ASPE Handbook

PUMPS & PUMP SYSTEMS

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DEFINITIONS

To more easily understand this book the terms listed below shall have the following meanings. No attempt is made to define ordinary words which are used in accordance with their established dictionary meaning, except where necessary to avoid misunderstanding.

Absolute Pressure. The sum of gage pressure and atmospheric pressure.

Atmospheric Pressure. The force exerted on a unit area by the weight of the atmosphere. Pressure at sea level due to the atmosphere is 14.7 psi.

Available N.P.S.H. The inherent energy in a liquid at the suction connection of a pump.

Axial Flow. Most of the pressure is developed by propelling or lifting action of the vanes on the liquid. The flow enters axially and discharges nearly axially.

Bernoulli's Theorem. Assuming no friction losses or the performance of extra work, the sum of three types of energy (heads) at any point in a system is the same in any other point in the system.

Brake Horsepower. Total power required by pump to do a specified amount of work.

Cavitation. When pressure in suction line falls below vapor pressure, vapor is formed and moves along with the stream. The collapse of vapor bubbles creates noise and vibration, (i.e. the cavitation).

Design Working Head. Head which must be available in the system at a specified location to satisfy design requirements.

Density. The weight per unit volume of a substance.

Diffuser. A point just before the tongue of casing where all the liquid has been discharged from the impeller. It is the final outlet of the pump.

Flat Head Curve. The head rises slightly as the flow is reduced. As with steepness, the magnitude of flatness is a relative term.

Fluid. A substance which in static equilibrium cannot sustain tangential or shear forces.

Friction Head. The rubbing of water particles against each other and against the walls of the pipe. This friction causes a pressure loss in the flow line, called the friction head.

Gage Pressure. The difference between a given pressure and that of the atmosphere.

Head. Energy per pound of fluid. Commonly used to represent the vertical height of a static column of liquid corresponding to the pressure of a fluid at the point in question.

Horsepower. Power delivered while doing work at the rate of 500 ft-lb per second or 33,000 ft-lb per minute.

Hydraulics. The study of fluids at rest and in motion.

Independent Head. Consists of head that does not change with flow, such as static head and minimum pressure at end of system.

KVA. The total power required to drive a motor.

Mechanical Efficiency. Ratio of power output to power input.

Mechanical Seals. Acts as a check valve to keep liquid from escaping from the pumping system. It also works as a slider bearing.

Mixed Flow. Pressure is developed partly by centrifugal force and partly by the lift of the vanes on the liquid. The flow enters axially and discharges in an axial and radial direction.

Multistage Pumps. In order to give high pressure, two or more impellers and casings can be assembled on one shaft as a single unit, forming a multistage pump. The discharge from the first stage; discharge from the second stage enters the suction of the third, and so on. The pump capacity is the rating in gallons per minute of one stage; the pressure rating is the sum of the pressure ratings of the individual stages, minus a small head loss.

Net Positive Suction Head. Amount of energy in the liquid at the pump datum. It must be defined to having meaning, as either available or required N.P.S.H.

Packing. A soft semiplastic material cut in rings and snugly fits around the shaft or shaft sleeve.

Potential Head. Energy position measured by work possible in dropping vertical distance.

Pressure. The force exerted per unit area of a fluid. The most common unit for designating pressure is pounds per square inch (psi).

Pump Affinity Laws. Pump affinity laws state that flow is proportional to impeller peripheral velocity, and head is proportional to the square of the peripheral velocity.

Pump in Parallel. The head for each pump must equal the system head and the sum of the individual pump capacities equal the system flow rate at the system head.

Pump in series. The total head/capacity characteristic curve for the two pumps in series can be obtained by adding the total heads of the individual pumps for various capacities.

Pump Performance Curve. Describes head horsepower, efficiency and the net positive suction head required for proper pump operation.

Radial Flow. Pressure is developed principally by centrifugal force action. Liquid normally enters the impeller at the hub and flows radially to the periphery.

Required N.P.S.H. The energy in the liquid a pump must have to operate satisfactorily.

Shut off BHP. One half full load BHP is normally considered to be the Shut off BHP.

Slip. Loss in delivery due to the escape of liquid inside pump from discharge to suction.

Specific Gravity. The ratio of a substance density to that of some standard substance.

Specific Speed. Specific speed is an index number which correlates pump capacity, head and speed.

Static Pressure Head. Energy per pound due to pressure. The height a liquid can be raised by a given pressure.

Static Suction Head. When supply source is above the pump. Vertical distance from free surface of liquid to pump datum.

Static Suction Lift. When supply source is below the pump. Vertical distance from free surface of liquid to pump datum.

Steep Head Curve. Head rises steeply and continuously as the flow is reduced.

Suction Specific Speed. An index number describing suction characteristics of a given impeller.

System Head Curve. A plot of system head versus system flow. System head varies with flow since friction and velocity head are both a function of flow.

Total Head. Net difference between total suction and discharge heads. The measure of energy increase imparted to the liquid by the pump.

Utility Horsepower (UHP). Utility horsepower (UHP) is simply BHP divided by drive efficiency.

Vacuum. Pressure below atmospheric pressure.

Vapor Pressure. The pressure at which a pure liquid can exist in equilibrium with its vapor at a specified temperature.

Variable Speed Pressure Booster Pumps. Variable speed pressure booster pumps are used to reduce power consumption to maintain a constant building supply pressure by varying pump speeds through coupling or mechanical devices.

Velocity Head. Kinetic energy per pound. The vertical distance a liquid would have to fall to acquire the velocity "V".

Viscosity. The internal friction of a liquid, or that property which resists any force tending to produce flow.

Water Hamer. The result of a strong pressure wave in a liquid caused by an abrupt change in flow rate.

Water Horsepower. Power required by pump motor for pumping only.

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CHAPTER I

PUMP SELECTION

1.1 Introduction

Pump selection is the process of choosing a pump for a particular application which will furnish quiet cavitation and maintenance-free performance over a long period of time. This, of course, includes future needs. Pump selection is not difficult, but proper pump selection does require some practice.

Generally speaking, the duty point for a pump application should be spotted on the performance curve as close to the best efficiency point as possible. The best efficiency point is where the pump operates the most quietly and experiences the most maintenance-free life. However, most pumps are chosen to operate at duties other than at the best efficiency point. In the event that a duty point cannot be spotted near the best efficiency point of any pump, it is wise to make a selection with the duty point falling to the left of the best efficiency point. Such selection will insure more successful operation due to (1) quieter operation; (2) longer maintenance-free life; and (3) less chance of cavitation.

A centrifugal pump normally lasts for many years and, too often, the day comes when the pump cannot meet increased demand. A larger pump must be purchased, and the new pump may require a new mounting pad, pipe change, electrical equipment changes, etc. Such an undesirable situation may be eliminated by initially planning for future pump needs.

The success of a pumping installation depends largely on the competency of the specification writer and the skill of the person who evaluates the quotations. A good installation is not necessarily the lowest initial cost but the lowest capital and operating cost over the economic life of the equipment, together with performance, reliability and freedom from down time.

1.2 Pump Selection And Building Owner

Historically, an owner's primary interest has been focused on first cost and low maintenance. Now, with the high cost of energy, total annual owning and operating expenses are becoming the dominating concerns when selecting motors.

When energy was relatively cheaper, most pumps were oversized. With rising fuel costs, the current emphasis is for owners to select pumps on the basis of their application. Therefore, when the application changes, the pump-type often changes too.

Another prevailing trend is utilizing a more sophisticated network analysis, often involving computer simulations. This allows for more reliability in pump selection, including use of variable speed, primary and/or secondary pumping systems.

When pump load varies and the pump motor is constantly in operation, sometimes just being on-line can be inefficient. Methods of determining the most efficient combination of pump selection, including the hand-calculator Hardy-Cross method, often can point out these inefficiencies.

Another area that is receiving greater owner attention is preventive maintenance. Monitoring and operating methods are being more closely watched.

Moving liquids from one place to another, similar to mechanical handling, adds cost to the product, but nothing to its value. Therefore, it should be done as cheaply as possible.

As previously mentioned, the overall cost of pumping is not confined to the initial capital cost, but consists of: installed cost-amortization; power or fuel cost; supervision and maintenance costs; and the cost of down-time or standby equipment.

If a continuous pumping system with 100 percent reliability is essential, standby unity must be provided. If the service is not critical and some down-time can be accommodated, a less exotic pumping system may be justified, particularly if pumping cost increases can be written off with some tax relief.

The cost of borrowed money has an important bearing on the design of the most economical scheme. With rising inflation and higher labor costs, a pumping system designed for an economical life of 20 years or more will invariably be a better proposition if the operating costs can be kept to a minimum, even at the expense of a higher initial capital investment.

If, however, borrowed capital is difficult to obtain, and the economy appears to be in a tight anti-inflationary depression and labor costs are likely to be fairly stable, higher maintenance costs may be acceptable on a short-term basis.

This approach is particularly true in underdeveloped countries where there is an abundant amount of cheap labor, where governments are anxious to increase employment and where foreign capital is extremely difficult to obtain.

The owner is responsible for the money authorized to spend. It is important, therefore, that the owner's cash flow position is made known to the engineer before the design is commenced. A convenient way of doing this is for the engineers to prepare a concept report of his proposed scheme in sufficient detail to enable the owner to understand the various implications involved and to advise the engineer of the financing details.

For example, there is not much logic in saving $20,000 or more in capital cost of a process industry at the expense of reliability, if down-time consists of several thousand dollars per hour.

Overall, owners will not tolerate use of oversized systems from designers. This means designers will have to be more knowledgeable about, and sensitive to, owner's concerns.

1.3 Pump Selection and Pump Manufacturer

In a competitive marketplace, manufacturers must present their design features and benefits reasonably. Because of stiff competition, it has become more and more difficult for manufacturers to provide the installation services. Resources are often stretched to their maximum limit.

One result is the increasing dominance of prepackaged systems, which has both advantages and disadvantages for the manufacturer and designer.

Also, in order to remain in business, a pump manufacturer must sell equipment. Since a pump is not regarded as a consumable item, a sale lost today is lost forever. On the other hand, a successful, conscientious pump supplier can usually look forward to a continuing business in spare parts and, when extensions to the plant are required, additional pumps.

Buyers are frequently obliged to accept the lowest bidder. It is sometimes difficult, without specific adverse experience with a particular manufacturer's product, to justify a more expensive unit which will, in the engineer's opinion, give better performance during its economic life.

The initial cost of a pumping unit, including motor or engine drive, can be kept to a minimum. A high impeller speed is possible; however, the maintenance cost is increased considerably. In the past, most pump and engine manufacturers have built slow-speed units. Depending on their capacity, pumps would operate at motor synchronous speeds of 900, 1200 and 1800 rpms, and with engine drives at even lower speeds.

Unfortunately, slow-speed pumps are no longer readily available since their intitial cost is higher and competition for the lowest bid forced them off the market. Some manufacturers have retained their old patterns and will still cast impellers and bowl assemblies as spare parts for existing pumps, but they are reluctant to market new pumps in the low-speed range. The frequency of spare-part replacement for slow-speed pumps is considerable less.

When the prices of spare impellers and bowl assemblies are compared to the initial cost of the pump, it becomes obvious that today's pump manufacturers must rely on the spare-part service for much of their business.

It is the same old story of built-in obsolescence. If pumps lasted indefinitely, the majority of pump manufacturers would soon be out of business.

It is frequently said that, with modern designs and with manufacturing techniques using new alloys and better materials, higher shaft speeds are perfectly satisfactory. These statements are valid only under ideal operating conditions.

Once the machine is subjected to wear, corrosion or erosion, the inevitable result is misalignment and vibration. The smooth, quiet-running pump becomes a veritable grinding machine, accentuated by its higher rotative speed. The frequency of repairs quickly absorbs any savings in the initial cost. Unfortunately, the magnitude of these costs is rarely appreciated unless the owner is aware of the problem and keeps accurate maintenance and operating costs against each item of equipment.

1.4 Consulting Engineer and Pump Selection

A consulting engineer's job is to study the problem and design the best scheme for achieveing the required result. He or she must consider pump hydraulics, power and fuel supplies, detailed pump design and, finally, capital and operating costs.

It is possible that initially four or five schemes will evolve with apparently equal merit. Then, it is necessary to develp each scheme in greater detail to resolve one or two viable schemes.

During preliminary studies, discussions should take place between the engineers and the pump suppliers to determine what equipment is available. Estimating prices will help considerably in determining the optimum scheme, but the final option will not be arrived until the final invitations to tender are issued, and the quotations are received.

The engineer must give the pump suppliers as much data and general information as possible to ensure that the final quotations are representative of the best the industry has to offer.

Each supplier is asked to complete the data sheet and submit it with his quotation, so that all submissions can be evaluated on equal terms. Unfortunately, some pump suppliers are reluctant to do this, and will submit the quotations only on their own standard formats.

In order to present a meaningful comparison, engineers must prepare a tabulation sheet containing all the relevant data. If the engineer has to search through the pump supplier's formal quotation to uncover the informations needed, there is a possibility that some data will be missing.

Consulting engineers are not usually a pump designer. They cannot be knowledgeable of all the new techniques available to pump manufactures. The majority of pump suppliers will welcome a general outline of the engineer's requirements in addition to the broad parameters of design, which will include the following:

- Head
- Capacity
- Available New Positive Suction Head (NPSH)
- Preferred shaft speed
- System head curve parameters
- Horse power characteristics
- Specific speed with respect to cavitation should be stated

- Suction conditions, including limits of submergence, suction head or suction lift
- Drives, electric motor or engine

In addition to the numerical data, reference should be made to specific applicable design codes.

- American Standard for Vertical Turbine Pumps
- Hydraulic Institute Standards
- Centrifugal Fire Pumps

Having provided the pump supplier with all the compliance parameters, the specific details of the pump's design should be left to the engineer. The completed questionnaire will inform the engineers of the quality of the equipment being offered.

Efficiency of energy conversion is a prime consideration in municipal or domestic installations since power is often the largest operation cost item. The annual power costs of each efficiency point can be calculated, and should be made known to the suppliers in the Invitations to Tender.

Checking the pump tests and the receipt of the certified performance curves ensures the engineer the pumps are capable of the required performance, but it is of little value to purchase pumps on the basis of their good efficiency characteristics if they cannot be maintained in practice.

The service facilities of the successful manufacturer and distribution agents are of considerable interest to the owner. Equipment manufactured overseas must be adequately serviced by a large stock of spare parts carried in this country, since overseas servicing arrangements can be unpredictable.

1.5 Number of Pump Units

Multiple pump systems save energy by allowing a small lead pump to maintain pressure during low flow periods. When a small lead pump or a jockey is used, it should be sized for a minimum requirement of 50 gpm. It would not be a good design practice to have a larger pump to be energized each time a fixture was used.

Duty point alone must not decide pump selection. If an impeller is more efficient at duty point, but would require a larger motor to prevent overload at run out, a smaller horsepower at run out would likely use less energy. Motors consume less power per unit of work at full load.

A three-pump system using idential electrically driven pumps, each capable of supplying 50 percent of the maximum demand, is a popular arrangement. The power supply only needs to be capable of operating two pumps at any one time. Sometimes, a three-pump system should be increased to a four-pump system on the basis that there will be two pumps in operation, one on standby, and one down for maintenance. The economics of a four-pump system, each capable of 33½ percent of the maximum demand, should also be considered.

If the power supply is subject to frequent failure, and continuous pumping operation is essential - for example, a fire pump station or a sewage lift station. Engine-driven pumps or a standby diesel generator should be considered.

If water demand is variable, variable speed pumps are required. The use of wound rotor motors with Flomatcher controls has some advantages, since the starting current for a wound rotor motor is equal to, or less than, the full load current.

In an endeavor to conserve power and to ensure that each pump is operating at its best efficiency, a "Cascade System" consisting of three or more pumps of various capacities, but with the same total developed head and all capable of paralleling together, has frequently been installed. An automatic controller is provided to select the optimum combination of units to suit the water demand. Unfortunately, certain pumps in the series seem to operate most of the time, while others are idle. For most installations, particularly for smaller systems, it is better to have all pumps the same capacity. The system is more flexible to meet water demand, maintenance is easier and wear can be uniform.

1.6 Economics

The economic evaluation of different water pumping systems for a public building is necessary to achieve the optimum type of water system with a proper balance between first costs and operating expenses.

The first cost of all water system components must be included in the evaluation (i.e. piping, tanks, pumps, electrical service and controls), along with the cost of building provisions - structurally and spacewise. Operating costs consist primarily of electrical energy charges on electric motor-driven pumps.

This demonstrates the complexity of water system evaluation, which will vary with building type and use. A number of design conditions concerning the actual building must be determined before the water pumping system evaluation can be initiated. These include:

- Type of water supply - well, water main or open tank (reservoir)
- Water demand - minimum and maximum flow rate in the system
- Load profiles - hours per day, and days per week
- Head losses - building pipe and valve friction, including meter and backflow preventor losses
- Code requirements - minimum flow capabilities and fire prevention storage needs
- Electrical charges - demand and commodity charges, as well as the availability of off-peak reductions (i.e. night rates). Trends in electrical charges for the past 10 years and projections for the next 20 years should be secured from the electric utility

Most of these design conditions are easy to determine for a specific building, with the one exception being minimum and maximum flow rates. Hunter's curve, using fixture units, has been criticized severely, but to this day no technical society has brought forward any data that can supplant it on all buildings.

Some consulting engineers, with extensive experience with a specific building-type, have developed their own criteria for water usage in that building. There have been attempts by pump manufacturers to provide data based upon specific studies conducted on serveral buildings. Such data should be rejected because there are too many variables in a building's water usage, including socio-economic factors. It is urged that the involved technical societies work together with the Bureau of Standards to produce a more exact method of predicting water usage in a building than Hunter's curve. If Hunter's curve is used, no safety factors should be applied to the resulting flow in gpm.

The total pump head for buildings is not difficult to determine, but frequently not enough time is taken to carefully design the head required for a specific building. Short cuts, such as taking 10 percent of the static head of the buildings for pipe friction or 50 percent of the pipe friction for fitting and valve losses, should be avoided. They will produce erroneous friction heads.

The pipe fittings to the farthest run should be counted, and friction losses determined in accordance with the Hydraulic Institute's method of calculating friction losses in pipe fittings. Also, the losses of water meters and backflow preventers should be determined as carefully as possible.

Added to the above technical design conditions must be financial requirements established by the building owner. The most significant factor of these is the amortization period allowed by the owner for increased first cost to achieve lower operating costs.

The speculative builder's decision is going to be different from that of an institutional owner, who may allow 20 to 30 years to amortize first costs.

The last consideration to be taken in the water pumping system evaluation is the fact that the water usage and friction cannot be determined exactly. Therefore, the design of the water system should allow operating pumps and controls to produce the flow and head actually required by the buildings, not the calculated design flow and head.

An additional condition necessitating operation at actual flow and head conditions is the fact that pump friction is calculated normally for 20-year-old pipe, new pipe friction being 50 to 60 percent of this.

1.7 References

1. "Pump Selection," by Rodger Walker, Ann Arbor Science, pp. 17-19, including Figure 5, 1982. Edited by permission.

