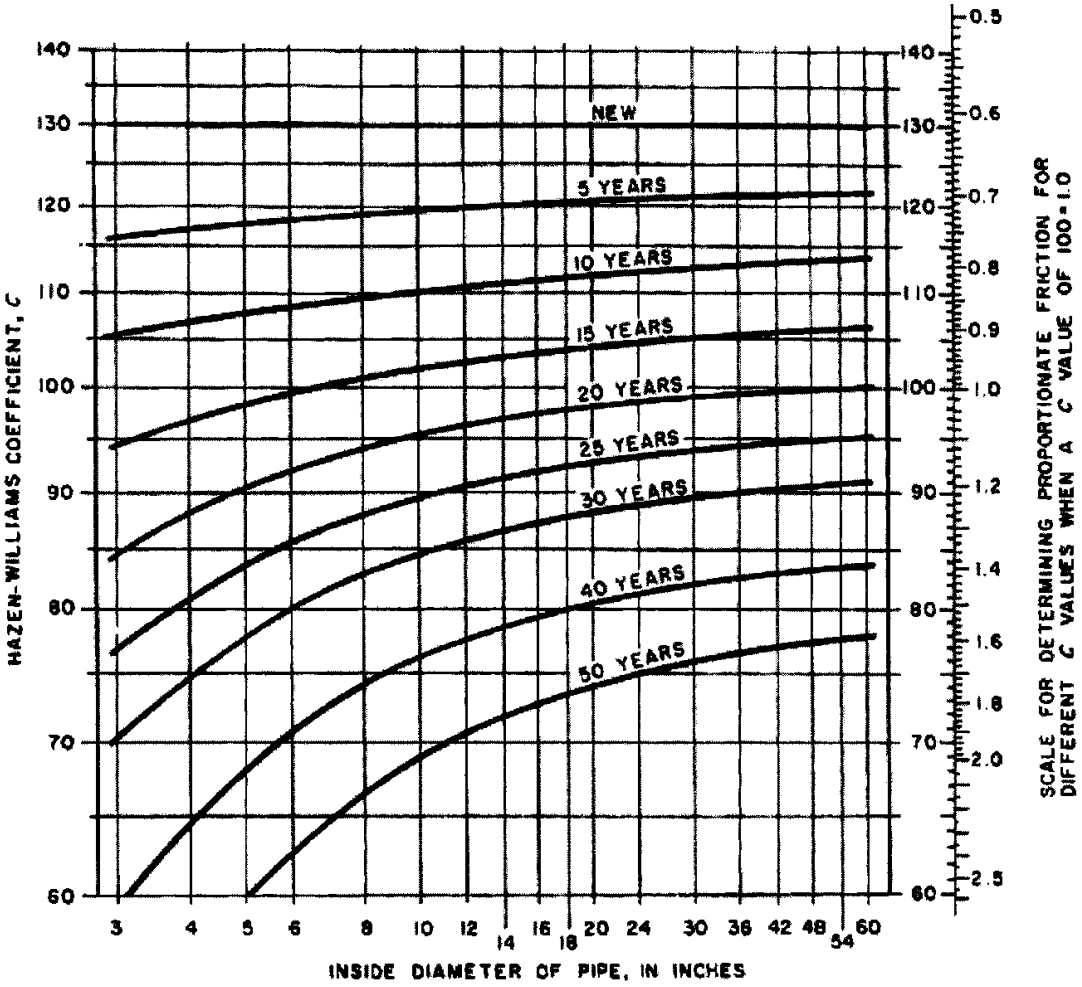


FIGURE 1 Change in Hazen-Williams coefficient C with years of service, for cast iron pipes handling soft, clear, unfiltered water at 60°F (15.6°C). To correct head loss from Table 4, which is based on $C = 100$, multiply by the conversion factor. For example, with a flow of 700 gpm through a 6-in pipe, the frictional head loss is 6.23 ft per 100 ft of pipe with $C = 100$. For $C = 130$, the conversion factor is 0.613; therefore, the frictional head loss is $6.23 \times 0.613 = 3.82$ ft per 100 ft of pipe (see also Section 8.1) (1 in = 25.4 mm).