2. "Water Pumping System Economic Evaluation," by James B. Rishel, ASSE Annual Meeting, Oct. 15, 1978. (Figures 1-5 and Table 1). Edited with permission.

CHAPTER 2

SUCTION AND DISCHARGE CONDITIONS

2.1 Pressure

The force exerted per unit area of a fluid. It can be considered a compressive stress. The most common unit for designating pressure is pounds per square inch (psi). According to Pascal's principle, if pressure is applied to the surface of a fluid, this pressure is transmitted undiminished in all directions.

PRESSURES

- Gage Pressure
- Atmospheric Pressure
- Absolute Pressure
  Gage + Atmosphere = Absolute
- 1 Atmosphere = 14.7 psi = 34 ft. water
  34/14.7 = 2.31
- PSI = (Head in Feet/2.31) x SP.GR.

Atmospheric pressure is the force exerted on a unit area by the weight of the atmosphere. The pressure at sea level due to atmosphere is 14.7.

Gage pressure is a corrected pressure and is the difference between a given pressure and that of the atmosphere.

The sum of gage pressure and atmospheric pressure is absolute pressure. The absolute pressure in a perfect vacuum is zero. Absolute pressure of the atmosphere at sea level is 14.7 psi (0 psi gage).

The term vacuum is frequently used in referring to pressures below atmospheric. Due to the common use of a column of mercury to measure vacuum, units are expressed in inches of mercury. (14.7 psi atmospheric pressure equals 30 inches of mercury.)

Since water weighs 0.0361 pounds per cubic inch, a column of water one square inch in area and one foot high will weigh 0.433 pounds. To increase the pressure 1 psi requires 2.31 feet increase in depth.

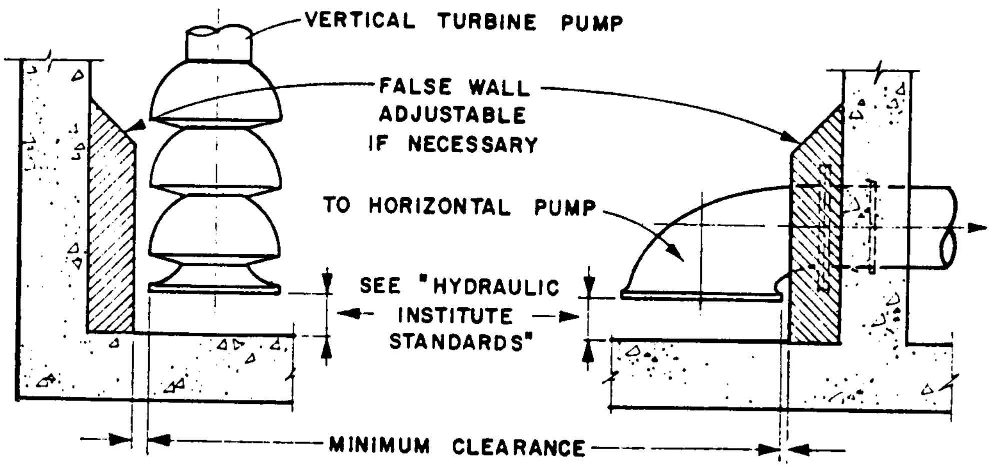
While discussing various types of pressures, one should consider vapor pressure. The vapor pressure of a liquid at a specified temperature is the pressure at which the liquid is in equilibrium with the atmosphere or with its vapor in a closed container. At pressures below this vapor pressure at a given temperature, the liquid will start to vaporize due to the reduction in pressure at the surface of the liquid. (At 50 degrees Fahrenheit, the vapor pressure of water is 0.256 psi. At 212 degrees Fahrenheit, it is 14.7 psi.)

2.2 Intake sump and protective screens

Care must be exercised in the design of intake sumps to avoid localized velocities that might create vortex formation. Vortices can reduce capacity and cause noise, vibration and possible damage to the pump. The Hydraulic Institute Standards contain recommendations on intake sump designs.

In designing a intake sump, it is important to ensure that one side of the suction bell-mouth is close to one wall of the pump chamber and that the bottom opening is reasonably close to the floor in accordance with the "Hydraulic Institute Standards."

Protective screens should be provided whenever there is suspended or floating debris entering the pump suction. Wire screens bolted or welded directly onto the suction bowl as protection devices are prohibited. These devices create serious suction prob lems if they become plugged, and they may corrode, fail and be drawn into the pump suction. This may eventually cause damage to the bowl assembly.



Pump inlet design.

2.3 Piping and Valves

Suction and discharge piping should be sized so that velocities are not excessive. Velocities of 5 ft/sec in suction piping and 8 ft/sec in discharge piping are reasonable maximums. Piping should have sufficient flexibility and be adequately supported so that no stresses are transmitted to the pump. Expansion joints or couplings that do not provide an axially restrained connection should not be used between the pump and a point of anchorage in the piping. Such an installation causes the hydraulic reaction of the pump to be carried by the pump, pump base and anchor bolts. This force could be of a magnitude to govern the structural design of the pump, and could make the construction economically infeasible.

Valves should be installed in the suction and discharge of the pumps to permit isolating the pump for maintenance and to control flow. The type of check valve required depends on the piping system into which the pump discharges. Ordinary swing check valves may be adequate, but in many systems power-operated, stop-check valves are needed to control surge or water hammer. If the pump is to be started and stopped against closed valves to control transient surge pressures, power-operated butterfly or ball valves interlocked with the pump start-stop controls can be used.

2.4 Controls

Pumps can be controlled by level, pressure or flow. Level controls can start and stop pumps at predetermined water levels in storage reservoirs or tanks. Controls for variable-speed pumps can be used to vary the pump discharge to maintain a predetermined level, pressure or flow.

Flow control is used to meet a fluctuating demand by varying the speed of the pumping unit. Sometimes flow control is provided by throttling. However, it is usually more economical to vary the pump speed.

Emergency controls are essential to good pump station design. Low-level shutoff should be provided in sumps, and low-pressure shut-off in suction lines, to prevent damage to pumping units by running them dry. High-level alarms should be provided in basins or storage tanks to indicate malfunctions and abnormal conditions.

2.5 Velocity Head

Velocity head represents the kinetic energy per unit weight that exists at a particular point. If velocity at a cross section were uniform, then the velocity head calculated with this uniform or average velocity would be the true kinetic energy per unit weight of fluid. But, in general, velocity distribution is not uniform. True kinetic energy is found by integrating the differential kinetic energies from streamline to streamline. The kinetic energy correction factor α to be applied to the $V_{av}^2/2g$ term is given by the expression

$$\alpha = {}^1/A \int_A (v/V)^3 \, dA$$

where
- V = average velocity in the cross section
- v = velocity at any point in the cross section
- A = area of the cross section

Studies indicate that $\alpha = 1.0$ for uniform distribution of velocity, $\alpha = 1.02$ to 1.15 for turbulent flows, and $\alpha = 2.00$ for laminar flow. In most fluid mechanics computations, α is taken as 1.0, without serious error being introduced into the result, since the velocity head is generally a small percentage of the total head (energy).

If the two velocity heads are unknown, relate them to each other by means of the equation of continuity.

$$Q = A_1 V_1 = A_2 V_2 = \text{constant (in ft}^3\text{/sec or gpm)}$$

where A_1 and V_1 are respectively the cross sectional area in ft^2 and the average velocity of the stream in ft/sec at Station 1, with similar terms for Station 2. Units of flow commonly used are cubic feet per second (cfs), although gallons per minute (gpm) and million gallons per day (mgd) are used in water supply work.

2.6 Net Positive Suction Head (NPSH)

NPSH is the total suction head in feet absolute, determined at the suction nozzle and corrected to pump datum, less the vapor pressure of the liquid in feet absolute. In other words, it is an analysis of energy conditions at the suction side of a pump to determine if the liquid will vaporize at the lowest pressure point in the pump. It should be stressed that only absolute pressures are used in all calculations to determine NPSH. To convert gage pressure (psig or psi) to feet absolute, add the barometric pressure (14.7 psi at sea level) to the liquid psi to obtain psi absolute and then multiply by 2.31.

The vapor pressure is a unique characteristic of every fluid and increases with increasing temperature (see Figure 2-1 for the vapor pressure of water). When the vapor pressure of the fluid reaches the pressure of the surrounding medium, the fluid begins to vaporize. The temperature at which this vaporization occurs decreases as the pressure of the surrounding medium decreases.

If a fluid is to be effectively pumped, it must be kept in a fluid state. Required NPSH is a measure of the suction head required to prevent vaporization at the lowest pressure point in the pump and NPSH available is a measure of the actual suction pressure provided by the system.

<u>NPSH required</u>

NPSH required is a function of pump design. As the liquid

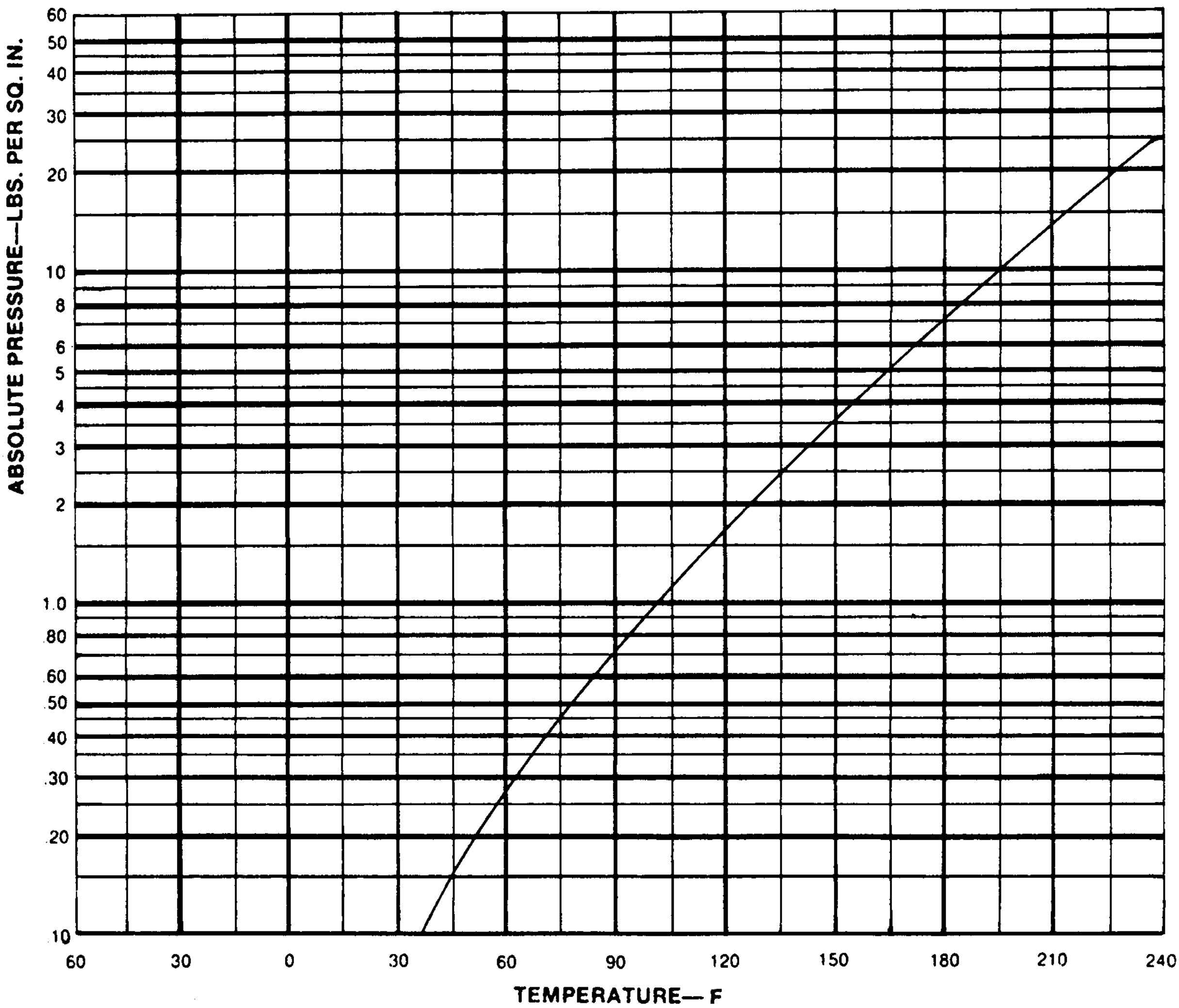


Figure 2-1. Vapor Pressure of Water.

flows from the pump suction to the eye of the impeller or vane, velocity increases and pressure decreases. Additional pressure losses occur due to shock and turbulence as the liquid strikes the impeller or vane. As the impeller or gear rotates, centrifugal force increases the velocity with a further decrease in the liquid pressure. The NPSH required is the positive head, in feet absolute, required at the pump suction to overcome these pressure drops in the pump and maintain pressure of the liquid above its vapor pressure. NPSH required varies with pump design, pump size, and operating conditions, and is supplied by the pump manufacturer.

In a centrifugal pump, required NPSH is that amount of energy (in feet of liquid) required:

1. To overcome friction losses from the suction opening to the impeller vanes.

2. To create the desired velocity of flow into the vanes.

In a rotary pump, required NPSH is that amount of energy (in psi) required:

1. To overcome friction losses from the suction opening into the gears or vanes.

2. To create the desired velocity of flow into the gears or vanes.

Available NPSH

Available NPSH is a characteristic of the system and is defined as the energy in a liquid at the suction connection of the pump (regardless of the pump type), over and above that energy in the liquid due to its vapor pressure. In other words, it is the excess pressure of the liquid, in feet absolute, over its vapor pressure at the pump suction. Figure 2-2 illustrates four typical suction conditions and the applicable $NPSH_A$ formula for each.

Since a liquid may have three types of energy, and since NPSH is an energy term, the two methods of determining available NPSH should take potential, pressure and kinetic energy into account.

To determine available NPSH by calculation of system head, consider the energy at station 1 of Figure 2(a):

$$L_H + P_B + V_1^2/2g$$

This is the sum of potential, pressure and kinetic energies at the liquid surface. Since the surface of the liquid supply is large compared to the area of the suction pipe, the velocity head is negligible and kinetic energy or velocity head is zero. Total energy at station 1 is:

$$L_H + P_B$$

P_B represents the pressure energy at station 1, or atmospheric pressure. To ensure the liquid does not vaporize in the suction line, subtract the vapor pressure (V_P) of the liquid from the pressure energy at station 1.

$$L_H + P_B - V_P$$

The pressure terms are expressed in psi absolute and converted to feet of head, the unit commonly used to express available NPSH:

$$L_H + (P_B - V_P) \times 2.31$$

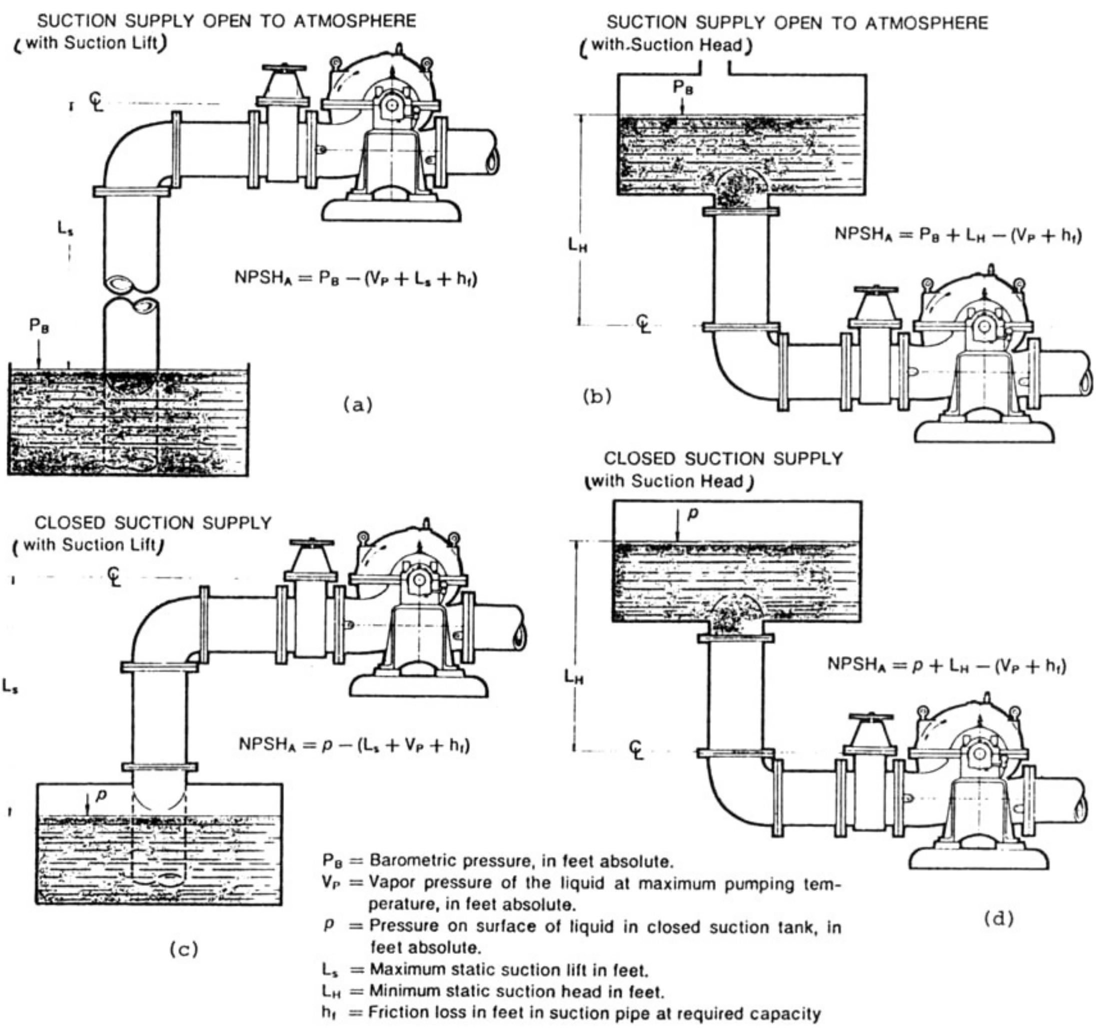


Figure 2-2. Calculation of System Net Positive Suction Head Available For Typical Suction Conditions.

It is equally important to correct for the specific gravity of the liquid if it is not water. The above equation can be rewritten:

$$L_H + \frac{(P_B - V_P)}{sp.gr.} \times 2.31$$

This is because the energy at station 2, (the point at which available NPSH is required), is equal to the energy at station 1 with the exception of losses due to friction. Available NPSH after subtracting these losses (h_f) at station 2 becomes:

$$LH + \frac{(P_B - V_P)}{sp.gr.} \times 2.31 - h_f$$

Example 2-1: A cold water system at 60°F has vapor pressure 0.256 psia and specific gravity 1.0, respectively. Determine the available NPSH for this system in Figure 2-3.

$$\text{Solution: The available NPSH} = L_H + \frac{(P_B - V_P)}{sp.gr.} \times 2.31 - h_f$$

$$= 10 + \frac{(14.7 - 0.256)}{1.0} \times 2.31 - 8$$

$$= 35.4 \text{ ft.}$$

Example 2-2: A hot water system at 200°F has vapor pressure 11.53 psia and specific gravity 0.965, respectively. Determine the available NPSH for this system in Figure 2-3.

$$\text{Solution: The available NPSH} = L_H + \frac{(P_B - V_P)}{sp.gr.} \times 2.31 - h_f$$

$$= 10 + \frac{(14.7-11.53)}{0.965} \times 2.31 - 8$$

$$= 9.85 \text{ ft.}$$

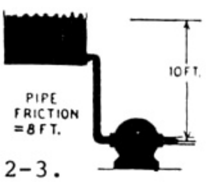


Fig. 2-3.

It is sometimes possible to determine available NPSH if test readings are available, see Figure 2-4.

Since station 2 is at the datum, the liquid has no potential energy and L_H equals zero. P_g equals gage reading. By adding atmospheric pressure to the gage reading to obtain absolute pressure head, subtracting vapor pressure, and correcting for the elevation of the suction gage (Y), available NPSH can be obtained:

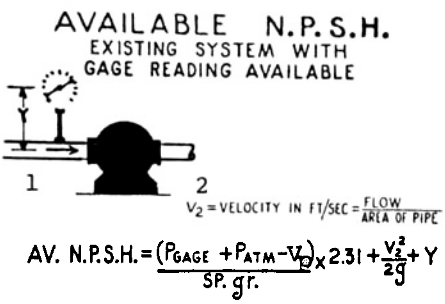


Figure 2-4.

$$\frac{(P_g + P_a - V_p) \times 2.31 + Y + V_2^2/2g}{\text{sp. gr.}}$$

V_2 is the velocity of the fluid in ft/sec.

All of the prior information is applicable when a single pump is used in a system. However, it is generally desirable to use two or more pumps in parallel, rather than a single larger pump. This is particularly advantageous when the system demand varies greatly and repairs or maintenance can be performed easily on one unit without completely shutting down the entire system.

Figure 2-5 shows the performance curves for system head and $NPSH_A$ for a system requirement of 16,000 gpm at a head of 140 ft. It should be noted that since friction losses increase with increased flow, $NPSH_A$ decreases with increased flow. Two pumps in parallel with each pump capable of single operation are desirable and economical.

Each pump must be sized for 8,000 gpm at 140 ft. total system head. Required NPSH for each pump must be less than 29 ft. for parallel operation (see NPSH curve in Figure 2-6). Consider the application of two pumps, each with similar characteristics as in Figure 2-5. To study both parallel and single operation, the head/capacity curve for both single and parallel operation must be plotted with the system head curve.

In Figure 2-6 the head-capacity curve for the two pumps in parallel can be plotted directly by adding the capacities ($Q_1 + Q_2$) of the individual pumps for various total heads selected at random. The required NPSH curve is plotted in the same manner; i.e., the required NPSH for one pump at 8000 gpm is 14.5 ft. and, therefore 16,000 gpm can be pumped in parallel operation with 14 ft. required NPSH by each pump. Figure 2-6 indicates that each pump will deliver 8,000 gpm at 140 ft. total system head when operating in parallel. Brake horsepower (BHP) for each unit will be 340 hp. The required NPSH is 14.5 ft. and the available NPSH is 29 ft.

With only one pump operating, the flow will be 11,000 gpm at 108 total system head (the point at which the head/capacity curve intersects the system head curve). BHP will be 355 hp, $NPSH_R$ is 26 ft. and $NPSH_A$ is 30 ft. This indicates that a 400- hp motor is required. If a 350-hp motor had been selected based upon parallel operation only, the motor would have been undersized for single pump operation. Single pump operation is also critical for the NPSH. If the system $NPSH_A$ had been about 20 ft., parallel pump operation would have been satisfactory, but single pump operation would result in cavitation.

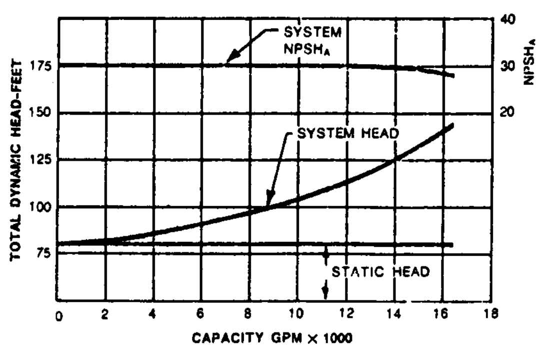


Figure 2-5. System Curves.

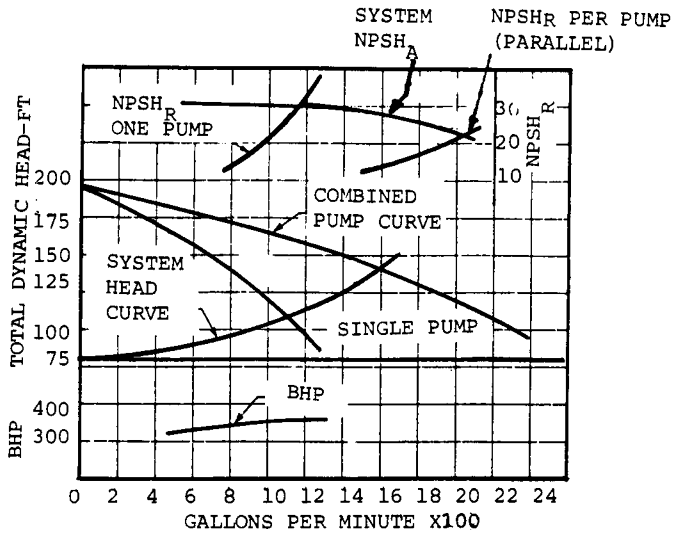


Figure 2-6. Parallel Operation

2.7 Pump Cavitation

When pressure in the suction line falls below vapor pressure, vapor forms and moves along with the stream. These vapor bubbles, or "cavities," collapse when they reach regions of higher pressure on their way through the pump.

The most obvious effects of cavitation are noise and vibration. These are caused by the collapse of the vapor bubbles as they reach the high pressure side of the pump. The bigger the pump, the greater the noise and vibration. If operated under cavitating conditions for a sufficient length of time, especially on water service, impeller vane pitting will occur. The violent collapse of vapor bubbles forces liquid at high velocity into vapor filled pores of the metal, producing surge pressures of high intensity on small areas. These pressures can exceed the tensile strength of the metal, and actually blast out particles, giving the metal a spongy appearance. This noise and vibration also can cause bearing failure, shaft breakage and other fatigue failures in the pump.

The other major effect of cavitation is a drop in pump efficiency, apparent as a drop in capacity. (See Figure 2-7.)

The drop in the efficiency and head capacity curve may occur before the vapor pressure is reached, particularly in petroleum oils, because of the liberation of light fractions, and dissolved and entrained air.

Pitting is not as serious when the pump is handling oils, due to the cushioning effect of the more viscous liquid.

In general, cavitation indicates insufficient available NPSH. Excessive suction pipe friction, combined with low static suction head and high temperatures contribute to this condition. If the system cannot be changed, it may be necessary to change conditions so that a different pump with lower NPSH requirements can be used. Larger pumps might require the use of a booster pump to add pressure head to the available NPSH.

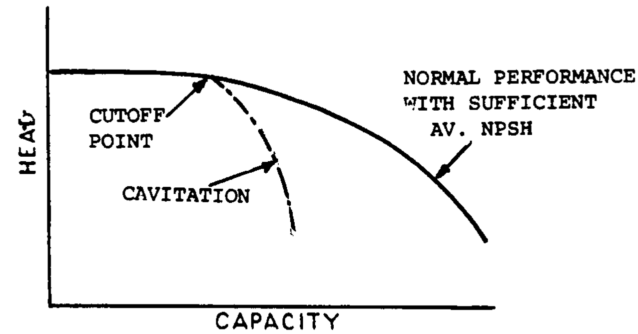


Figure 2-7. Cavitation - Effect On Pump Capacity.

2.8 Head

Head is commonly used to represent the vertical height of a static column of liquid corresponding to the pressure of a fluid at the point in question. Head can also be considered as the amount of work necessary to move a liquid from its origin to the required delivery position. This includes the extra work necessary to overcome the resistance to flow in the line.

In general, a liquid may have three kinds of energy, or the capacity to do work may be due to three factors:

1. Potential Head (energy of position) - measured by work possible in dropping vertical distance.
2. Static Pressure Head (energy per pound due to pressure). The height liquid can be raised by a given pressure.
3. Velocity Head (kinetic energy per pound). The vertical distance a liquid would have to fall to acquire the velocity "V".

Static Suction Lift. When the supply source is below the pump (see Figure 2-8). The vertical distance from the free surface of liquid to pump datum is called static suction lift. The net suction head, in this case, is the sum of static suction lift plus friction losses (negative net suction lift).

Static Suction Head. When supply is above the pump. Vertical distance from the free surface of liquid to pump datum is called static suction head. The net suction head is the sum of the static suction head minus friction losses (either positive or negative).

Total Head. The total head developed by the pump can be expressed by one of the following equations (see Figure 2-8):

PUMP WITH SUCTION LIFT

$$H = h_d + h_s + f_d + f_s + (V^2/2g)$$

PUMP WITH SUCTION HEAD

$$H = h_d - h_s + f_d + f_s + (V^2/2g)$$

where: H: total head (in feet) of liquid pumped when operating at the desired capacity.

h_d: static discharge head (in feet) equal to the vertical distance between the pump datum and the surface of liquid in the discharge reservoir.

h_s: static suction head or lift (in feet) equal to the vertical distance from the water surface to the pump datum. (This value is positive when operating with a suction lift and negative when operating with a suction head.)

f_d: friction head loss in the discharge piping (in feet).

f_s: friction head loss in the suction piping (in feet).

$V^2/2g$: $V^2/2g$ is velocity head (in feet). For vertical turbine and submersible pumps, the velocity head is measured at the discharge flange. However, for booster pumps and centrifugal pumps the velocity head developed by the pump is the difference between the $V^2/2g$ at the discharge flange and the $V^2/2g$ at the suction flange. That is,

$$V^2/2g = (V_d^2/2g) - (V_s^2/2g)$$

where $V_d^2/2g$ is velocity head at the discharge flange

$V_s^2/2g$ is velocity head at the suction flange.

Since the discharge flange is usually a size smaller than the suction flange, the difference in the velocity head is always positive. Usually, it is a small percentage of the total head, and frequently is erroneously neglected.

2.9 Bernoulli Theorem

The energy equation results from application of the principle of energy conservation to fluid flow. The energy possessed by a flowing fluid consists of internal energy and energies due to pressure, velocity and position. In the direction of flow, the energy principle is summarized by a general equation as follows:

$$\text{Energy at Section 1} + \text{Energy Added} - \text{Energy Lost} - \text{Energy Extracted} = \text{Energy at Section 2}$$

This equation, for steady flow of incompressible fluids where the change in internal energy is negligible, simplifies to

$$(p_1/w + V_1^2/2g + z_1) + H_A - H_L - H_E = (P_2/w + V_2^2/2g + z_2)$$

Energy cannot be created or destroyed. The sum of three types of energy (heads) at any point in a system is the same in any other point in the system, assuming there are no friction losses or performance of extra work. The above equation, in this case, can be further simplified to

$$(p_1/w + V_1^2/2g + z_1) = p_2/w + V_2^2/2g + z_2)$$

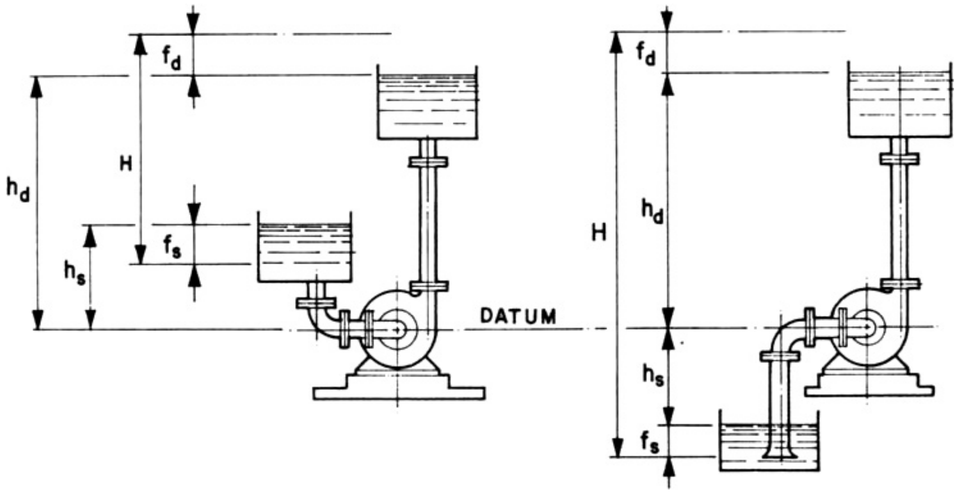
This equation is known as the Bernoulli Theorem. The units used are ft-lb/lb of fluid or feet of the fluid. Almost all problems dealing with liquid flow utilizes this equation as the solution basis.

Application of the Bernoulli Theorem should be rational and systematic. Suggested procedures follow.

(1) Draw a sketch of the system, choosing and labeling all stream cross sections under consideration.

(2) Apply the Bernoulli equation in the direction of flow. Select a datum plane for each equation. The low point is logical, as minus signs are avoided and mistakes reduced.

(3) Evaluate the energy upstream at Section 1. The energy is in ft-lb/lb units, which reduce to feet of fluid units. For liquids, the pressure head may be expressed in gage or absolute units, but the same basis must be used for the pressure head at Section 2. Gage units are simpler for liquids. Absolute pressure head units must be used where specific weight w is not constant. As in the equation of continuity, V_1 is taken as the average velocity at the



PUMP WITH SUCTION HEAD. PUMP WITH SUCTION LIFT.

Figure 2-8. Pump heads.

section, without loss of acceptable accuracy.

(4) Add, in feet of the fluid, any energy contributed by mechanical devices, such as pumps.

(5) Subtract, in feet of the fluid, any energy lost during flow.

(6) Subtract, in feet of the fluid, any energy extracted by mechanical devices, such as turbines.

(7) Equate this energy summation to the sum of pressure head, velocity head and elevation head at Section 2.

(8) If the two velocity heads are unknown, relate them to each other by means of the equation of continuity.

Example 2-3: Water flows through the turbine in Figure 2-9 at the rate of 7.55 cfs. The pressure at A and B, respectively, are 21.4 psi and -5.00 psi. Determine the horsepower delivered to the turbine by the water.

Step 1. Find the velocities at A and B respectively.

$$V_{12} = 7.55/A_{12} = 9.60 \text{ and } V_{24} = 9.60/4 = 2.40 \text{ ft/sec,}$$

Step 2. Set up the energy equation.

$$(P_A/w + V_{12}^2/2g + z_A) + 0 - H_{turbine} = (P_B/w + V_{24}^2/2g + z_B)$$

$$\frac{21.4(144)}{62.4} + \frac{(9.60)^2}{2g} + 3.00 - H_t = \frac{-5.00(144)}{62.4} + \frac{2.40^2}{2g} + 0$$

and $H_t = 65.4$ ft.

Step 3. Determined the horsepower deliverd to the turbine by the water.

$$\text{Horsepower} = wQH_t/550 = 62.4(7.55)(65.4)/550 = 56.0$$

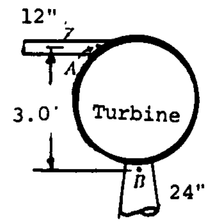


Fig. 2-9.

2.10 Friction Head

Friction head is the head lost in overcoming pipe friction. It depends on pipe size, smoothness of the inside surface, the number and type of fittings, orifice plates and control valves, flow velocity, and liquid viscosity and density. The following is a list of principal, semitheoretical flow equations for estimating frictional head losses.

Darcy-Weisbach Equation

The Darcy-Weisbach equation can be used to estimate friction head losses:

$$H_L = f\frac{L}{2gd}\frac{V^2}{} = f\frac{L\ V^2}{8gR} = \frac{\Delta P}{g\rho}$$

where: H_L = friction head loss, ft

f = friction factor

L = length of pipe

d = inside diameter, ft

R = hydraulic radius, ft

V = fluid velocity, fps

g = acceleration due to gravity (32.2 ft/s^2)

ρ = fluid density

Hazen-Williams Equation

For fluid velocity, the Hazen-Williams equation is

$$V = 1.318\ C\ R^{0.63}\ S^{0.54}$$

where: C = coefficient of roughness

S = slope of the energy line = H_L/L

V = Fluid velocity (fps)

A more useful form of the equation is

$$H_L = 0.2083\ (100/C)^{1.85}\ \frac{q^{1.85}}{d^{4.8655}}$$

where: H_L = friction head loss, ft per 100 ft of pipe

d = inside diameter of pipe, inches

q = quantity of flow, gpm

C = a dimensionless constant which reflects the roughness of pipe

Manning Equation

The Manning equation is an empirical expression originally developed for open-channel flows. However, it can be applied to full-flow pipe:

$$V = \frac{1.486}{\beta}\ R^{2/3}\ S^{1/2}$$

or

$$H_L = \frac{4.6666\beta^2 L}{D^{5.333}}\ Q^2$$

Friction Factor

Friction factor (f) can be derived mathematically for laminar flow, but there is no simple mathematical relation for the variation of f with Reynolds Number available for turbulent flow. Furthermore, Nikuradse and others found that the relative roughness of the pipe (the ratio of the size of surface imperfections to the pipe inside diameter affects the value of f also.)

(a) For laminar flow, the following equation can be used:

$$\text{Lost head} = 64\ \frac{L\ V^2}{V\ d\ d\ 2\ g} = \frac{64}{R_E}\frac{L}{d}\frac{V^2}{2g}$$

Thus, for laminar flow in all pipes for all fluids, the value of f is

$$f = 64/R_E \quad (R_E\text{: Reynolds Number})$$

R_E has a practical maximum value of 2000 for laminar flow.

(b) For turbulent flow, many hydraulicans have endeavored to evaluate f from the results of their own, and others, experiments.

(1) For turbulent flow in smooth and rough pipes, universal resistance laws can be derived from:

$$f = 8\tau_o/(\rho V^2) = 8V_*^2/V^2 \quad (V_*\text{: Shear Velocity})$$

(2) For smooth pipes Blasius suggests, for Reynolds Numbers between 3000 and 100,000

$$f = 0.316/R_E^{0.25}$$

For values of R_E up to about 3,000,000, Von Karman's equation modified by Prandtl is:

$$1/\sqrt{f} = 2 \log (R_E\sqrt{f}) - 0.8$$

(3) For rough pipes;

$$1/\sqrt{f} = 2 \log (r_o/\varepsilon) + 1.74 \quad (\varepsilon\text{: Roughness})$$

(4) For all pipes, the most up-to-date correlation of these factors is expressed in the Colebrook equation:

$$1/\sqrt{f} = -2 \log (k/3.7D + 2.51/R_E\sqrt{f}) \quad (k\text{=Relative Roughness})$$

This equation is cumbersome and contains too many variables to be of practical use in the above format. However, the Institution of Water Engineers (the Manual of British Water Engineering Practice) has published a nomograph entitled "Universal Pipe Friction Diagram" based on the work of Prandtl, Von Karman, Nikuradse and Colebrook. This nomograph is sufficiently accurate for most practical purposes, and is superior to the Hazen-Williams equation. Greater accuracy can be achieved by using this nomograph if velocity (V) is precalculated and plugged into the nomograph together with the internal pipe diameter (D), instead of using (Q) and the pipe diameter (D), which has too short a "length of sight" for accurate alignment.