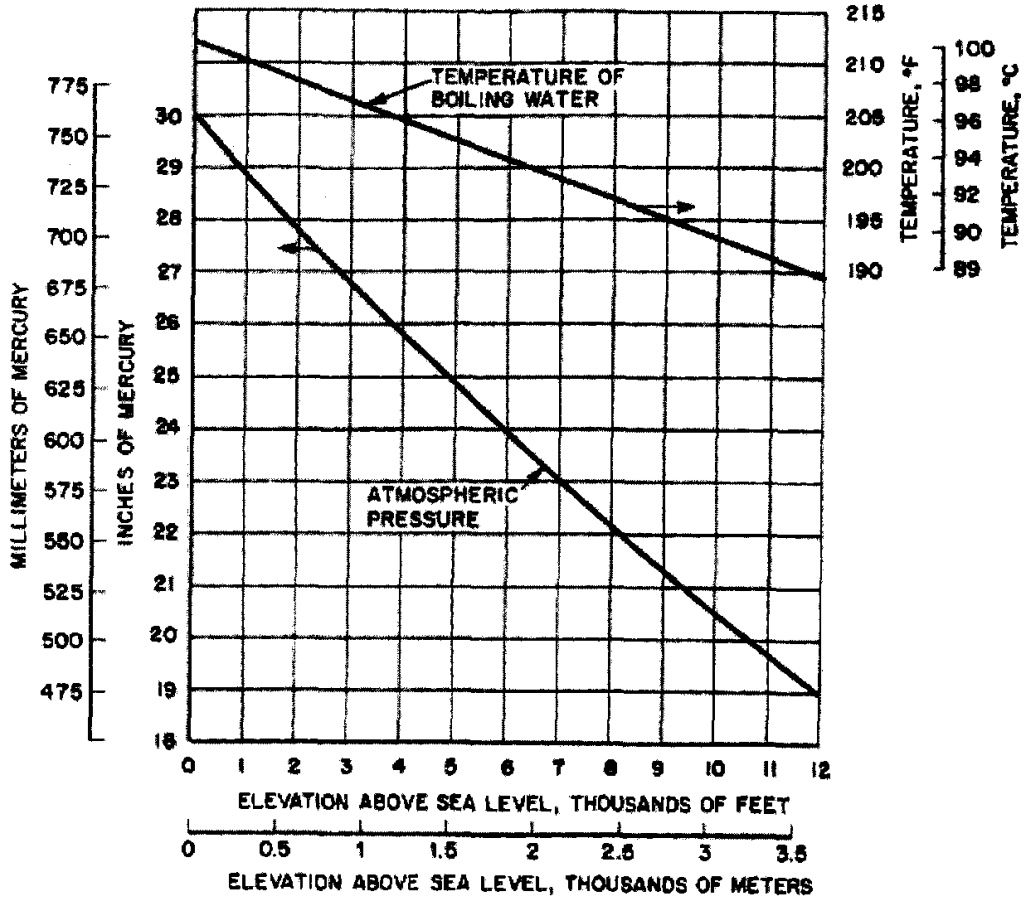


FIGURE 2 Atmospheric pressures for altitudes up to 12,000 ft (3600 m)

TABLE 5 Velocity and frictional head loss in new schedule 40 (standard weight) steel piping based on Darcy-Weisbach

gpm[a]	v[b]	f[c]	v	f	v	f	v	f	v	f	v	f	v	f	v	f	gpm
					1-in pipe[d] (1.049-in ID)		$1\frac{1}{4}$-in pipe (1.380-in ID)		$1\frac{1}{2}$-in pipe (1.610-in ID)								
1					0.37	0.11											1
2					0.74	0.39	0.43	0.10									2
3					1.11	0.82	0.64	0.21	0.47	0.10							3
4					1.49	1.37	0.86	0.36	0.63	0.17	2-in pipe						4
5					1.86	2.08	1.07	0.54	0.79	0.26	(2.067-in ID)						5
													$2\frac{1}{2}$-in pipe				
6					2.23	2.83	1.28	0.76	0.95	0.35	0.57	0.10	(2.469-in ID)				6
8					2.97	4.88	1.72	1.29	1.26	0.61	0.76	0.17			3-in pipe		8
10					3.71	7.12	2.14	1.95	1.57	0.90	0.96	0.26	0.67	0.11	(3.068-in ID)		10
15	$3\frac{1}{2}$-in pipe				5.56	15.0	3.21	4.06	2.36	1.87	1.43	0.54	1.00	0.23			15
20	(3.548-in ID)				7.41	25.6	4.28	6.80	3.15	3.12	1.91	0.92	1.34	0.38	0.87	0.13	20
			4-in pipe														
25	0.81	0.10	(4.026-in ID)				5.35	10.3	3.94	4.70	2.38	1.39	1.67	0.58	1.08	0.20	25
30	0.97	0.13					6.43	14.4	4.72	6.60	2.86	1.92	2.00	0.81	1.30	0.28	30
40	1.30	0.23	1.01	0.12					6.30	11.2	3.82	3.35	2.68	1.36	1.73	0.47	40
50	1.62	0.34	1.26	0.18	5-in pipe				7.87	16.6	4.77	5.00	3.34	2.06	2.16	0.72	50
60	1.94	0.48	1.51	0.25	(5.047-in ID)						5.72	7.00	4.02	2.85	2.60	0.99	60
70	2.27	0.63	1.76	0.34	1.12	0.11	6-in pipe				6.68	9.40	4.68	3.80	3.03	1.33	70
80	2.59	0.82	2.01	0.43	1.28	0.15	(6.065-in ID)				7.62	11.9	5.35	4.95	3.46	1.72	80
90	2.91	1.00	2.27	0.54	1.44	0.18					8.60	14.7	6.02	6.05	3.89	2.13	90
100	3.24	1.24	2.52	0.65	1.60	0.22	1.11	0.09			9.56	18.7	6.70	7.47	4.83	2.58	100
125	4.05	1.82	3.15	1.00	2.00	0.33	1.39	0.13					8.37	11.1	5.41	3.90	125
150	4.86	2.55	3.78	1.39	2.40	0.47	1.66	0.18	8-in pipe				10.0	15.4	6.50	5.44	150
175	5.66	3.40	4.40	1.90	2.80	0.62	1.94	0.24	(7.981-in ID)				11.7	20.8	7.58	7.30	175
200	6.48	4.35	5.04	2.40	3.20	0.80	2.22	0.31							8.66	9.18	200
225	7.30	5.44	5.66	2.98	3.60	0.97	2.50	0.38	1.44	0.10					9.75	11.6	225
250	8.10	6.59	6.29	3.68	4.00	1.19	2.77	0.47	1.60	0.12					10.8	14.0	250

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275	8.91	7.90	6.92	4.35	4.40	1.43	3.05	0.56	1.76	0.15	10-in pipe		11.9	16.9	275
300	9.72	9.30	7.55	5.04	4.80	1.65	3.32	0.66	1.92	0.17	(10.020-in ID)		13.0	19.6	300
350	11.3	12.2	8.80	6.85	5.60	2.21	3.88	0.88	2.24	0.23					350
400	13.0	15.9	10.1	8.67	6.40	2.89	4.44	1.12	2.56	0.29	1.62	0.10			400
450	14.6	20.0	11.3	10.9	7.20	3.56	4.99	1.40	2.88	0.37	1.82	0.12			450
500			12.6	13.3	8.00	4.36	5.54	1.72	3.20	0.45	2.03	0.15			500
550			13.9	16.0	8.80	5.17	6.10	2.06	3.52	0.55	2.23	0.18			550
600			15.1	19.1	9.60	6.16	6.65	2.42	3.84	0.63	2.44	0.21			600
650					10.4	7.22	7.20	2.78	4.16	0.73	2.64	0.24			650
700					11.2	8.29	7.75	3.25	4.47	0.85	2.84	0.28			700
750					12.0	9.40	8.31	3.63	4.80	0.97	3.04	0.31			750
800					12.8	10.3	8.87	4.11	5.11	1.11	3.25	0.35			800
900					14.4	13.0	9.96	5.12	5.75	1.33	3.65	0.44			900
1,000					16.0	15.8	11.1	6.17	6.40	1.64	4.06	0.55			1,000
1,100					17.6	19.0	12.2	7.45	7.04	1.98	4.46	0.64			1,100
1,200							13.3	8.73	7.67	2.36	4.87	0.75			1,200
1,300							14.4	10.2	8.31	2.71	5.27	0.88			1,300
1,400							15.5	11.9	8.95	3.10	5.68	1.02			1,400
1,500							16.7	13.2	9.60	3.49	6.09	1.18			1,500
1,600							17.8	15.0	10.2	3.92	6.49	1.31			1,600
1,800							20.0	18.5	11.5	4.99	7.30	1.60			1,800
2,000									12.8	5.96	8.11	1.97			2,000
2,500									16.0	9.00	10.2	2.95			2,500
3,000									19.2	12.5	12.2	4.15			3,000
3,500									22.4	16.6	14.2	5.60			3,500
4,000											16.2	6.90			4,000
4,500											18.3	8.80			4,500
5,000											20.3	10.8			5,000
5,500											22.3	13.0			5,500
6,000											24.4	15.3			6,000