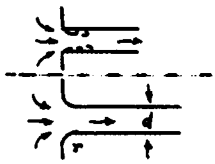
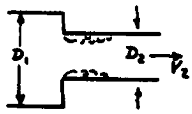
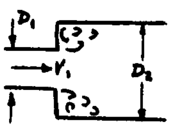
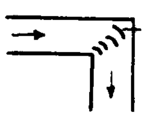
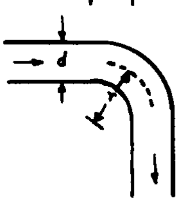
Loss Coefficients for Various Transitions and Fittings

The loss coefficients for various types of elbows along with a number of other fittings and flow transitions, are given in Table 2-1. The primary effect of head loss due to entrances, bends and other flow transitions is to cause the energy grade line to drop an amount equal to the head loss produced by that transition. Generally, this drop will occur over a distance of several diameters downstream of the transition.

2.11 System Head Curves

The system head curve is represented by a plot of total head versus system discharge. Such plots are very useful in selecting pump units. It should be clear that the system head curve will vary with flow. In addition, the static head may vary as a result of fluctuating water levels and similar factors. It is often necessary to plot system-head curves covering the range of variations in static head.

Table 2-1. Loss Coefficients For Various Transitions and Fittings

Description	Sketch	Additional Data	K_L
Pipe entrances		Square-edged	$K_e = 0.50$
		Rounded: $r/d > 0.12$	$K_e = 0.10$
Contractions and expansions	 $h_L = K_e V_2^2/2g$  $h_L = K_E V_1^2/2g$	D_2/D_1 for K_e or D_1/D_2 for K_E:	K_e K_E
		0.0	0.50 1.00
		0.1	0.49 0.98
		0.2	0.48 0.94
		0.4	0.44 0.71
		0.6	0.32 0.41
		0.7	0.23 0.22
		0.8	0.15 0.13
		0.9	0.06 0.04
90° miter bend		Without vanes	$K_b = 1.1$
		With vanes	$K_b = 0.2$
90° smooth bend		r/d	
		1	$K_b = 0.35$
		2	0.19
		4	0.16
		6	0.21
		8	0.28
		10	0.32
Threaded pipe fittings	Globe valve – wide open		$K_v = 10.0$
	Angle valve – wide open		$K_v = 5.0$
	Gate valve – wide open		$K_v = 0.2$
	Gate valve – half open		$K_v = 5.6$
	Return bend		$K_b = 2.2$
	Tee		$K_t = 1.8$
	90° elbow		$K_b = 0.9$
	45° elbow		$K_b = 0.4$

There are two general types of pumping systems - closed systems and open systems. For a hydraulic closed system, the system pressure can be changed arbitrarily. These systems usually have a pressure regulator or a compression tank to control the required system pressure. An open system has a breaking point in the hydraulic line, and requires information on the static pressure difference between the suction and discharge ends.

In both types of pumping systems, the resistance to fluid flow because of pipe friction is always there. This resistance is a function of the square of the flow rate. This relation can be shown graphically and is normally called a system head curve. (See Figure 2-10).

$$h_1/h_2 = (Q_1/Q_2)^{1.85}$$

where:

h_1 = friction head at design flow Q_1
h_2 = friction head at assumed flow Q_2

To evaluate system operation, the capacity head curve and the system head curve are plotted on a common graph. The two curves will intersect at maximum flow for the sytem. As in Figure 2-10, a system flow greater than the system operating point cannot occur since the required system head exceeds the total capacity head.

An open system with more static pressure head on the suction end of the pump than on the discharge end has static pressure available to increase system flow. This extra static pressure is illustrated by shifting the system head curve down by an amount equal to the static pressure Z in Figure 2-11. If a system has less static pressure on the suction end than on the discharge end, the amount of static pressure decreases the system flow. This is graphically illustrated by shifting the system head curve up by an amount equal to the static pressure difference between suction and discharge ends. (See Figure 2-12).

Sometimes estimated friction losses in hydraulic circuits are adjusted by a control device, causing prediction of pump operation at a relatively high head. When actual system head losses are less than the predicted, the actual system head curve is lower than the predicted curve. Consequently, the actual pump operating point will shift as shown in Figure 2-13, and power consumption is increased. This shift in the pump operating point will also increase NPSH requirements and can create noise and mechanical damage to the pumping system.

Example 2-4: What will be the discharge in this water system if the pump has the characteristics shown in the following graph? Assume f = 0.015.

Step 1. First write the energy equation from water surface to water surface:

$$p_1/w + V_1^2/2g + Z_1 + h_p = P_2/w + V_2^2/2g + Z_2 + h_L$$

$$0 + 0 + 600 + h_p = 0 + 0 + 700 + (fL/D + K_e + K_b + K_E)\ V^2/2g$$

Here $K_e = 0.5$, $K_b = 0.35$, and $K_E = 1.0$. Hence

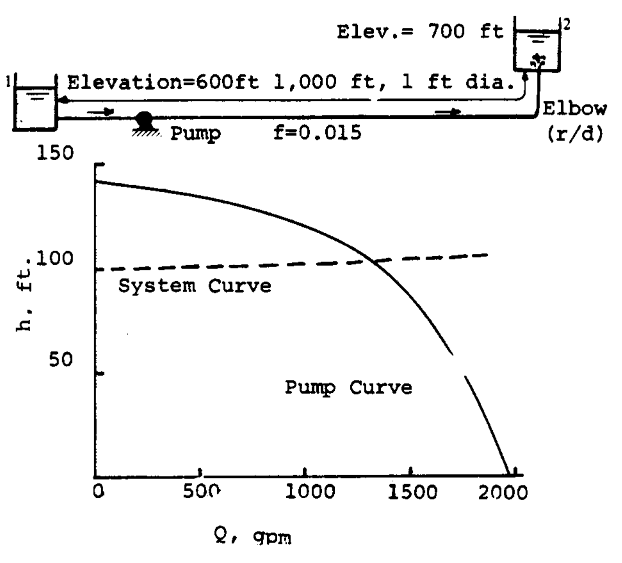
$$h_p = 100 + 1/2g(Q/A)^2 \left\{ \frac{0.015(1000)}{1} + 0.5 + 0.35 + 1 \right\}$$

$$= 100 + Q^2/39.6\ (16.85) = 100\text{ ft} + 0.43\ Q^2\text{ ft}$$

Step 2. Produce a system head curve:

Now let us make a table of Q vs. h_p to give values to produce a system curve which will be plotted with the pump curve. When the system curve is plotted on the same graph as the pump curve (dashed line), it is seen that the operating condition occurs at Q = 1,320 gpm

Q, cfs	Q, gpm	Q^2	ft^6/sec^2	$0.43\ Q^2$	$h_p = 100 + 0.43\ Q^2$
0	0	0		0	100
1	449	1		0.4	100
2	898	4		1.7	102
3	1344	9		3.9	104
4	1796	16		6.9	107



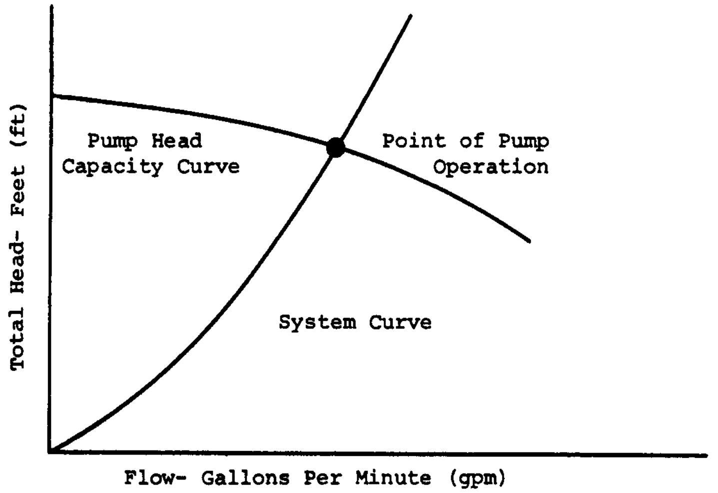


Figure 2-10. Relation Between Pump Performance Curve and System Characteristic Curve.

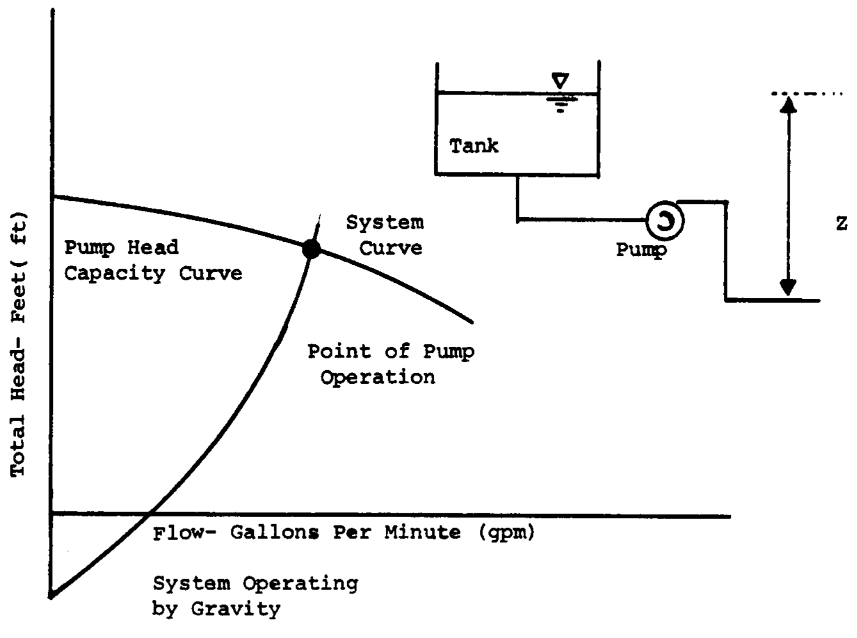


Figure 2-11. Shift in System Curve Caused by Higher Static Head on Suction Side of Pump.

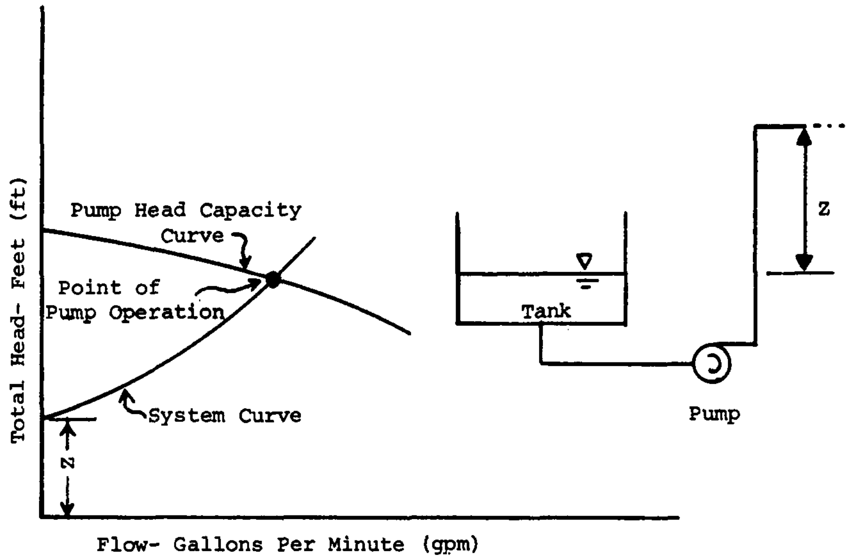


Figure 2-12. Shift in System Curve Caused by Lower Static Head on Suction Side of Pump.

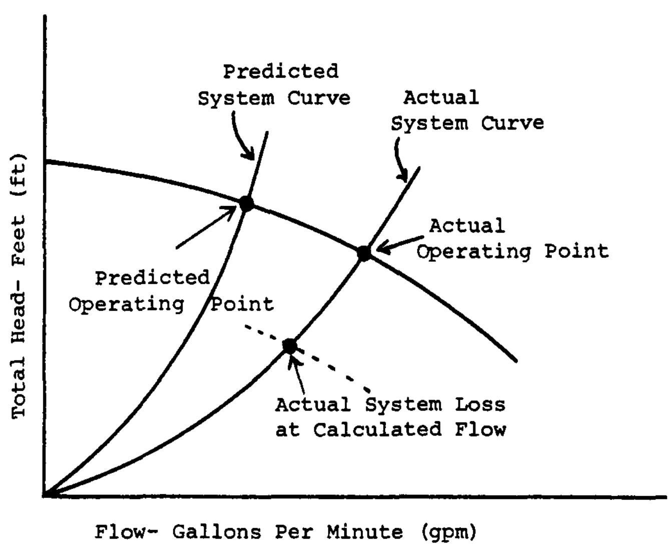


Figure 2-13. The Difference Between Predicted and Actual System Curve.

2.12 Optimum Pipe Diameter

Calculation of the most economical, or optimum, pipe diameter for a given system application is a very complex process. The two most important items are piping and energy costs. These, in turn, are dependent upon numerous factors such as pipe size, piping material costs, fuel or power costs, equipment costs, maintenance costs and discount rate on capital for energy costs. Figure 2-14 shows in a simplified manner the relationship of pipe size and pumping cost, and how they combine for total cost.

As pipeline costs increase, the pumping costs decrease. Total cost is the sum of pipeline and energy costs.

It can be seen that for small pipe diameters, pumping and total costs decrease rapidly as pipe diameter increases. But as the pipe size increases, the cost to install each additional unit of pipe becomes increasingly larger. Also, the cost of pumping begins to level off as the energy savings for each additional unit of pipe size decrease. This can be easily understood by superimposing pumping and pipeline costs on the same graph.

The economical pipe diameter is selected on the basis of friction losses only, and is not influenced by static head.

Having selected the optimum pipe diameter, the system head curve should be plotted (head against flow). The static head is the system head curve datum at zero flow. If the suction well level is subjected to seasonal variations which will affect the system head curve, a parallel curve should be shown for the system head curve under each set of conditions. Likewise, if the pumps discharge into an elevated tank, with upper and lower level limitations, this too should be indicated on the system head curve to ensure that all conditions of pump services are presented.

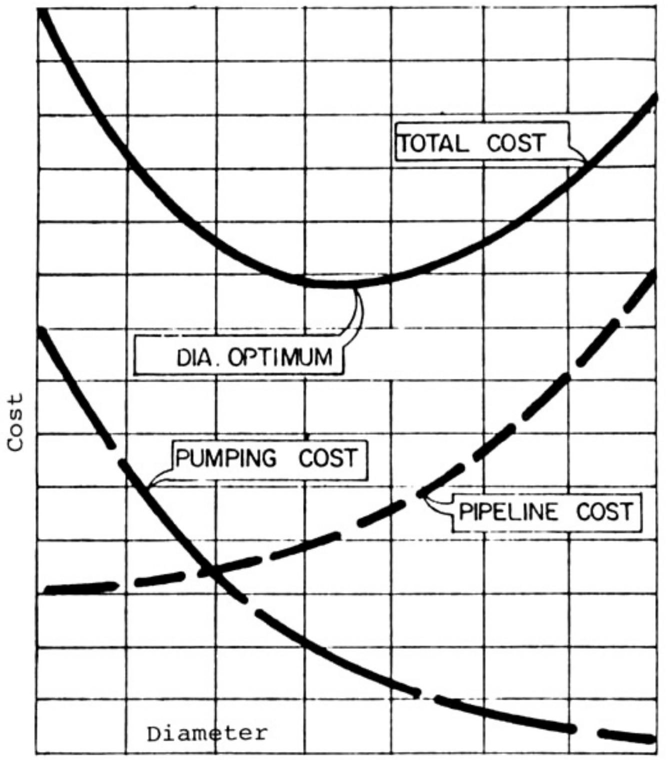


Figure 2-14. The Optimum Pipe Diameter.

2.13 References

1. "Centrifugal Pumps: Part 1," by Wen-Yung W. Chan, Plumbing Engineer, pages 22-23, including Figs. 1-6, Nov./Dec., 1982. Edited by permission.

2. "Pump Cavitation and NPSH," by Dr. Alfred Steele, Plumbing Engineer, pp. 19-21, including Figs. 1-4, Jan/Feb 1981. Edited by permission.

3. "Pump Selection," by Rodger Walker, Ann Arbor Science, pp. 15-16, including Fig. 4, 1982. Edited by permission.

CHAPTER 3

PUMP SPECIFICATIONS

3.1 Specific Speed

Specific speed correlates pump flow, head and speed at optimum efficiency. It shows the relation of pump impellers to their geometric similarity. Specific speed is expressed as:

$$N_s = \frac{N\sqrt{Q}}{H^{3/4}}$$

where: N_s = pump specific speed
N = shaft speed in revolutions per minute, rpm
Q = flow at optimum efficiency, gpm

The specific speed of a given impeller is defined as the revolutions per minute for a geometrically similar impeller would run if it were sized to discharge 1 gpm against a 1 ft. head. Specific speed is an index of the impeller's shape and characteristics.

Centrifugal pumps are classified into three categories:

- Radial flow
- Mixed flow
- Axial flow

The radial flow impeller develops head basically by the action of centrifugal force. The axial flow impeller develops most of its head by the propelling or lifting action of the liquid. There is a continuous change from the radial flow impeller to the axial flow impeller. More specifically, centrifugal pumps can also be classified by physical characteristics relating to the specific speed range of the design, Figures 3-1 through 3-5. Once the values for head and capacity are established for a specific application, the pump's specific speed range can be determined to ascertain the selection of a pump with optimal efficiency.

3.2 Suction Specific Speed

While specific speed (N_s) is an index number indicating pump type, the index known as suction specific speed (S) is essentially a number describing of the suction characteristics of a given impeller. It is defined as:

$$S = \frac{\text{rpm}\sqrt{Q}}{\text{NPSH}^{0.75}}$$

where: rpm = shaft speed in revolutions per minute
NPSH = the required NPSH for satisfactory operation in feet
Q = the flow in gallons per minute

It should be noted that for double suction impellers, the flow gpm should be taken as one-half the total flow.

The upper limits of specific speed (N_s) and suction specific speed (S) are given in Figures 54-57 of the Hydraulic Institute Standards.