[a]1 gpm = 6.31×10^{-5} m³/s = 0.227 m³/h = 0.0631 liters/s

[b]Velocity, in feet per second. 1 ft/s = 0.305 m/s

[c]Frictional head loss, in feet of water per 100 feet of pipe. 1 ft = 0.305 m. Friction values apply to pipe carrying water at 60°F (15.6°C).

[d]1 in = 25.4 mm

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TABLE 6 Viscosity of common liquids
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Liquid	Sp. gr. at 60°F (15.6°C)[a]	Viscosity			
		SSU	Centistokes	At °F	At °C
Freon	1.37 to 1.49 @ 70°F (21°C)		0.27–0.32	70	21
Glycerine (100%)	1.26 @ 68°F (20°C)	2950	648	68.6	20.3
		813	176	100	38
Glycol					
Propylene	1.038 @ 68°F (20°C)	240.6	52	70	21
Triethylene	1.125 @ 68°F (20°C)	185.7	40	70	21
Diethylene	1.12	149.7	32	70	21
Ethylene	1.125	88.4	17.8	70	21
Hydrochloric acid (31.5%)	1.05 @ 68°F (20°C)		1.9	68	20
Mercury	13.6		0.118	70	21
			0.11	100	38
Phenol (carbolic acid)	0.95 to 1.08	65	11.7	65	18
Silicate of soda	40 Baumé	365	79	100	38
	42 Baumé	637.6	138	100	38
Sulfuric acid (100%)	1.83	75.7	14.6	68	20
		Fish and animal oils			
Bone oil	0.918	220	47.5	130	54
		65	11.6	212	100
Cod oil	0.928	**150**	32.1	100	38
		95	19.4	130	54
Lard	0.96	**287**	62.1	100	38
		160	34.3	130	54
Lard oil	0.912 to 0.925	**190** to **220**	41 to 47.5	100	38
		112 to 128	23.4 to 27.1	130	54

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Menhaden oil	0.933	**140**	29.8	100	38
		90	18.2	130	54
Neatsfoot oil	0.917	**230**	49.7	100	38
		130	27.5	130	54
Sperm oil	0.883	**110**	23.0	100	38
		78	15.2	130	54
Whale oil	0.925	**163** to **184**	35 to 39.6	100	38
		97 to 112	19.9 to 23.4	130	54
		Mineral oils			
Automobile crankcase oils (average mid-continent paraffin base):					
SAE 10	0.880 to 0.935[b]	165 to 240	35.4 to 51.9	100	38
		90 to **120**	18.2 to 25.3	130	54
SAE 20	0.880 to 0.935[b]	240 to 400	51.9 to 86.6	100	38
		120 to **185**	25.3 to 39.9	100	54
SAE 30	0.880 to 0.935[b]	400 to 580	86.6 to 125.5	100	38
		185 to **255**	39.9 to 55.1	130	54
SAE 40	0.880 to 0.935[b]	580 to 950	125.5 to 205.6	100	38
		255 to	55.1 to	130	54
		80	15.6	210	99
SAE 50	0.880 to 0.935[b]	950 to 1600	205.6 to 352	100	38
		80 to **105**	15.6 to 21.6	210	99
SAE 60	0.880 to 0.935[b]	1600 to 2300	352 to 507	100	38
		105 to **125**	21.6 to 26.2	210	99
SAE 70	0.880 to 0.935[b]	2300 to 3100	507 to 682	100	38
		125 to **150**	26.2 to 31.8	210	99
SAE 10W	0.880 to 0.935[b]	**5000** to **10,000**	1100 to 2200	0	−18
SAE 20W	0.880 to 0.935[b]	**10,000** to **40,000**	2200 to 8800	0	−18

(*Continued*)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Mineral oils					
Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Automobile transmission lubricants					
SAE 80	0.880 to 0.935[b]	**100,000** max	22,000 max	0	−18
SAE 90	0.880 to 0.935[b]	**800** to **1500**	173.2 to 324.7	100	38
		300 to 500	64.5 to 108.2	130	54
SAE 140	0.880 to 0.935[b]	950 to 2300	205.6 to 507	130	54
		120 to **200**	25.1 to 42.9	210	99
SAE 250	0.880 to 0.935[b]	Over 2300	Over 507	130	54
		Over 200	Over 42.9	210	99
Crude oils					
Texas, Oklahoma	0.81 to 0.916	40 to 783	4.28 to 169.5	60	16
		34.2 to 210	2.45 to 45.3	100	38
Wyoming, Montana	0.86 to 0.88	74 to 1215	14.1 to 263	60	16
		46 to 320	6.16 to 69.3	100	38
California	0.78 to 0.92	40 to 4840	4.28 to 1063	60	16
		34 to 700	2.4 to 151.5	100	38
Pennsylvania	0.8 to 0.85	46 to 216	6.16 to 46.7	60	16
		38 to 86	3.64 to 17.2	100	38
Diesel engine lubricating oils (based on average mid-continent paraffin base)					
Federal specification no. 9110	0.880 to 0.935[b]	165 to 240	35.4 to 51.9	100	38
		90 to **120**	18.2 to 25.3	130	54
Federal specification no. 9170	0.880 to 0.935[b]	300 to 410	64.5 to 88.8	100	38
		140 to **180**	29.8 to 38.8	130	54
Federal specification no. 9250	0.880 to 0.935[b]	470 to 590	101.8 to 127.8	100	38
		200 to **255**	43.2 to 55.1	130	54