3.3 Variable Speeds

Variable speed drives are becoming increasingly popular. Pump characteristics influenced by pump speed are flow (gpm)

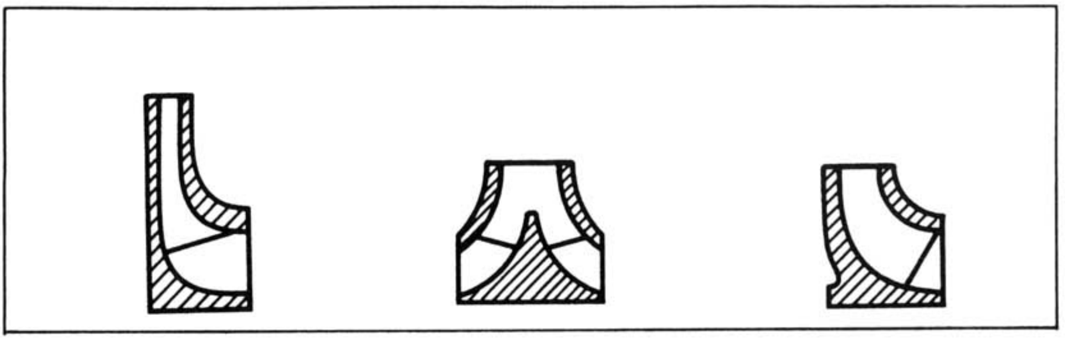


Figure 3-1. Single suction, horizontal or vertical centrifugal pumps with narrow port impellers have low capacities and deliver high heads. Specific speed range is 500 to 1000.

Figure 3-2. Single or multistage, double (illustrated) and single suction, volute and diffuser design centrifugal pumps deliver medium capacity and medium heads. Specific speed range is 1000 to 2000.

Figure 3-3. Single or multistage and single (illustrated) and double suction, Francis-type impellers, operating in volute or diffusor type casings produce medium to low speeds. Specific speed range is 2000 to 4000.

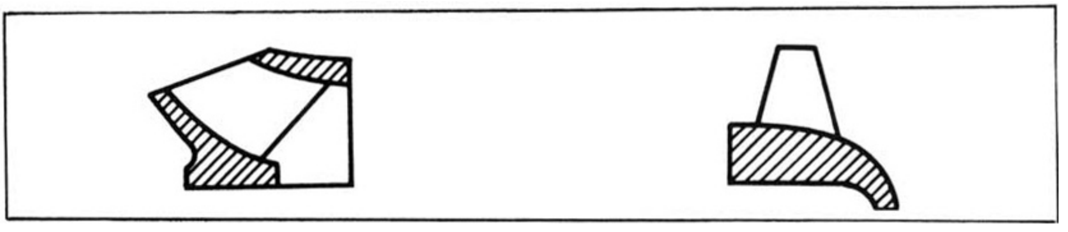


Figure 3-4. Single or multistage single suction centrifugal pumps, in volute or diffuser-type casings with mixed flow impellers, deliver high capacity at low head. Specific speed range is 4000 to 6000.

Figure 3-5. Single or multistage pumps with mixed flow and propeller-type impellers have very high capacities and deliver very low heads. Specific speed range is 6000 to 10,000.

varies directly with (rpm); head (ft) varies as the $(rpm)^2$; and horsepower (bhp) varies as the $rpm)^3$.

The ability to reduce the rpm is analogous to having an impeller of variable diameter.

Controls for variable-speed pumps can be used to vary pump discharge and maintain a predetermined level, pressure or flow. Flow control is used to meet a fluctuating demand by varying the speed of the pumping unit. Sometimes flow control is provided by throttling. However, it is usually more economical to vary the pump speed. Speed variation can be provided by magnetic couplings, hydraulic couplings, wound rotor motors and liquid rheostats, and variable frequency and voltage controller. Speed adjustments can be manual or automatic.

Location of Sensing Equipment

The sensing equipment must perform three functions: (1) monitor the system variable; (2) compare the value of the system variable to the required value; and (3) cause the variable-speed pump to restore the variable to the required value.

The total system operation for a variable speed pump must be evaluated at the location where the system variable is being monitored.

When equipment in the system causes a change in flow, the path of pump operating points will not follow the unaltered system head curve. An infinite number of new system head curves will occur when system components cause flow change. For example, a system requiring a fixed design working head, the path of pump and system operating points can be predicted once the location of the system monitoring is known. This path of operation will be controlled by the location of the monitor, and will affect the operating speeds and power consumption of the pump.

Figure 3-6 shows a simple system in which flow change is controlled by a valve at the far end of the system. With the valve in a fixed position, the system head curve for this example, as shown in Figure 3-7, includes static head (H_{s1}) and friction head H_f between A and C.

Pressure monitor at the point of entry:

A variable-speed pump with a pressure monitor at the point of entry to the system, A, will produce a constant total available head at A for any given system flow. The setting for the pressure monitor is determined from the following formula:

$$P_1 = P_2 + H_f + H_s$$

where:
- P_1 = total available head (in feet) to be maintained at the pressure monitor
- P_2 = design working head, (in feet)
- H_f = friction head (in feet) between monitor and the location in the system where P_2 is to be maintained
- H_s = static head (in feet) between the monitor and the location in the system where P_2 is to be maintained

In Figure 3-7, P_2 is zero and $P_{1A} = H_f + H_{s1}$ (the pressure monitor setting).

As the system flow is throttled, a series of new system head curves will be produced, as shown in Figure 3-8. System flow will occur at those points where the system head curves intersect the available head curve.

Pressure monitor at the far end:

Considering the same system with pressure monitor located at point B in Figure 3-6, the total system head required is still as shown in Figure 3-7. The pressure to be maintained at the monitor is determined from the same formula. Design working head in this example, P_2, is zero, and static head is equal to H_{S3}. The total head that must be available at point A to satisfy the monitor at point B will be equal to the pressure monitor setting, P_{1B}, plus the static head, H_{S2}, plus the friction head from A to B. This is shown in Figure 3-9.

As the system flow is throttled, a series of system head curves will be produced, as shown in Figure 3-10. System flow will occur at those points where the system head curves intersect the total available head curve. By comparing Figure 3-8 and 3-10, the throttling losses are less with the pressure monitor at B than with the pressure monitor at A for specific values of flow. This demonstrates that the pump speed should be controlled by the remote pressure sensing equipment.

System Characteristics of Variable-speed Pumps

There is no simple mathematical relationship that can be developed for computing the interacting characteristics of systems and variable-speed pumps. The affinity laws, $Q_2 = Q_1(H_2/H_1)^{1.85}$, are helpful in computing performance for individual variable-speed pumps.

Determining the precise $Q_2 - H_2$ point on the pump curve usually requires iterative procedure. For example, the point on the pump curve of the known speed that produces the required system values $Q_1 - H_1$ at reduced speed can be determined by calculating the initial $Q_2 - H_2$ condition as a reference point. This is done by entering known system conditions $Q_1 - H_1$ and an initial value of H_2 in the flow-head formula. The initial value of H_2 is taken from the known pump performance curve at system flow, Q_1. By solving the formula for the initial value of Q_2, a reference point for initial $Q_2 - H_2$ is established.

If the initial value of Q_2 is off the pump curve, a new value of $Q_2 - H_2$ should enter into the formula. This procedure will allow generation of a new set of $Q_2 - H_2$ which is close to the pump curve. A simple mathematical formula used to describe the iterative procedure follows:

$$Y_{i+1} = Y_i - \frac{f(Q_2,H_2)}{f'(Q_2,H_2)}$$

where: $f(Q_2,H_2)$ = a given pump curve

$f'(Q_2,H_2)$ = slope of a given pump curve

Y_{i+1} = a new set of $Q_2 - H_2$

Y_i = an old set of $Q_2 - H_2$

Convergence will be noticeably fast if the reference point for initial $Q_2 - H_2$ on the pump curve is close to the precise point.

A simple triangular process recommended by pump manufacturers can also be used to determine the precise $Q_2 - H_2$ point on the pump curve. This is simply done by connecting the initial values of $Q_2 - H_2$ with required system condition $Q_1 - H_1$, an intersection point with the known pump curve is obtained. This intersection point becomes the approximate $Q_2 - H_2$ value that will, at reduced operating speed, produce system condition $Q_1 - H_1$. The triangular procedure is illustrated in Figure 3-11.

$Q_2 - H_2$ point on equally sized variable-speed pumps operating in parallel:

In a parallel pumping system, where system flow is divided equally by all pumps, the procedure is similar to that for a single pump system, except that system flow (Q_1) must be divided by the number of pumps operation (n). Figure 3-12 shows the triangular process for a system containing two equally sized pumps operating in parallel.

The flow and head developed by unequally sized variable-speed pumps operating in parallel:

When pumps are selected to share unequal portions of the total system flow, the individual pump flow-head formula must be rearranged to $Q_1 = Q_2\ (H_1/H_2)^{1.85}$. This arrangement is important since the flow from each pump under a selected system flow (Q_1) is unknown. To determine the flow and head contributed by unequally sized variable-speed pumps, one must solve the basic triangular process for system flow (Q_1) on the combined pump characteristic curve (see Figure 3-13). The intersection point on the combined pump curve establishes the H_2 value for the individual pump. By using the formula $Q_1 = Q_2\ (H_1/H_2)^{1.85}$, individual pump flow for the system operation at Q_1 can then be determined.

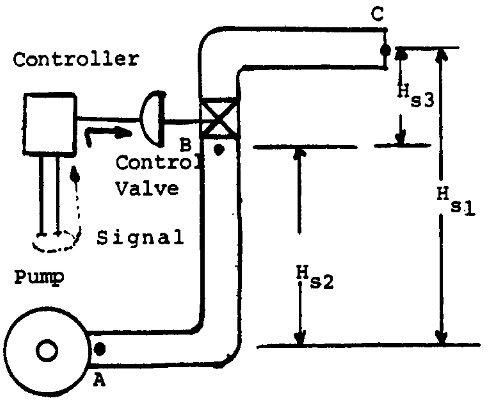


Fig. 3-6. A Simple System in Which Flow Change is Caused by a Valve Near the End of the System.

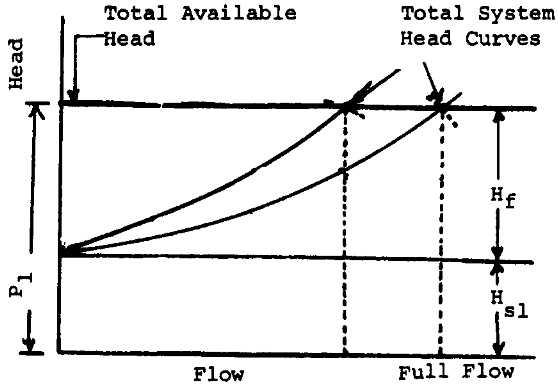


Fig. 3-8. Series of Head Curves Generated for Example Problem.

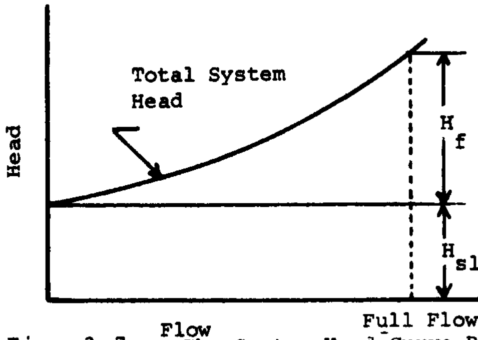


Fig. 3-7. The System Head Curve Between Points A and C with the Valve in a Fixed Position.

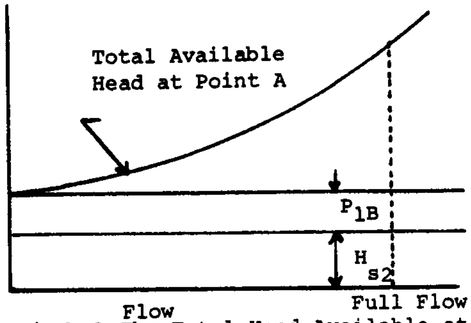


Fig 3-9. The Total Head Available at Point A to Satisfy Monitor at Point B.

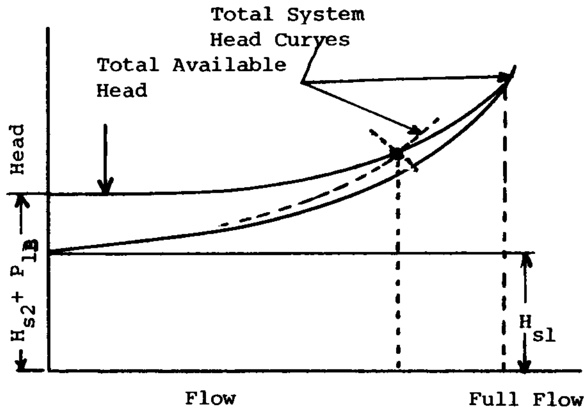


Figure 3-10. Series of Head Curves Generated for Example Problem.

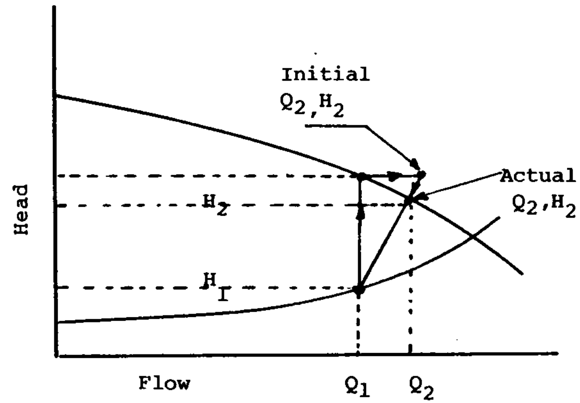


Figure 3-11. The Actual Q_2 on The Pump Curve.

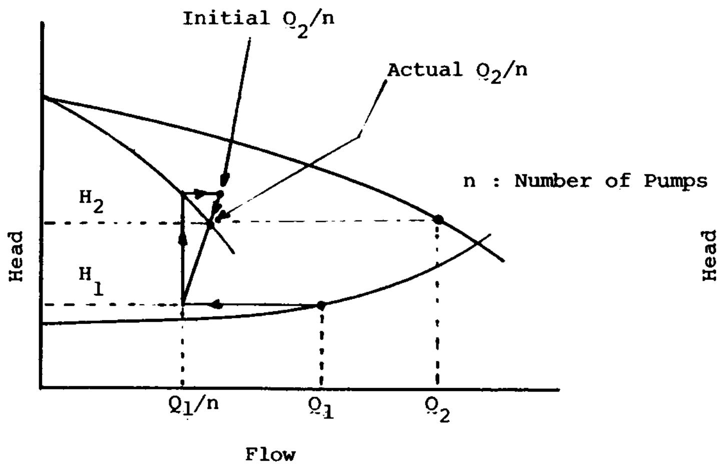


Figure 3-12. The Actual Q_2 Point on Two Equally Sized Variable Speed Pumps Operating in Parallel.

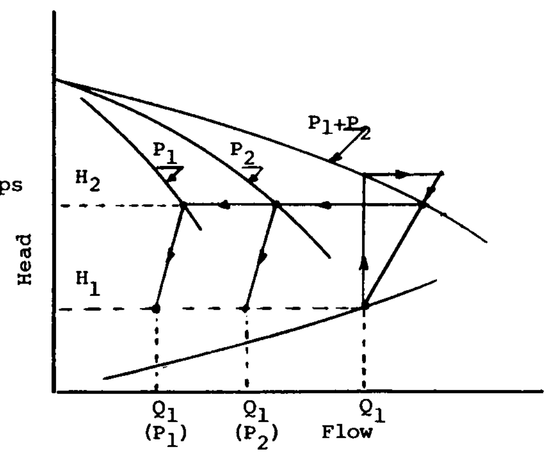


Figure 3-13. The Flow and Head Developed by Unequal Sized Variable-speed Pumps Operating in Parallel.

3.4 Affinity Laws

Affinity laws describe the relationships between head, capacity, brake horse power and impeller diameter of a given pump. The first law states the performance data of constant impeller diameter with change of speed. The second law assumes the performance data of constant speed with change in diameter of the impeller. These laws are written as follows:

Law 1.

$Q_1/Q_2 = N_1/N_2$ $\quad H_1/H_2 = N_1^2/N_2^2$ $\quad bhp_1/bhp_2 = N_1^3/N_2^3$

Law 2.

$Q_1/Q_2 = D_1/D_2$ $\quad H_1/H_2 = D_1^2/D_2^2$ $\quad bhp_1/bhp_2 = D_1^3/D_2^3$

The nomenclature for the equations is Q = capacity; H = head; N = speed; D = impeller diameter; and bhp = brake horsepower. Thus:

Q_1= gpm at N_1 or D_1 $\quad Q_2$= gpm at N_2 or D_2

H_1= head at N_1 or D_1 $\quad H_2$= head at N_2 or D_2

bhp_1= brake horsepower at N_1 or D_1 $\quad bhp_2$= brake horsepower at N_2 or D_2

Law 1 can be used for common types of pumps, including horizontal centrifugal-type fire pumps and vertical turbine fire pumps. Law 2 can be applied to centrifugal pumps with reasonably close agreement between calculated and tested performance data. For example, the data in Table 3-1 illustrate a projection of pump performance from 1760 rpm to 1450 rpm. The subscript 2 is designated for the calculated performance and subscript 1 for the values selected from the 1760 rpm pump curve.

Table 3-1
A Projection of Pump Performance Data

Tabulated Performance Data from 1760 rpm curve				Calculated Performance Data for 1450 rpm Operation			
Q_1	H_1	Eff_1	bhp_1	Q_2	H_2	Eff_2	bhp_2
1000	184	61	76.2	824	125	61	42.6
1500	175	76	87.2	1236	119	76	48.9
2000	166	84	99.8	1648	113	84	56.0
2500	151	86	110.8	2060	103	86	62.3
3000	128	82	118.3	2478	87	82	66.4
3250	110	73	123.7	2678	75	73	69.5

3.5 Performance Curves

A typical pump performance curve describes head, horsepower, efficiency, and the net positive suction head required for proper pump operation. Pump performance can be shown as a single line curve describing one impeller diameter (Figures 3-14 to 3-17), or as multiple curves for the performance of several impeller diameters in one casing (Figure 3-18).

Drooping Head/Continually Rising Head

Drooping head characteristics are those in which the head at zero flow is less than the head developed at some duty-point flow. Continually rising head characteristics are those in which the head rises continually as the flow is gradually reduced to zero. Discontinuous head characteristics are those in which a specific head is developed by the pump at more than one given flow rate. Many pumps in the high specific speed ranges offer these characteristics.

Steep Head/Flat Head

There is a common classification of centrifugal pumps into steep head flat head curved pumps, which refers to the shape of the head/capacity curve as shown in Figure 3-15. Flat head curved pumps are frequently used for closed circuit systems, because large changes in pump capacity result in a small change in head. Steep head curved pumps are often selected for open circuit systems, because a change in head has a minimum effect on capacity.

If the pump is designed for a low head duty, an axial-flow impeller can be used. The pump exhibits a steep-head curve and may operate safely over its entire head/capacity curve (Figure 3-16). When pumps are used in water boosting system a differently designed impeller, called radial blade, is used to develop the required head. The head/capacity curve is typically "flat" rather than "steep."