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Federal specification no. 9370	0.880 to 0.935[b]	800 to 1100	173.2 to 238.1	100	38
		320 to **430**	69.3 to 93.1	130	54
Federal specification no. 9500	0.880 to 0.935[b]	490 to 600	106.1 to 129.9	130	54
		92 to **105**	18.54 to 21.6	210	99
Diesel fuel oils					
No. 2 D	0.82 to 0.95[b]	32.6 to 45.5	2 to 6	100	38
		39	1 to 3.97	130	54
No. 3 D	0.82 to 0.95[b]	**45.5** to **65**	6 to 11.75	100	38
		39 to 48	3.97 to 6.78	130	54
No. 4 D	0.82 to 0.95[b]	**140** max	29.8 max	100	38
		70 max	13.1 max	130	54
No. 5 D	0.82 to 0.95[b]	**400** max	86.6 max	122	50
		165 max	35.2 max	160	71
Fuel oils					
No. 1	0.82 to 0.95[b]	34 to 40	2.39 to 4.28	70	21
		32 to **35**	2.69	100	38
No. 2	0.82 to 0.95[b]	36 to 50	3.0 to 7.4	70	21
		33 to **40**	2.11 to 4.28	100	38
No. 3	0.82 to 0.95[b]	**35** to **45**	2.69 to 0.584	100	38
		32.8 to 39	2.06 to 3.97	130	54
No. 5A	0.82 to 0.95[b]	**50** to **125**	7.4 to 26.4	100	38
		42 to 72	4.91 to 13.73	130	54
No. 5B	0.82 to 0.95[b]	125 to **400**	26.4 to 86.6	100 122	38 50
		72 to 310	13.63 to 67.1	130	54
No. 6	0.82 to 0.95[b]	**450** to **3000**	97.4 to 660	122	50
		175 to 780	37.5 to 172	160	71

(Continued)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Mineral oils					
Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Fuel oil, navy specification	0.989 max[b]	**110** to **225**	23 to 48.6	122	50
		63 to 115	11.08 to 23.9	160	71
Fuel oil, navy II	1.0 max	**1500** max	324.7 max	122	50
		480 max	104 max	60	71
Gasoline	0.68 to 0.74		0.46 to 0.88	60	16
			0.40 to 0.71	100	38
Gasoline (natural)	76.5 degrees API		0.41	68	20
Gas oil	28 degrees API	73	13.9	70	21
		50	7.4	100	38
Insulating oil					
Transformer, switches, and circuit breakers		115 max	24.1 max	70	21
		65 max	11.75 max	100	38
Kerosene	0.78 to 0.82	35	2.69	68	20
		32.6	2	100	38
Machine lubricating oil (average Pennsylvania paraffin base)					
Federal specification no. 8	0.880 to 0.935[b]	112 to 160	23.4 to 34.3	100	38
		70 to **90**	13.1 to 18.2	130	54
Federal specification no. 10	0.880 to 0.935[b]	160 to 235	34.3 to 50.8	100	38
		90 to **120**	18.2 to 25.3	130	54
Federal specification no. 20	0.880 to 0.935[b]	235 to 385	50.8 to 83.4	100	38
		120 to **185**	25.3 to 39.9	130	54
Federal specification no. 30	0.880 to 0.935[b]	385 to 550	83.4 to 119	100	38
		185 to 255	39.9 to 55.1	130	54

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Mineral lard cutting oil					
Federal specification grade 1		**140** to **190**	29.8 to 41	100	38
		86 to 110	17.22 to 23	130	54
Federal specification grade 2		**190** to **220**	41 to 47.5	100	38
		110 to 125	23 to 26.4	130	54
Petrolatum	0.825	100	20.6	130	54
		77	14.8	160	71
Turbine lubricating oil					
Federal specification (penn base)	0.91 average	400 to 440	86.6 to 95.2	100	38
		185 to **205**	39.9 to 44.3	130	54
		Vegetable oils			
Castor oil	0.96 @ 68°F (20°C)	1200 to 1500	259.8 to 324.7	100	38
		450 to 600	97.4 to 129.9	130	54
China wood oil	0.943	**1425**	308.5	69	21
		580	125.5	100	38
Coconut oil	0.925	**140** to **148**	29.8 to 31.6	100	38
		76 to 80	14.69 to 15.7	130	54
Corn oil	0.924	135	28.7	130	54
		54	8.59	212	100
Cotton seed oil	0.98 to 0.925	**176**	37.9	100	38
		100	20.6	130	54
Linseed oil, raw	0.925 to 0.939	**143**	30.5	100	38
		93	18.94	130	54
Olive oil	0.912 to 0.918	**200**	43.2	100	38
		115	24.1	130	54

(*Continued*)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Vegetable oils					
Palm oil	0.924	**221**	47.8	100	38
		125	26.4	130	54
Peanut Oil	0.920	**221**	47.8	100	38
		125	26.4	130	54
Rape seed oil	0.919	**250**	54.1	100	38
		145	31	130	54
Rosin oil	0.980	**1500**	324.7	100	38
		600	129.9	130	54
Rosin (wood)	1.09 avg.	500 to **20,000**	108.2 to 4,400	200	93
		1,000 to 50,000	216.4 to 11,000	190	88
Sesame oil	0.923	**184**	39.6	100	38
		110	23	130	54
Soybean oil	0.927 to 0.98	**165**	35.4	100	38
		96	19.64	130	54
Turpentine	0.86 to 0.87	33	2.11	60	16
		32.6	2.0	100	38
Sugar, syrups, molasses, and so on					
Corn syrups	1.4 to 1.47	5,000 to 500,000	1,100 to 110,000	100	38
		1,500 to 60,000	324.7 to 13,200	130	54
Glucose	1.35 to 1.44	35,000 to 100,000	7,700 to 22,000	100	38
		4,000 to 11,000	880 to 2,420	150	66
Honey (raw)		340	73.6	100	38
Molasses "A" (first)	1.40 to 1.46	1,300 to 23,000	281.1 to 5,070	100	38
		700 to 8,000	151.5 to 1,760	130	54

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Molasses "B" (second)	1.43 to 1.48	6,400 to 60,000	1,410 to 13,200	100	38
		3,000 to 15,000	660 to 3,300	130	54
Molasses "C" (blackstrap or final)	1.46 to 1.49	17,000 to 250,000	2,630 to 5,500	100	38
		6,000 to 75,000	1,320 to 16,500	130	54
Sucrose solutions (sugar syrups)					
60 Brix	1.29	230	49.7	70	21
		92	18.7	100	38
62 Brix	1.30	310	67.1	70	21
		111	23.2	100	38
64 Brix	1.31	440	95.2	70	21
		148	31.6	100	38
66 Brix	1.326	650	140.7	70	21
		195	42.0	100	38
68 Brix	1.338	1,000	216.4	70	21
		275	59.5	100	38
70 Brix	1.35	1,650	364	70	21
		400	86.6	100	38
72 Brix	1.36	2,700	595	70	21
		640	138.6	100	38
74 Brix	1.376	5,500	1,210	70	21
		1,100	238	100	38
76 Brix	1.39	10,000	2,200	70	21
		2,000	440	100	38
Tars					
Tar, coke oven	1.12+	**3,000 to 8,000**	600 to 1,760	71	21
		650 to 1,400	140.7 to 308	100	38
Tar, gas house	1.16 to 1.30	**15,000 to 300,000**	3,300 to 66,000	70	21
		2,000 to 20,000	440 to 4,400	100	38

(Continued)

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TABLE 6 Viscosity of common liquids (*Continued*)
Typical SSU values as standardized by reputable authorities are shown in boldface type.

Liquid	Sp. gr. at 60°F (15.6°C)[a]	SSU	Centistokes	At °F	At °C
Tars					
Road tar					
Grade RT-2	1.07+	**200 to 300**	43.2 to 64.9	122	50
		55 to 60	8.77 to 10.22	212	100
Grade RT-4	1.08+	**400 to 700**	86.6 to 154	122	50
		65 to 75	11.63 to 14.28	212	100
Grade RT-6	1.09+	**1,000 to 2,000**	216.4 to 440	122	50
		85 to 125	16.83 to 26.2	212	100
Grade RT-8	1.13+	**3,000 to 8,000**	660 to 1,760	122	50
		150 to 225	31.8 to 48.3	212	100
Grade RT-10	1.14+	**20,000 to 60,000**	4,400 to 13,200	122	50
		250 to 400	53.7 to 86.6	212	100
Grade RT-12	1.15+	114,000 to 456,000	25,000 to 75,000	122	50
		500 to 800	108.2 to 173.2	212	100
Pine tar	1.06	2,500	559	100	38
		500	108.2	132	56
Miscellaneous					
Corn starch solutions					
22 Baumé	1.18	150	32.1	70	21
		130	27.5	100	38
24 Baumé	1.20	600	129.8	70	21
		440	95.2	100	38
25 Baumé	1.21	1,400	303	70	21
		800	173.2	100	38

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Ink, printers	1.00 to 1.38	2,500 to 10,000	550 to 2,200	100	38
		1,100 to 3,000	238.1 to 660	130	54
Tallow	0.918 avg.	56	9.07	212	100
Milk	1.02 to 1.05		1.13	68	20
Varnish, spar	0.9	1425	313	68	20
		650	143	100	38
Water, fresh	1.0		1.123	60	16
			0.516	130	54

[a]Unless otherwise noted

[b]Depends on origin or percent and type of solvent

Source: *Hydraulic Institute Engineering Data Book,* 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ.

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TABLE 7 Viscosity conversion table

This table gives an approximate comparison of ratings so that, if a viscosity is given in terms other than Saybolt seconds universal, it can be translated quickly by following horizontally to the SSU column.

Saybolt universal seconds SSU	Kinematic viscosity, cSt[a]	Approx. seconds, Mac Michael	Approx. Gardner Holt bubble	Seconds, Zahn cup 1	Seconds, Zahn cup 2	Seconds, Zahn cup 3	Seconds, Zahn cup 4	Seconds, Zahn cup 5	Seconds, Demmler cup 1	Seconds, Demmler cup 10	Approx. seconds, Stormer 100-g load	Seconds, Pratt and Lambert "F"
31	1.00	—	—	—	—	—	—	—	—	—	—	—
35	2.56	—	—	—	—	—	—	—	—	—	—	—
40	4.30	—	—	—	—	—	—	—	1.3	—	—	—
50	7.40	—	—	—	—	—	—	—	2.3	—	2.6	—
60	10.3	—	—	—	—	—	—	—	3.2	—	3.6	—
70	13.1	—	—	—	—	—	—	—	4.1	—	4.6	—
80	15.7	—	—	—	—	—	—	—	4.9	—	5.5	—
90	18.2	—	—	—	—	—	—	—	5.7	—	6.4	—
100	20.6	125	—	38	18	—	—	—	6.5	—	7.3	—
150	32.1	145	—	47	20	—	—	—	10.0	1.0	11.3	—
200	43.2	165	A	54	23	—	—	—	13.5	1.4	15.2	—
250	54.0	198	A	62	26	—	—	—	16.9	1.7	19	—
300	65.0	225	B	73	29	—	—	—	20.4	2.0	23	—
400	87.0	270	C	90	37	—	—	—	27.4	2.7	31	7
500	110.0	320	D	—	46	—	—	—	34.5	3.5	39	8
600	132	370	F	—	55	—	—	—	41	4.1	46	9
700	154	420	G	—	63	22.5	—	—	48	4.8	54	9.5
800	176	470	—	—	72	24.5	—	—	55	5.5	62	10.8
900	198	515	H	—	80	27	18	—	62	6.2	70	11.9
1000	220	570	I	—	88	29	20	138	69	6.9	77	12.4
1500	330	805	M	—	—	40	28	18	103	10.3	116	16.8
2000	440	1070	Q	—	—	51	34	24	137	13.7	154	22
2500	550	1325	T	—	—	63	41	29	172	17.2	193	27.6
3000	660	1690	U	—	—	75	48	33	206	20.6	232	33.7