Such pumps have both a maximum and minimum safe-flow limit. If operated beyond its maximum safe-flow limit, the pump enters a high-flow low-head cavitation area. In this area, the pump generates disturbing noises that can be transmitted throughout the entire piping system. The destructive cavitation will also produce extra bearing load and sharply increased shaft deflection. On the other hand, if this pump is operated below its recommended low flow limit, it enters an area of low-flow hydraulic turbulance. This is also destructive and produces disturbing noises (Figure 3-17).

Continuous, Discontinuous and Peaking Horsepower Characteristics

Low specific speed pumps with continually increasing horsepower characteristics are those pumps that the horsepower increases at flows greater than the best efficiency point, and decreases at flows to the left of the best efficiency point. For those medium specific speed pumps with peaking horsepower characteristics, the maximum horsepower occurs in the best efficiency point region and decreases at all other values of flow. Discontinuous horsepower characteristics are those pumps with high specific speeds. The horsepower increases at flows less than BEP and decreases with flow to the right of BEP. The discontinuity usually occurs at the lower flow region.

3.6 Efficiency

It should be noted that one horse power is defined as being equivalent to 33,000 ft. lb./min. In Figure 3-19, the pump curve illustrates that gpm can be converted to a liquid flow rate expressed in lb./min. The pump increases the energy head of the liquid by the amount shown as ft. head. The consequent ft. lb./min relationship shown by the capacity curve is then:

$$\frac{\text{ft. lb}}{\text{min}} = \text{ft. head} \times \frac{\text{gal.}}{\text{min.}} \times \frac{8.33\ \text{lb}}{\text{gal.}}$$

Ft. lb/min. can be converted to HP since one HP equals to 33,000 ft. lb./min. The power into the water (water horsepower or WHP) can be calculated as below:

$$\text{WHP} = \text{ft. hd.} \times \frac{\text{gal.}}{\text{min.}} \times \frac{8.33\ \text{lb.}}{\text{gal.}} \times \frac{\text{HP}}{33{,}000\ (\text{ft. lb/min})}$$

$$= \frac{\text{ft. hd.} \times \text{gpm}}{3{,}960}$$

Figure 3-14. Pump Performance Curves.

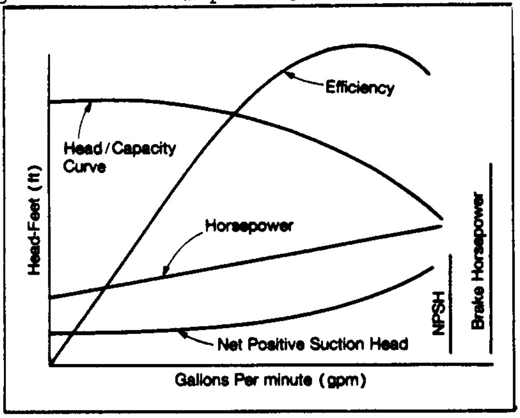


Figure 3-15. Performance Curves For Steep-Head and Flat-Head Pumps.

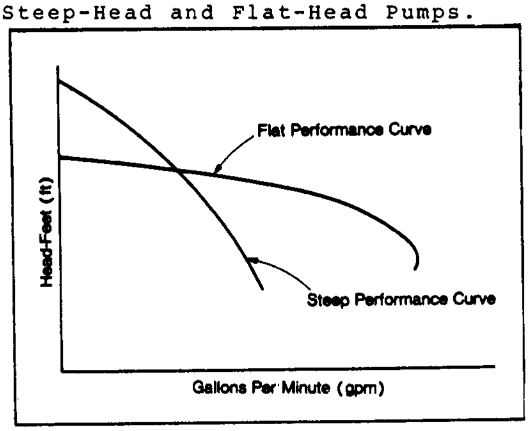


Figure 3-16. Head/Capacity Curve For Pump With Axial-Flow Impeller.

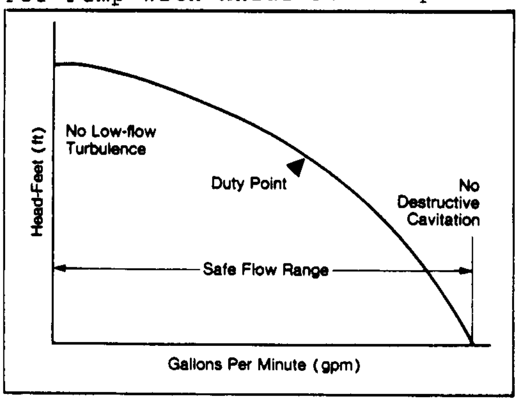


Figure 3-17. Maximum and Minimum Safe-Flow Limits

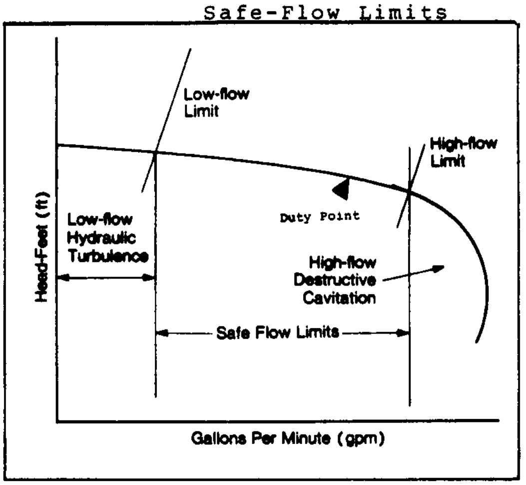
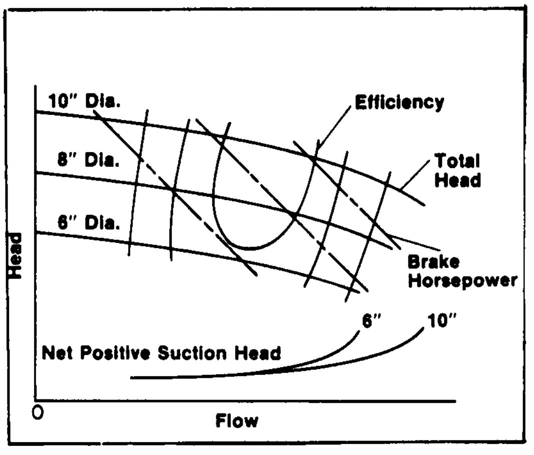


Figure 3-18. The Performance of Several Impeller Diameter in One Casing.



The pump has bearing friction and internal slip losses, meaning that more power must be put into the pump than is delivered to the water.

Pump efficiency (E_p) is WHP divided by BHP. This allow derivation of the final BHP relationships:

$$BHP = \frac{WHP}{E_p} = \frac{ft.\ hd.\ x\ gpm}{3,960\ x\ E_p}$$

Figure 3-19 also shows an important pump power basic; pump BHP will decrease at shut-off or low flow to about 50% of BHP draw at the pump best efficiency point (BEP).

Most pump casings allow usage of different pump impeller diameters. Each different impeller will provide a different pump capacity curve. The usual published pump curve is then an illustration of the various curves associated with specific impeller diameters. Efficiency and BHP curves are cross-plotted across the pump curves as shown in Figure 3-20.

Pump efficiency varies with pump size. Large pumps are more efficient than smaller pumps. Figure 3-21 shows "average" or generalized 3500 rpm pump efficiency at BEP, referenced to gpm and ft. hd. It should be noted that "design" pump efficiency increases with increased flow specification and with decreased head specification.

The horsepower at the output shaft of the prime mover for a variable-speed pump is obtained by dividing the pump brake horsepower by both the prime mover efficiency and the variable-speed drive efficiency.

When making variable speed pump brake horsepower calculations, it is necessary to know the equivalent point on the full speed characteristic curve which equates to the required system flow and head, since the efficiency at the equivalent point will reoccur at the reduced flow and head condition. The reduced pump operating speed necessary to satisfy the system flow and head can be projected from the affinity law derivation.

3.7 Packing

Advantages of using pump packing are:

- Inexpensive
- Easy installation
- Easy replacement without taking the pump apart

Disadvantages of using pump packing are:

- Requires some liquid leakage
- Causes shaft sleeve wear
- Requires periodic maintenance
- Incurs some horsepower loss

Packing is a compression type of seal. It is compressed by the packing grand which moves outward and inward to physically contact with the packing box and shaft sleeve (See Figure 3-22). As the shaft rotates, the packing cannot be compressed so tightly as to eliminate all leakage. In fact, small leakage is needed to insure proper packing lubrication to prevent burning out the packing and wearing the shaft sleeve. In spite of this lubrication, the packing will still wear and shrink, thereby requiring timely maintenance to tighten the packing gland and control the leakage rate.

For packing lubrication in a suction lift pump, it is necessary to facilitate a by-pass tubing between the pump casing and the packing box to insure a supply of clean liquid to the entire area between the packing and shaft sleeve. The tubing allows

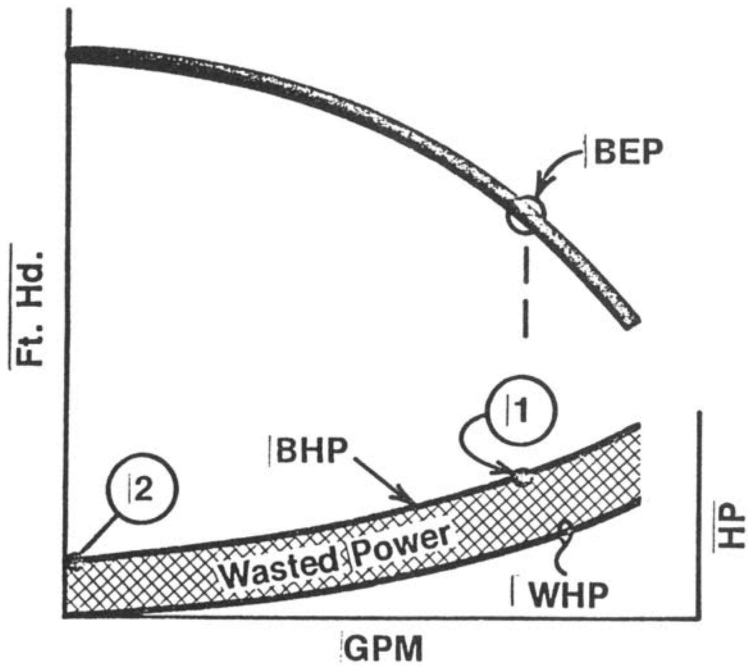


Figure 3-19. Pump Power Basic.

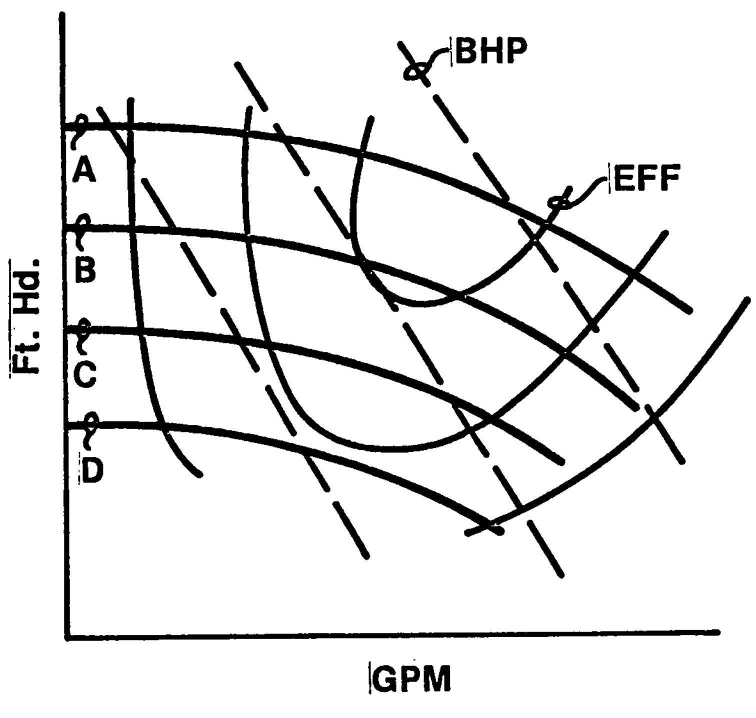


Figure 3-20. Efficiency and BHP Curves.

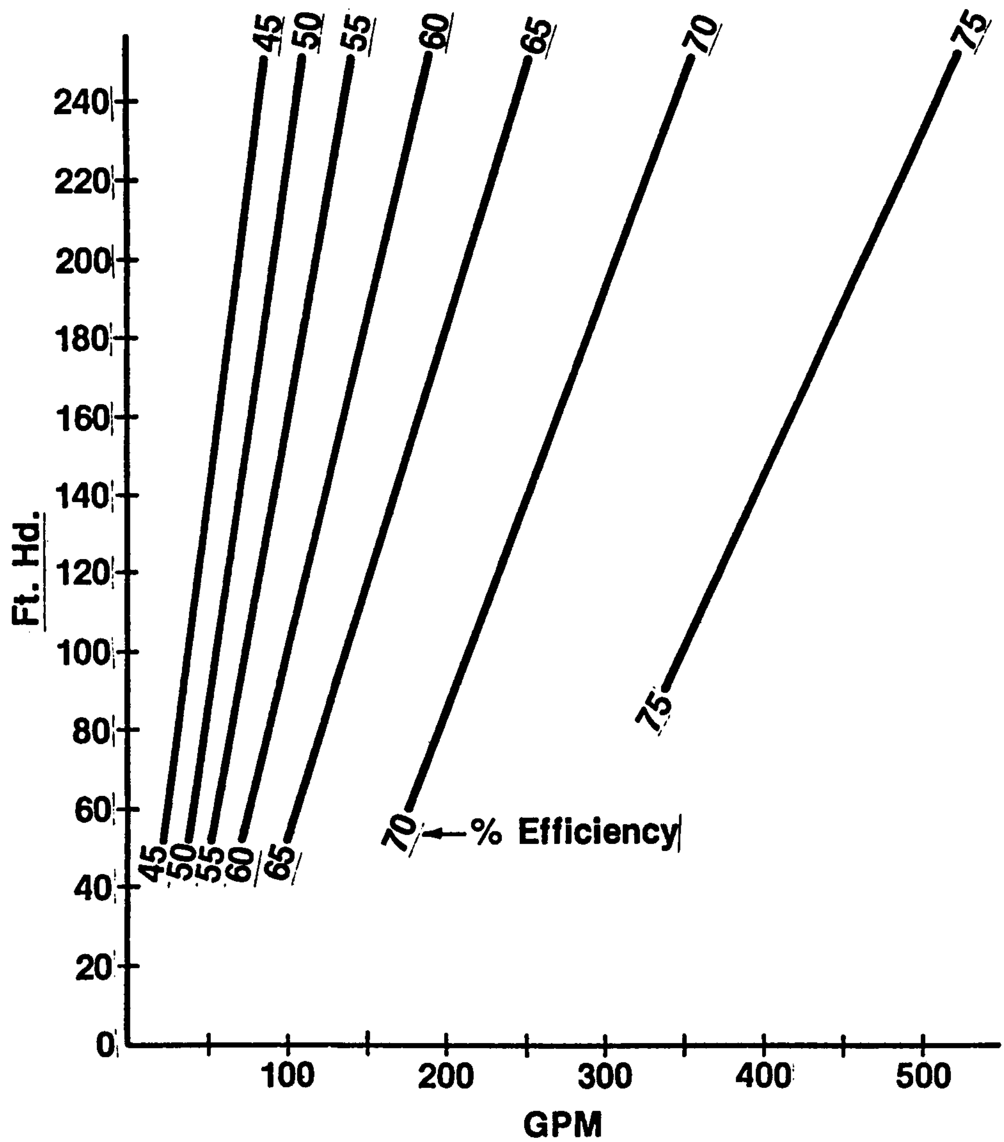


Figure 3-21. Average 3500 rpm Pump Efficiency at BEP.

clean liquid to enter the lantern ring, which distributes it uniformly around the shaft. This clean liquid is then forced through the gap between the packing and shaft sleeve by pump pressure. This by-pass system prevents air from being drawn into the pump through the packing gland.

When large amounts of abrasive materials are involved, pump by-pass tubing is replaced by an external source of clear, clean liquid. This will prevent abrasive materials from entering the packing box and greatly prolongs packing life. When an external source of liquid is supplied, the supply pressure must be higher than pump pressure to insure liquid supply to the inner packing.

Various types and constructions of packing are available for pump application. Manufacturers usually provide a general service type packing, which operates satisfactorily for water service up to 250°F. Some liquids require special packing. Therefore, engineers should consult with manufacturers, and indicate the liquid being pumped, the liquid temperature, the duty point and the pump suction pressure.

3.8 Mechanical Seals

Disadvantages of using mechanical seals are:

- Seals are more expensive
- Seals are easily damaged
- Seals require disassembly of the pump for replacement

Advantages of using mechanical seals are:

- Seals eliminate leakage
- Seals require no periodic maintenance
- Seals eliminate shaft sleeve wear
- Seals prevent product contamination
- Seals can handle higher temperatures

To apply a mechanical seal in a pumping system satisfactorily, it must perform two minimum operating requirements. First, it must act as a check valve to keep liquid from escaping from the pumping system when suction pressure is below atmospheric pressure. The mechanical seal, in this situation, also prevents air from entering the pump shaft. Second, it must function as a bearing slider to accommodate the rotating and stationary members.

Figure 3-23 illustrates the components of a typical pump mechanical seal. The seal prevents leakage in three areas. The bellows prevents leakage between the seal and the pump shaft; the seat O-ring prevents leakage between the seat and pump bracket; the mechanical seal between the rotating and stationary members prevents leakage at the interface between the rotating washer and stationary seat.

The stationary seat is sometimes installed with a cup (see Figure 3-24) rather than with an O-ring. The cup offers more contact surface around the stationary seat and also acts as an insulator preventing heat dissipation from the seat through the adjacent material. For this reason, the stationary seat with an O-ring mounting is superior for pumps used for hot temperature service.

The spring which holds the two sealing faces together under a predetermined pressure is designed to insure against leakage. Distance A in Figure 3-23 must be controlled to insure the proper spring compression. If distance A is too long, the spring force will be reduced automatically and permit the faces to pull apart and leak. If distance A is too short, the spring will be compressed, resulting a high pressure between the faces, causing excessive seal wear and eventual failure.

The spring acts only to hold the seal faces together. It does not drive the rotating seal assembly. The bellows fit snugly between the shaft and drive ring. The drive ring is fitted with the retainer, and the carbon washer is fitted to the retainer in a similar manner. As the shaft rotates, it drives the carbon washer through this assembly of rubber bellows, driving ring and retainer.

System pressure and the spring hold the washer and seat together. System pressure acts against the seal cage and bellows from the spring side, and against the carbon washer through the interface and retainer OD. See Figure 3-25. Figure 3-25 shows that system pressure exerts force upon an area on the spring side, which is greater than the area on the carbon washer. Therefore, system pressure holds the carbon washer and seat together more tightly as pressure increases.

The system pressure will, sometimes, increase to a point where the washer and seat are pushed together so tightly that a suitable lubricant cannot reach the interface. Seal failure eventually will occur. For this reason, balanced mechanical seals are furnished for pressures above 200 psi, as shown in Figure 3-26. The contact area of the retainer and bellows on the spring side is approximately equal to the area of the carbon seat side of the rotating element. This will eventually minimize pressure force from pushing the washer and seat together too tightly.