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4000	880	2110	V	—	—	—	63	43	275	27.5	308	45
5000	1100	2635	W	—	—	—	77	50	344	34.4	385	55.8
6000	1320	3145	X	—	—	—	—	65	413	41.3	462	65.5
7000	1540	3670	—	—	—	—	—	75	481	48	540	77
8000	1760	4170	Y	—	—	—	—	86	550	55	618	89
9000	1980	4700	—	—	—	—	—	96	620	62	695	102
10,000	2200	5220	Z	—	—	—	—	—	690	69	770	113
15,000	3300	7720	Z2	—	—	—	—	—	1030	103	1160	172
20,000	4400	10500	Z3	—	—	—	—	—	1370	137	1540	234

[a]Kinematic viscosity (in centistokes) = $\frac{\text{absolute viscosity (in centipoises)}}{\text{density}}$

When the SI units centistokes and centipoises are used, the density is numerically equal to the specific gravity. Therefore, the following will be sufficiently accurate for most calculations:

Kinematic viscosity (in centistokes) = $\frac{\text{absolute viscosity (in centipoises)}}{\text{specific gravity}}$

When USCS units are used, the density must be used rather than the specific gravity.

Note: For values of 70 cSt and above, use the conversion

$$\text{SSU} = \text{centistokes} \times 4.635$$

Above the range of this table and within the range of the viscosimeter, multiply the particular value by the following approximate factors to convert to SSU:

Viscosimeter	Factor
Mac Michael	1.92 (approx.)
Demmler 1	14.6
Demmler 10	146
Stormer	13 (approx.)

Source: *Hydraulic Institute Engineering Data Book,* 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ.

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TABLE 8 Properties of water at temperatures from 40 to 705.4°F (4.4 to 374.1°C)

°F	°C	Specific volume[a], ft³/lb	Specific gravity 39.2°F reference	Specific gravity 60°F reference	Specific gravity 70°F reference	Wt[b], lb/ft³	Vapor pressure[c], lb/in²abs
40	4.4	0.01602	1.000	1.001	1.002	62.42	0.1217
50	10.0	0.01603	0.999	1.001	1.002	62.38	0.1781
60	15.6	0.01604	0.999	1.000	1.001	62.34	0.2563
70	21.1	0.01606	0.998	0.999	1.000	62.27	0.3631
80	26.7	0.01608	0.996	0.998	0.999	62.19	0.5069
90	32.2	0.01610	0.995	0.996	0.997	62.11	0.6982
100	37.8	0.01613	0.993	0.994	0.995	62.00	0.9492
120	48.9	0.01620	0.989	0.990	0.991	61.73	1.692
140	60.0	0.01629	0.983	0.985	0.986	61.39	2.889
160	71.1	0.01639	0.977	0.979	0.979	61.01	4.741
180	82.2	0.01651	0.970	0.972	0.973	60.57	7.510
200	93.3	0.01663	0.963	0.964	0.966	60.13	11.526
212	100.0	0.01672	0.958	0.959	0.960	59.81	14.696
220	104.4	0.01677	0.955	0.956	0.957	59.63	17.186
240	115.6	0.01692	0.947	0.948	0.949	59.10	24.97
260	126.7	0.01709	0.938	0.939	0.940	58.51	35.43
280	137.8	0.01726	0.928	0.929	0.930	58.00	49.20
300	148.9	0.01745	0.918	0.919	0.920	57.31	67.01
320	160.0	0.01765	0.908	0.909	0.910	56.66	89.66
340	171.1	0.01787	0.896	0.898	0.899	55.96	118.01
360	182.2	0.01811	0.885	0.886	0.887	55.22	153.04
380	193.3	0.01836	0.873	0.874	0.875	54.47	195.77
400	204.4	0.01864	0.859	0.860	0.862	53.65	247.31
420	215.6	0.01894	0.846	0.847	0.848	52.80	308.83
440	226.7	0.01926	0.832	0.833	0.834	51.92	381.59
460	237.8	0.0196	0.817	0.818	0.819	51.02	466.0
480	248.9	0.0200	0.801	0.802	0.803	50.00	566.1
500	260.0	0.0204	0.785	0.786	0.787	49.02	680.8
520	271.1	0.0209	0.765	0.766	0.767	47.85	812.4
540	282.2	0.0215	0.746	0.747	0.748	46.51	962.5
560	293.3	0.0221	0.726	0.727	0.728	45.3	1133.1
580	304.4	0.0228	0.703	0.704	0.704	43.9	1325.8
600	315.6	0.0236	0.678	0.679	0.680	42.3	1542.9
620	326.7	0.0247	0.649	0.650	0.650	40.5	1786.6
640	337.8	0.0260	0.617	0.618	0.618	38.5	2059.7
660	348.9	0.0278	0.577	0.577	0.578	36.0	2365.4
680	360.0	0.0305	0.525	0.526	0.527	32.8	2708.1
700	371.1	0.0369	0.434	0.435	0.435	27.1	3093.7
705.4	374.1	0.0503	0.319	0.319	0.320	19.9	3206.2

[a] 1 m^3/kg = 16.02 ft^3/lb

[b] 1 kg/m^3 = 0.06243 lb/ft^3

[c] 1 kPa = 0.145 lb/in^2; 1 bar = 14.5 lb/in^2

Source: *Hydraulic Institute Engineering Data Book*, 2nd Edition, 1990, Hydraulic Institute, Parsippany, NJ.

TABLE 9 Conversions of USCS to SI units

The first two digits of each numeral entry represent a power of 10. An asterisk follows each number that expresses an exact definition.

Acceleration

To convert from	to	multiply by
foot/second^2	meter/second^2	−01 3.048*
free fall, standard	meter/second^2	+00 9.806 65*
gal (galileo)	meter/second^2	−02 1.00*
inch/second^2	meter/second^2	−02 2.54*

Area

To convert from	to	multiply by
acre	meter^2	+03 4.046 856 4224*
circular mil	meter^2	−10 5.067 074 8
foot^2	meter^2	−02 9.290 304*
inch^2	meter^2	−04 6.4516*
mile^2 (U.S. statute)	meter^2	+06 2.589 988 110 336*
yard^2	meter^2	−01 8.361 273 6*

Density

To convert from	to	multiply by
gram/centimeter^3	kilogram/meter3	+03 1.00*
lbm/inch^3	kilogram/meter^3	+04 2.767 990 5
lbm/foot^3	kilogram/meter^3	+01 1.601 846 3
slug/foot^3	kilogram/meter^3	+02 5.153 79

Energy

To convert from	to	multiply by
British thermal unit (ISO/TC 12)	joule	+03 1.055 06
British thermal unit (International Steam Table)	joule	+03 1.055 04
British thermal unit (mean)	joule	+03 1.055 87
British thermal unit (thermochemical)	joule	+03 1.054 350 264 488
British thermal unit (39°F)	joule	+03 1.059 67
British thermal unit (60°F)	joule	+03 1.054 68
calorie (International Steam Table)	joule	+00 4.1868
calorie (mean)	joule	+00 4.190 02
calorie (thermochemical)	joule	+00 4.184*
calorie (15°C)	joule	+00 4.185 80

Length

To convert from	to	multiply by
fathom	meter	+00 1.8288*
foot	meter	−01 3.048*
foot (U.S. survey)	meter	+00 1200/3937*
foot (U.S. survey)	meter	−01 3.048 006 096
furlong	meter	+02 2.011 68*
inch	meter	−02 2.54*
league (U.K. nautical)	meter	03 5.559 552*
league (international nautical)	meter	+03 5.556*
league (statute)	meter	+03 4.828 032*
light-year	meter	+15 9.460 55
meter	wavelengths Kr 86	+06 1.650 763 73*
micron	meter	−06 1.00*
mil	meter	−05 2.54*
mile (U.S. statute)	meter	+03 1.609 344*
mile (U.K. nautical)	meter	+03 1.853 184*
mile (international nautical)	meter	+03 1.852*
mile (U.S. nautical)	meter	+03 1.852*
nautical mile (U.K.)	meter	+03 1.853 184*
nautical mile (international)	meter	+03 1.852*
nautical mile (U.S.)	meter	+03 1.852*
rod	meter	+00 5.0292*
statute mile (U.S.	meter	+03 1.609 344*
yard	meter	−01 9.144*

Mass

To convert from	to	multiply by
carat (metric)	kilogram	−04 2.00*
grain	kilogram	−05 6.479 891*
gram	kilogram	−03 1.00*
kgf second^2 meter (mass)	kilogram	+00 9.806 65*
kilogram mass	kilogram	+00 1.00*
lbm (pound mass, avoirdupois)	kilogram	−01 4.535 923 7*
ounce mass (avoirdupois)	kilogram	−02 2.834 952 312 5*
ounce mass (troy or apothecary)	kilogram	−02 3.110 347 68*
pennyweight	kilogram	−03 1.555 173 84*
pound mass, lbm (avoirdupois)	kilogram	−01 4.535 923 7*
pound mass (troy or apothecary)	kilogram	−01 3.732 417 216*
slug	kilogram	+01 1.459 390 29
ton (assay)	kilogram	−02 2.916 666 6
ton (long)	kilogram	+03 1.016 046 908 8*

(*Continued*)

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TABLE 9 Conversions of USCS to SI units (*Continued*)

The first two digits of each numeral entry represent a power of 10. An asterisk follows each number that expresses an exact definition.

Energy

To convert from	to	multiply by
calorie (20°C)	joule	+00 4.181 90
calorie (kilogram, International Steam Table)	joule	+03 4.1868
calorie (kilogram, mean)	joule	+03 4.190 02
calorie (kilogram, thermochemical)	joule	+03 4.184*
electronvolt	joule	−19 1.602 10
erg	joule	−07 1.00*
foot pound force (ft • lbf)	joule	+00 1.355 817 9
foot poundal	joule	−02 4.214 011 0
joule (international of 1948)	joule	+00 1.000 165
kilocalorie (International Steam Table)	joule	+03 4.1868
kilocalorie (mean)	joule	+03 4.190 02
kilocalorie (thermochemical)	joule	+03 4.184*
kilowatthour	joule	+06 3.60*
kilowatthour (international of 1948)	joule	+06 3.600 59
ton (nuclear equivalent of TNT)	joule	+09 4.20
watthour	joule	+03 3.60*

Energy/Area Time

To convert from	to	multiply by
Btu (thermochemical)/foot^2 second	watt/meter^2	+04 1.134 893 1
Btu (thermochemical)/foot^2 minute	watt/meter^2	+02 1.891 488 5
Btu (thermochemical)/foot^2 hour	watt/meter^2	+00 3.152 480 8
Btu (thermochemical)/inch^2 second	watt/meter^2	+06 1.634 246 2
calorie (thermochemical)/cm^2 minute	watt/meter^2	+02 6.973 333 3
erg/centimeter^2 second	watt/meter^2	−03 1.00*
watt/centimeter^2	watt/meter^2	+04 1.00*

Force

To convert from	to	multiply by
dyne	newton	−05 1.00*
kilogram force (kgf)	newton	+00 9.806 65*
kilopond force	newton	+00 9.806 65*
kip	newton	+03 4.448 221 615 260 5*
lbf (pound force, avoirdupois)	newton	+00 4.448 221 615 260 5*
ounce force (avoirdupois)	newton	−01 2.780 138 5
pound force lbf (avoirdupois)	newton	+00 4.448 221 615 260 5*
poundal	newton	−01 1.382 549 543 76*