When a pump is in operation, a relative motion between the carbon washer and the stationary seat exists. It is therefore imperative to have the mechanical seal functioning as a bearing slider. If lubrication is not provided between these two faces, friction will cause excessive wear, resulting in rapid failure. For this reason, care must be taken to insure that a fresh, clean fluid is delivered to the seal faces.

Care must also be taken to insure that no air is trapped around the outside diameter of the seal faces. This can be done by using some by-passing arrangement, as shown in Figure 3-27 to continually supply fresh liquid to the seal interface area. This by-pass arrangement prevents excessive temperature increase in the seal cavity water due to friction of the rubbing seal faces, as well as air binding.

Most design engineers feel that a seal permits no leakage. In actual application some leakage always get through the faces to insure proper lubrication. Often, a little oxidation can be noted outside the box near the shaft after a long period of time, as evidence of this slight leakage.

Pumping Abrasive Solutions

Mechanical seals can be successfully applied in systems containing chromate treatments up to 2000 ppm. Standard seal construction may be utilized up to 1250 ppm. Beyond that point, tungsten carbide stationary seals must be used.

When a mechanical seal is used in an abrasive system, a by-pass tubing should be installed between the pump casing and the seal chamber, as shown in Figure 3-27. This by-pass tubing maintains a steady supply of liquid to wash the seal interface area. The washing action keeps the light abrasive particles from settling around the seal interfaces. This will eventually prolong seal life.

In a closed system, where neither loss nor make-up of water is allowed, mechanical seals are superior to packing.

Iron and Iron Oxide

The main reason for seal failures is the existence of iron filings or black iron oxide in the pumping system. Iron filings are small particles of pipe threads or pipe materials which are broken off during pump installation or operation. It is imperative that a new system be cleaned to remove these particles.

The seal cavity is like a collection box for any particles left over in the system. These particles at the seal face grind away the carbon washer, resulting in premature failure.

Sometimes the iron particles are difficult to remove from the pumping system. In such cases, the seal by-pass tubing should be passed through an abrasive separator to remove the filings.

Iron oxide is an oxidation product of all steel or iron components in a system. This black material may fill the entire seal area, decreasing the flexibility of seal as well as the seal faces. Proper water treatment prevents this oxidation process. While the water treatment itself may introduce some abrasive materials to the system, the treatment is still superior to the formation of black iron oxide.

Most Engineers believe that iron filings and iron oxides are the main cause of premature seal failure. This has been confirmed by water analyses from systems causing seal failures in the field. Four out of five water samples sent in for testing were loaded with iron or iron oxide particles.

In the hydronic industry, almost all pumps are equipped with mechanical seals. Packing is used for cooling tower pumps, which uses part of the tower reclaimed water piped through the packing box to lubricate the packing. Most pumps, however, still are equipped with mechanical seals.

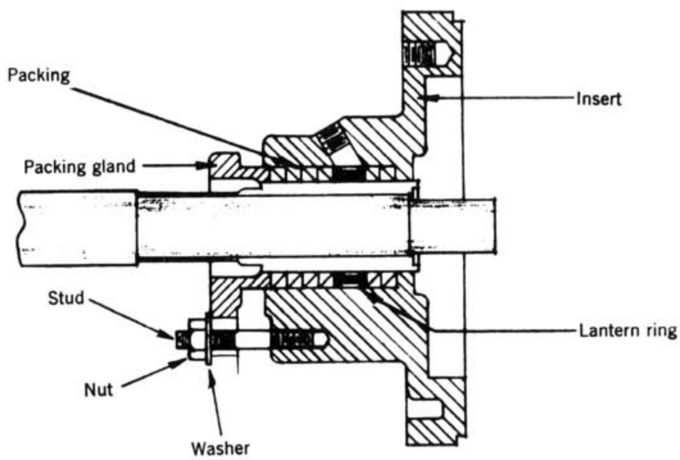


Figure 3-22. Pump sealed by packing.

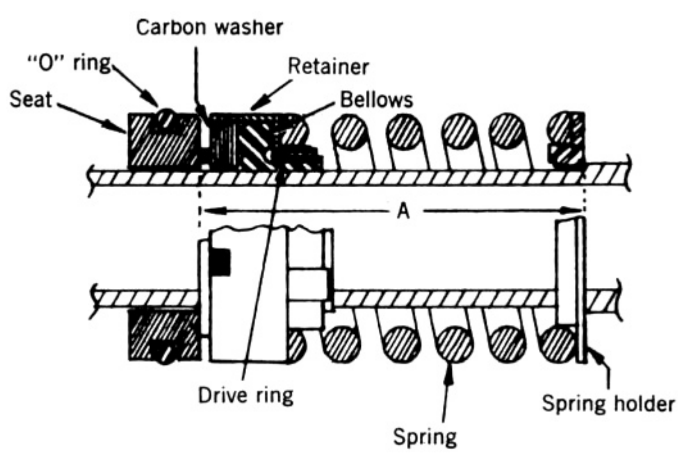


Figure 3-23. Mechanical Seal.

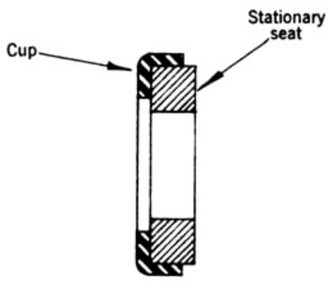


Figure 3-24. Seat with Cup.

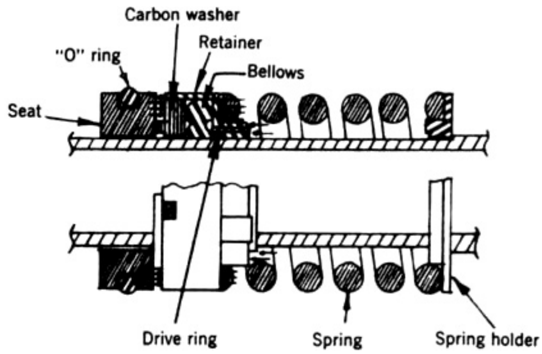


Figure 3-25. System Pressure Assists Spring.

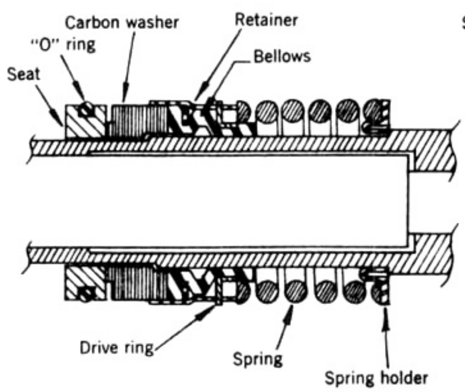


Figure 3-26. Balanced Seal Construction.

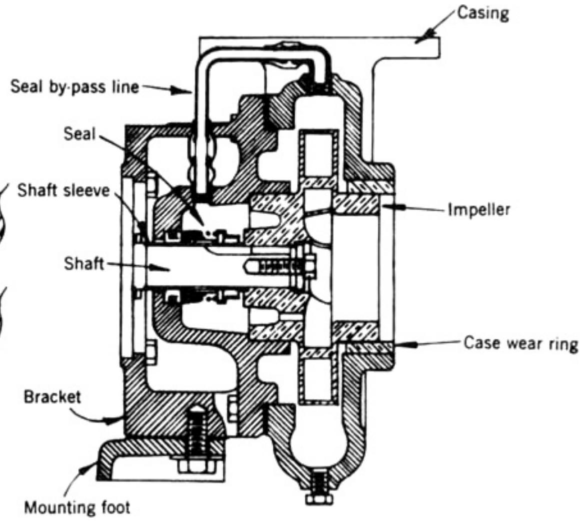


Figure 3-27. Bypass for Abrasive Pumping.

3.9 Noise and Vibration Control

The following items are some of the major sources of noise in pumps employed in plumbing systems:

- Unbalanced Motors
- Pulsation of the air mass flow from electric fans
- Pulsation of the magnetic field in the electric motor
- Motor/gear/pump journal and thrust bearings
- Contacting of the components in parallel-shaft and epicyclic gears
- Unbalance of the pump impeller
- Pulsation in the pumps
- Cavitation caused by the vaporization of the fluid and the subsequent rapid collapse of the vapor bubbles adds considerable impulsive high-frequency noise to the overall system. Impellers constructed with open-grain material will disintegrate because of the implosion effect of cavitation

Noise Control

Some of the possible modifications which the plumbing engineer might consider in order to reduce the noise output from the system are as follows:

- Gear box - a silencing enclosure or cladding should be provided
- Motor fan - a silencer should be provided at the air inlet and outlet or, if possible, the design should be modified
- Motor rotor - the number of slots should be changed or, if possible, its design should be modified
- Pump and pump bearings - sleeve type should be employed
- Pump operation - the pump should be operated near design flow conditions in order to achieve the correct system matching
- Pump impeller blades - the clearance between the tip of the impeller and cut water should be increased
- Impeller and guide-vane tips - should be dressed in order to reduce the thickness and intensity of the trailing wakes
- Out of balance - should be balanced to the fine limits
- Cavitation - the suction characteristics of the installation/ system should be improved

Vibration Control

Adequate criteria should be established for pump vibration to assure that there are no excessive forces which must be isolated or which will adversely affect the performance or the life span of the pumping equipment. There are many ways to develop pump vibration criteria. A simple, but satisfactory, approach would be to use the criteria which have been developed on the basis of the experience of persons and firms involved with vibration-testing of mechanical equipment in the building construction industry.

When providing vibration isolation for any pumping system or component, the engineer must consider and treat all possible vibration-transmission paths that may by-pass the primary vibration isolator. Flexible connectors are commonly used in piping and electrical cable connecting between isolated and unisolated pumping systems and components.

Concrete based are preferred for all floor-mounted pumps. It is common practice to isolate a pump in a manner similar to that illustrated in Figure 3-28.

3.10 Water Hammer Control

The physical parameters of a hydraulic system such as length, diameter, materials and thickness of pipe, the head, quantity of flow and friction losses can all be established from the design. Information on valves, pumps, or equipment characteristics can be obtained from manufacturers easily. Thus, there remains only the application of some fundamental principles to come up with the solution of a water hammer problem.

Typical examples are given in the following:

Example 3-1: If the initial water velocity in a rigid pipe is 5 ft/sec and if the initial pressure is 40 psi, what maximum pressure will result with rapid closure of a valve in the pipe? Assume $T_t = 60^oF$

Solution:

$$C = \sqrt{K/\rho}$$

$$h = \rho V_o C$$

where:

K = bulk modulus of water = 3.2×10^5 psi
ρ = density of water = 1.94 slugs/ft^3 at 60^oF
C = velocity of pressure wave
V_o = water velocity = 5 ft/sec
h = pressure rises

$$C = \sqrt{K/\rho} = \frac{320{,}000 \times 144}{1.94} = 4{,}800 \text{ ft/sec}$$

Therefore:

$$h = \rho V_o C = 1.94 \times 5 \times 4{,}800 = 46{,}560 \text{ psf} = 323 \text{ psi}$$

The maximum pressure is the initial pressure plus the pressure change, which is

$$P_{max} = 40 + 323 \approx 363 \text{ psi}$$

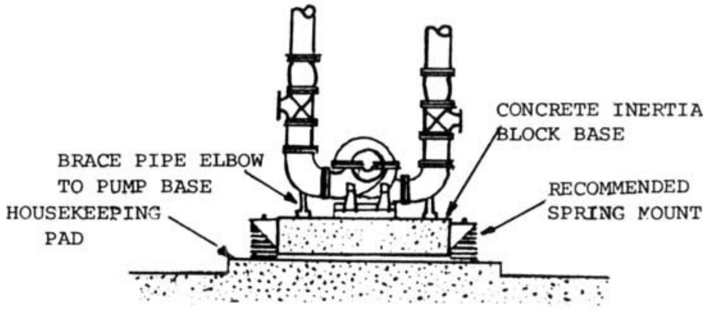


Figure 3-28. Vibration Isolation of Flexible-Coupled, Horizontally-Split Centrifugal Pumps.

Example 3-2: If the pipe in Example 1 leads from the source and is 3,800 ft. long, what is maximum time of closure for the generation of the maximum pressure of 363 psi

Solution: $T_c = 2L/C$

where: T_c = maximum time of closure (critical time) to produce maximum pressure
L = length of pipe = 3,800 ft
C = velocity of pressure wave = 4,800 ft/sec
$T_c = 2\text{x}3{,}800/4{,}800 = 1.58$ sec.

As indicated by Example 1, water hammer pressures can be quite large. Therefore, engineers must design hydraulic systems to keep the pressure within acceptable design limits.

Water Hammer in Pumping Systems

The major category deals with pumping systems connected to long discharge lines calling for water from the supply source to the discharge end. The regular sequence of pumping and stopping the pump station can be adjusted by timing the control valves, particularly the check valves on the pumping units, to a safe flow rate. This will limit pressure rises to a minimum.

Electric motors are most commonly used as the driving forces of a pump. It is called an emergency when the power supply is shut down. Power supply to the pump unit is cut off abruptly, the water column continues to flow along the pipes, comes to rest under the influence of inertial force and finally reverses its direction of flow. If no check valves are present in the line, the water column will reverse and drive the pump backward at a runaway speed. Pressure rises during this critical moment are affected by the pump characteristics and the hydraulics of the pipe line.

If some automatic type of check valve is installed, its closure time becomes critical, particularly if it is closed too rapidly. If the outward flow is cut off before the water column comes at rest, there is a tendency to generate a vacuum on the line, separating the water column followed by a regrouping of the column, and the corresponding instantaneous water hammer is thus produced.

If the flow is cut off at the zero flow, as with most swing check or tilting disc check valves, a surge equivalent to the downward surge generated by the tripping off the pumping unit would be produced.

If the valve is closed after the flow has reversed and reached some backward velocity, the rate of closure must be considered in relation to the pump characteristics, and to the safe flow at which such reverse flow can be tolerated.

There are many different approaches to the problem, and there is no single solution to water hammer problems, nor is there one well-designed valve that will totally eliminate all surges.

Water Hammer Control

Often, manufacturers do not recognize major changes in design practice or the progress of the art. Manufacturers should keep abreast of technical developments so their product will reflect the knowledge gained from experience and provide new methods for solving problems. Operating groups within the water works industry must pass their valuable experiences to manufacturers with the performance of device and obtain reliable information on whether or not the device is adequate for the service.

There are many types of water hammer control devices offered to the industry. They generally fall in the following categories:

- Relief valves actuated by overpressure
- Valves on pumping systems which open rapidly and close slowly to act as relief valves
- Controlled timing check valves and controlled timing regulating valves
- Air chambers installed either at the pumping station or at other points on the system
- Surge tanks (could be in the form of stand pipes)
- The use of reverse operating characteristics of pumping units to act as relief valves
- Spring loaded check valves of various types mounted on the discharge of pumping units

3.11 Drives

Electric Motors

When applying an electric motor, the following characteristics are important: (1) Mechanical arrangement, including position of motor and shaft, type of bearing, portibility desired, drive connection, mounting and space limitations; (2) desired speed range; (3) power requirement; (4) torque; (5) inertia; (6) starting frequency; and (7) ventilation requirements. Motor characteristics frequently used are generally presented in curve form.

The majority of pumps are driven by squirrel cage induction motors. Synchronous motors are also used for larger horsepowers, particularly if power factor correction is important. Wound motors are normally used for variable-speed drives. Reduced voltage starting and low inrush current motors are used for higher horsepower requirements, where a reduction in voltage at starting would adversely affect other users. In the selection of low inrush current motors, it is important to ensure the starting torque developed by the motor at its full speed is in excess of the torque requirements of the pump; if not, an overheat condition will occur. Water-cooled motors are quieter and offer advantages in larger horsepower applications where building air ventilation is required.

Engines

If electric power is unreliable or expensive, and environmental problems such as noise, vibration, space and maintenance are acceptable, the use of engines should be considered. Following are some of the selection parameters for any engine-driven system:

- Exhaust pipes or lines must be vented out-of-doors
- Noise levels must be acceptable to OSHA standards
- Cooling towers should be sized to handle the engine heat rejection. Exact figures for engine heat rejection should be obtained from the manufacturer. Towers should be oversized 30% in areas where the wet bulb temperature is 78°F or less. In areas with higher design temperature, towers should be oversized 40%
- Gaseous-fuel engines are physically larger than diesel engines with the same power output

If a pumping system is designed to operate on a continuous basis, it is better to operate an engine at a lower speed and install a gear chain to increase the speed to meet the pump requirements. If a stand-by pump is only required to operate during peak demand or during a power failure, then a higher speed engine should be used.

The various types and makes of engines as well as the speed and horsepower ratings of engines are extremely critical with regard to maintenance. Slow-speed units with higher capital cost and low maintenance costs, as compared to higher-speed units with low capital cost and high maintenance costs, can be determined only after a detailed cost analysis for each specific application has been made.

It is wise to choose a diesel rather than a gasoline or natural gas engine, if an engine-driven pump is required to start immediately during the electric power failure. Although the capital cost of the diesel engine is much higher than the gas engines, they are easier to start and more reliable. A well-maintained diesel engine will start immediately on the first try. Furthermore, a diesel engine will take full load almost immediately, while a gasoline engine needs to operate at its normal operating temperature.

Thrusts

Water initially enters a pump in an upward direction and is forced radially outward by the pump impeller. This change in the direction of flow pattern results in an upthrust which uplifts the pump shaft. As soon as the pump develops head, the upthrust diminishes and a downthrust develops. The magnitude of the downthrust is a function of discharge head and impeller areas. In the selection of a pump driver, whether it be an electric motor or a gear drive from an engine, it is essential to ensure the thrust bearings have adequate capacity to absorb the thrusts involved and a reasonable life span under normal usage conditions.

3.12 Motor Efficiency

The most important factors affecting motor efficiency are: sizing of the motor to the load, type of motor specified, motor design speed, and type of bearing specified. Oversizing the motor to the load results in poor efficiency. The type of motor specified is significant in selection of the "high-efficiency." For example, the lower the locked rotor torque specified for polyphase motors, the higher the obtainable design efficiency. Higher speed induction motors are inherently more efficient, as are ball bearings with rolling friction compared to sleeve bearings with sliding friction.

Total power required to drive a three-phase motor is a function of both efficiency and power factor. The formula is:

$$\text{KVA} = \frac{.746 \times \text{hp}}{\text{Efficiency} \times \text{Power Factor}}$$

The value of KVA which is the total power required to drive the motor will increase as both motor efficiency and power factor fall. For example, a typical 20 hp "high-efficiency" motor would be two percentage points higher in efficiency than a "standard" motor. This same high-efficiency motor might have a higher power factor of six percentage points. In larger sizes, the high efficiency and standard units approach each other in both efficiency and power factor. (See Figures 3-29 and 3-30). As shown, a "high-efficiency" motor primarily has an improved power factor.