Mass

To convert from	to	multiply by
ton (metric)	kilogram	+03 1.00*
ton (short, 2000 pound)	kilogram	+02 9.071 847 4*

Power

To convert from	to	multiply by
Btu (thermochemical)/second	watt	+03 1.054 350 264 488
Btu (thermochemical)/minute	watt	+01 1.757 250 4
calorie (thermochemical)/second	watt	+00 4.184*
calorie (thermochemical)/minute	watt	−02 6.973 333 3
foot lbf/hour	watt	−04 3.766 161 0
foot lbf/minute	watt	−02 2.259 696 6
foot lbf/second	watt	+00 1.355 817 9
horsepower (550 foot lbf/second)	watt	+02 7.456 998 7
horsepower (boiler)	watt	+03 9.809 50
horsepower (electric)	watt	+02 7.46*
horsepower (metric)	watt	+02 7.354 99
horsepower (U.K.)	watt	+02 7.457
horsepower (water)	watt	+02 7.460 43
kilocalorie (thermochemical)/minute	watt	+01 6.973 333 3
kilocalorie (thermochemical)/second	watt	+03 4.184*
watt (international of 1948)	watt	+00 1.000 165

Pressure

To convert from	to	multiply by
atmosphere	newton/meter^2	+05 1.013 25*
bar	newton/meter^2	+05 1.00*
centimeter of mercury (0°C)	newton/meter^2	+03 1.333 22
centimeter of water (4°C)	newton/meter^2	+01 9.806 38
dyne/centimeter^2	newton/meter^2	−01 1.00*
foot of water (39.2°F)	newton/meter^2	+03 2.988 98
inch of mercury (32°F)	newton/meter^2	+03 3.386 389
inch of mercury (60°F)	newton/meter^2	+03 3.376 85
inch of water (39.2°F)	newton/meter^2	+02 2.490 82
inch of water (60°F)	newton/meter^2	+02 2.4884
kgf/centimeter^2	newton/meter^2	+04 9.806 65*
kgf/meter^2	newton/meter^2	+00 9.806 65*
lbf/foot^2	newton/meter^2	+01 4.788 025 8

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Pressure

To convert from	to	multiply by
lbf/inch^2 (psi)	newton/meter^2	+03 6.894 757 2
millibar	newton/meter^2	+02 1.00*
millimeter of mercury (0°C)	newton/meter^2	+02 1.333 224
pascal	newton/meter^2	+00 1.00*
psi (lbf/inch^2)	newton/meter^2	+03 6.894 757 2

Speed

To convert from	to	multiply by
foot/hour	meter/second	−05 8.466 666 6
foot/minute	meter/second	−03 5.08*
foot/second	meter/second	−01 3.048*
inch/second	meter/second	−02 2.54*
kilometer/hour	meter/second	−01 2.777 777 8
knot (international)	meter/second	−01 5.144 444 444
mile/hour (U.S. statute)	meter/second	−01 4.4704*
mile/minute (U.S. statute)	meter/second	+01 2.682 24*
mile/second (U.S. statute)	meter/second	+03 1.609 344*

Temperature

To convert from	to	proceed as follows
Celsius	Kelvin	$t_K = t_C + 273.15$
Fahrenheit	Kelvin	$t_K = (\frac{5}{9})(t_F + 459.67)$
Fahrenheit	Celsius	$t_C = (\frac{5}{9})(t_F - 32)$
Rankine	Kelvin	$t_K = (\frac{5}{9})t_R$

Viscosity

To convert from	to	multiply by
centistoke	meter^2/second	−06 1.00*
stoke	meter^2/second	−04 1.00*
foot^2/second	meter^2/second	−02 9.290 304*
centipoise	newton second/meter^3	−03 1.00*
lbm/foot second	newton second/meter^2	+00 1.488 163 9
lbf second/foot^2	newton second/meter^2	+01 4.788 025 8

Viscosity

To convert from	to	multiply by
poise	newton second/meter^2	−01 1.00*
poundal second/foot^2	newton second/meter^2	+00 1.488 163 9
slug/foot second	newton second/meter^2	+01 4.788 025 8
rhe	meter^2/newton second	+01 1.00*

Volume

To convert from	to	multiply by
acre foot	meter^3	+03 1.233 481 9
barrel (petroleum, 42 gallons)	meter^3	−01 1.589 873
bushel (U.S.)	meter^3	−02 3.523 907 016 688*
cord	meter^3	+00 3.624 556 3
cup	meter^3	−04 2.365 882 365*
dram (U.S. fluid)	meter^3	−06 3.696 691 195 312 5*
fluid ounce (U.S.)	meter^3	−05 2.957 352 956 25*
foot^3	meter^3	−02 2.831 684 659 2*
gallon (U.K. liquid)	meter^3	−03 4.546 087
gallon (U.S. dry)	meter^3	−03 4.404 883 770 86*
gallon (U.S. liquid)	meter^3	−03 3.785 411 784*
gill (U.S.)	meter^3	−04 1.182 941 2
inch^3	meter^3	−05 1.638 706 4*
liter	meter^3	−03 1.00*
ounce (U.S. fluid)	meter^3	−05 2.957 352 956 25*
peck (U.S.)	meter^3	−03 8.809 767 541 72*
pint (U.S. dry)	meter^3	−04 5.506 104 713 575*
pint (U.S. liquid)	meter^3	−04 4.731 764 73*
quart (U.S. dry)	meter^3	−03 1.101 220 942 715*
quart (U.S. liquid)	meter^3	−04 9.463 529 5
tablespoon	meter^3	−05 1.478 676 478 125*
teaspoon	meter^3	−06 4.928 921 593 75*
ton (register)	meter^3	+00 2.831 684 659 2*
yard^3	meter^3	−01 7.645 548 579 84*

Volume/Unit Time

To convert from	to	multiply by
gallon/minute	meter^3/second	−05 6.309 020
gallon/minute	meter^3/hour	−01 2.271 247
gallon/minute	liter/second	−02 6.309 020

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