The use of "high-efficiency" motors will eventually be helpful in any pumping systems design. But more emphasis should be placed on proper pump sizing and selection. Any motor, whether it is high efficiency or standard NEMA-type will have a lower power factor at reduced load. For example, a 25 hp motor at 40% reduced load will have an efficiency of 0.82 and a power factor of 0.55, as it compares to a full load efficiency of 0.86 and a full load power factor of 0.80. Furthermore, most of the time, the pump will operate at low flow. The pump performance curve will indicate that this motor will only draw 10 hp at reduced load. If a 15 hp pump motor combination was selected for this application, the reduced load efficiency would be 0.86

and the corresponding power factor would be 0.68. In this situation, the "high-efficiency" motor with lower cost can be reached. Figure 3-31 illustrates a plot of power factor and more efficiency at various loads.

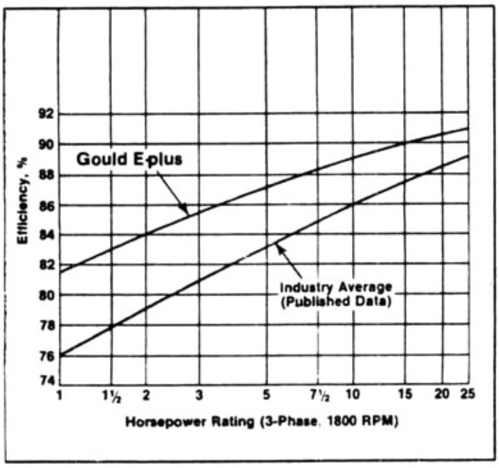


Figure 3-29. Efficiency Comparison.

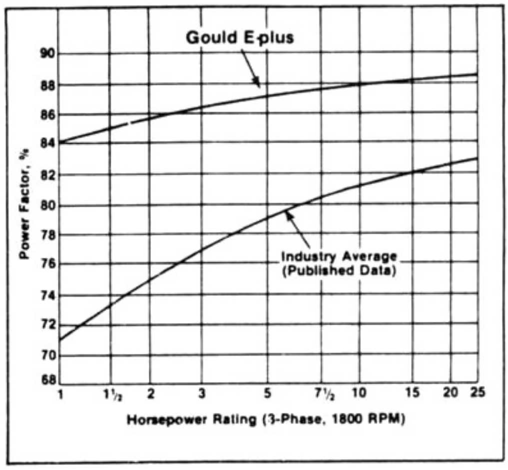


Figure 3-30. Power Factor Comparison.

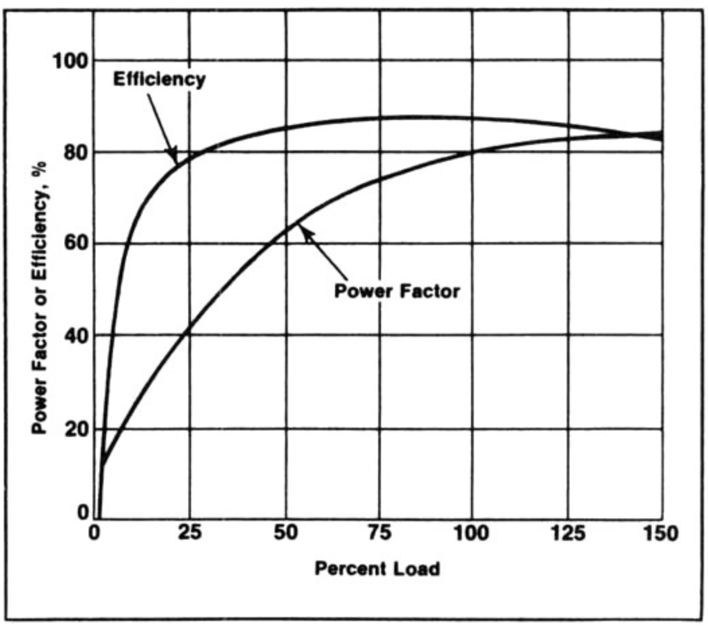


Figure 3-31. Efficiency and Power Factor of Typical Three-Phase Motor.

3.13 References

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5. "System Analysis for Pumping Equipment Selection," by Peerless Pump, Brochure B-4003, pp. 24-26, pp. 30-33, including Figs. 46-50, Figs. 55-60. Edited by permission

6. "System Analysis for Pumping Equipment," by Peerless Pump, pp. 8-11, including Fig. 10. Edited by permission.

7. "System Analysis for Pumping Equipment Selection," by Peerless Pump, Brochure B-4003, page 11, Table 4, edited by permission.

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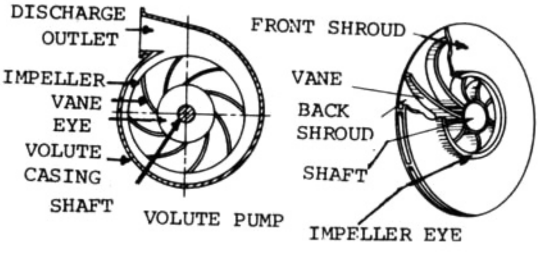
CHAPTER 4

TYPES OF PUMPS

4.1 Principles of Operation

The two major components of a centrifugal pump are a impeller and the casing in which it rotates (Figure 4-1). The pump operates by converting kinetic energy to velocity and pressure energy. Power from the driver is transmitted directly to the pump through the shaft, rotating the impeller at high speeds. This driver can be an electric motor, internal combustion engine or a steam turbine.

Centrifugal pumps used in plumbing systems are classified, on the basis of internal casing design, as volute or regenerative (turbine). On the basis of the main direction of discharge of liquid, impellers are classified as radial, axial or mixed flow. Other means of classification are: casing design (vertical or horizontal split case); axis of shaft rotation (vertical, horizontal or inclined); direction of pump suction or discharge (side, top or bottom); number of impellers or stages (single- or multistage); type of coupling of motor to pump (close-coupled or flexible coupled base-mounted); position of the pump in relation to the liquid supply (wet- or dry-pit mounted or in-line); and pump service (water, sewage, corrosive chemical, slurries, etc.).



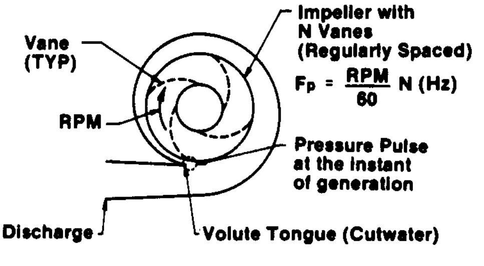


Figure 4-1. A Volute Casing and Impeller.

4.2 Horizontal Volute Pumps

The volute centrifugal pump is probably the most frequently specified by plumbing designers. Liquid entering at the eye of the impeller is thrown to the periphery in a progressively widening spiral casing, creating a partial vacuum and causing more liquid to flow.

As the pumped liquid spreads within the casing (gradually reducing in velocity), conversion of the velocity into pressure head results. Both turbulence and recirculation are reduced as the number of impeller vanes is increased.

The effect of reduced flow recirculation is to increase the developed head. The blade tip angle, however, has a profound effect on the centrifugal pump's characteristics. Enclosed impellers generate head between the two rotating impeller shrouds and one stationary casing wall. Open impellers generate head between two stationary casing walls.

A single-suction pump used one or several single-suction impellers; a double-suction pump used one or several double-suction impellers.

Most double-suction pumps have horizontally split casings, while single-suction pumps primarily have vertically split casings. One obvious advantage of horizontally split casings is they can be disassembled for service without removing the lower half from the piping.

An enclosed impeller (Figure 4-2) has several surfaces, relatively uncritical, and original efficiencies can be maintained over most of its useful life. Conversely, open or semi-open impellers (Figure 4-2), require close clearances between rotating vanes and the adjacent casing wall.

Wear in these conditions results in increased clearances, greater leakage losses and lower efficiencies. Open impellers are used in pumps discharging liquid containing suspended solids.

Figure 4-4 illustrates an end suction, single-stage, single-suction volute pump with an enclosed impeller. Its bearing must be designed to withstand axial and radial thrust resulting from unbalanced hydraulic pressures on the impeller.

Figure 4-5 illustrates a typical double-suction volute pump. It is normally operated at pressures and volumes higher than the single-suction volute. This is because the liquid is supplied from identical suction chambers located at each impeller side, substantially reducing hydraulic unbalance. Double-suction pumps, by virtue of lower velocities at the eye of the pump, may have advantageous lower net positive suction-head requirements.

The horizontal shaft, single-stage, double-suction volute pump (Figure 4-6) is the type most commonly applied to fire protection or commercial use. With these pumps, water flow from the suction inlet in the casing divides and enters the impeller from each side through an opening called the "eye."

Impeller rotation drives water by centrifugal force from the eye to the rim and through the casing volute to the pump discharge outlet. The kinetic energy acquired by water in its passage through the impeller is converted to pressure energy by gradual velocity reduction in the volute.

Some vertically split pumps must be removed from the piping system before they can be disassembled, but most of the vertically split case end suction pumps are available with a back-pullout design. This allows for removing the pumps from service without disturbing the piping or the pump body.

One of the ways that internal recirculation is reduced(or efficiency increased) is by the use of casing seal (or wearing) rings, either renewable or permanent. Worn renewable wearing rings can be readily replaced, restoring original internal pump clearances.

Permanent wearing rings cannot be replaced, and original

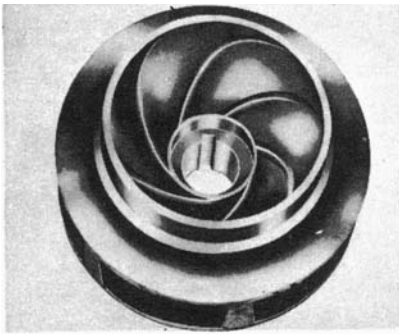


Figure 4-2. Illustration of an enclosed impeller.

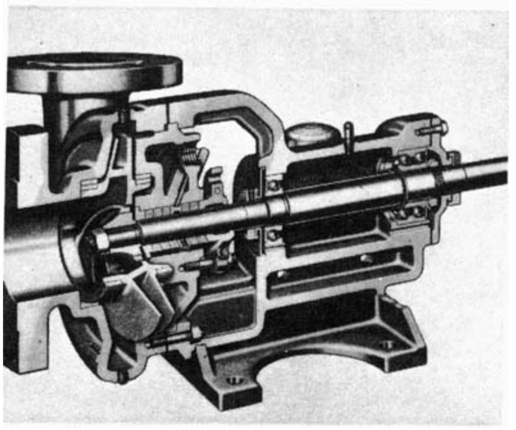


Figure 4-4. Typical end suction, single-stage and single-suction volute pump.

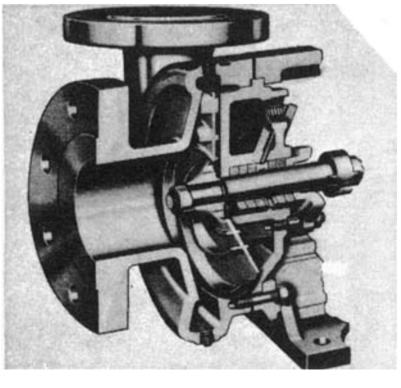


Figure 4-3. Illustration of Open Impeller.

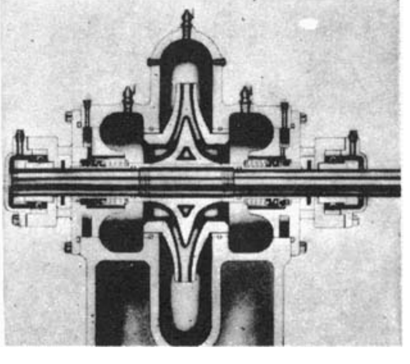


Figure 4-5. Typical double-suction volute pump.

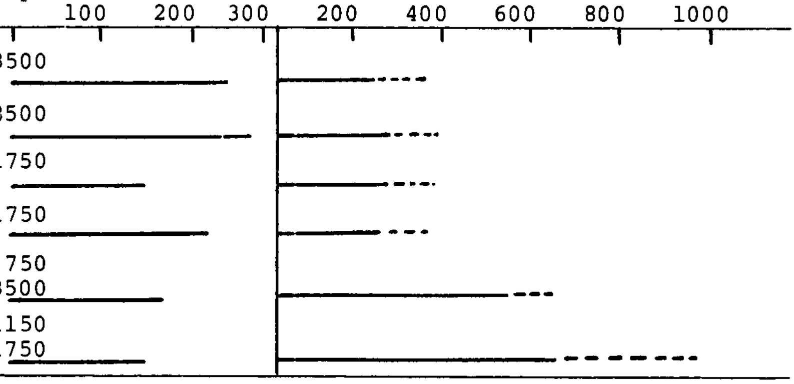
clearances can be restored only by machining the worn surfaces and installing replaceable rings; renewing worn surfaces by building them up by welding and re-machining; or replacing the impeller, the casing or both.

Replacement is the usual practice with the smaller, less costly pumps. Also, wearing rings are not used with open impellers, where close clearances between exposed vanes and the casing wall provide the required seal. Thus, open impellers are replaced when they become worn.

Table 4-1 shows the operating ranges of some common single-stage volute centrifugal pumps. For heads in excess of 500 feet (152.4 m), when operating at 3500 rpm, multistage pumps are required. Figure 4-7 illustrates a typical multistage volute centrifugal pump with horizontally split case and enclosed impellers.

Table 4-1. Operating ranges for common centrifugal pumps

Service*	Impeller	Rpm	Head, feet**(m): (30.5) 100	(61) 200	(91.4) 300	Operating temperature, °F***(°C): (93.3) 200	(204.4) 400	(315.6) 600	(426.7) 800	(537.8) 1000
General	Open	3500								
General	Closed	3500								
General	Open	1750								
General	Closed	1750								
Moderate	Closed	1750 3500								
Heavy-duty	Closed	1150 1750								



*All pumps are single-stage, single-suction designs. Medium-duty and heavy duty service pumps are both end-suction and top-suction pumps. All pumps have packed stuffing boxes.

**Heads taken at the best efficiency point.

***Key —— Operating ranges for cast-iron, bronze and ductile-iron pumps.

----- Operating range extensions with alloy pumps.

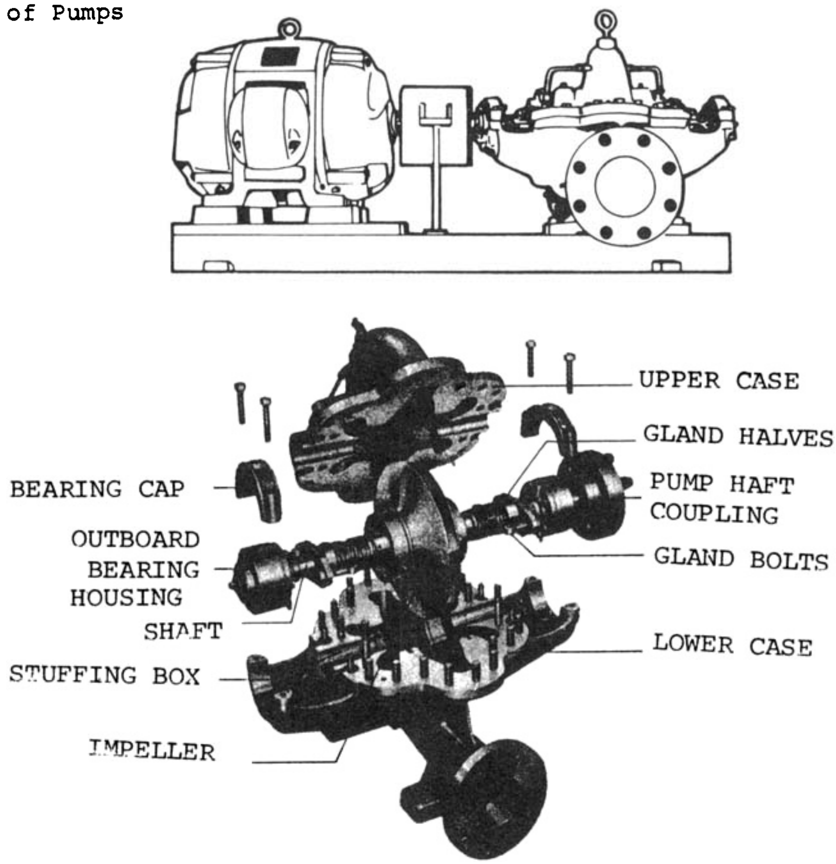


Figure 4-6. A Horizontal Shaft Single-Stage Centrifugal Pump, With Cutaway View of Pump.

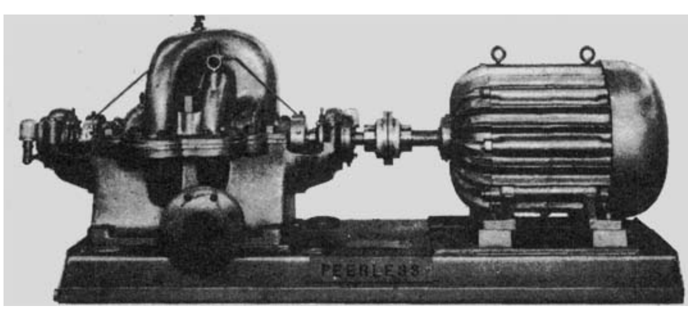
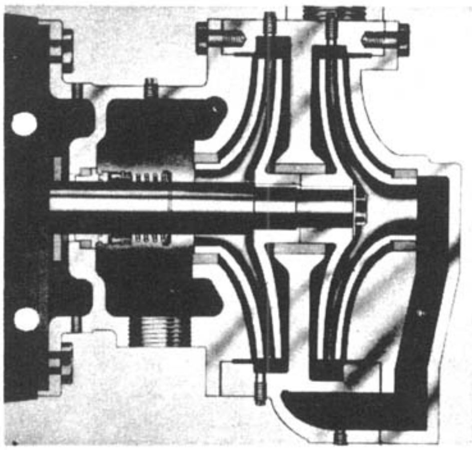


Figure 4-7. Typical Multistage Pump With Horizontally Split Case.

Figure 4-8 illustrates a single-suction vertically split case multistage pump. Its mechanical seal is under suction pressure, allowing the pump to operate at a higher discharge pressure using a standard mechanical seal.



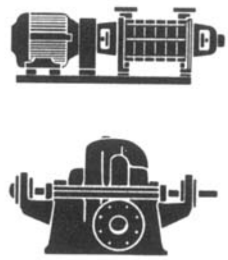


Figure 4-8. Typical Multistage Pump With Vertically Split Case.

4.3 Multistage Pumps

In order to give high pressure, two or more impellers and casings can be assembled on one shaft as a single unit, forming a multistage pump. Discharge from the first stage enters the suction of the second, and so on. Pump capacity is the rating in gallons per minute of one stage; the pressure rating is the sum of the individual stage's pressure ratings, minus a small head loss.

Multistage centrifugal pumps may use single- or double-suction impellers. Single-suction impellers are hydraulically unbalanced and, when used in multistage pumps, have equal numbers of nozzles discharging in opposite directions. Since double-suction impellers are not subject to hydraulic unbalance, they are not so limited.

Volute centrifugal pumps may be frame- or cradle-mounted, or close-coupled. In frame-mounted pumps, the motor drive is connected to the pump shaft through a flexible coupling (see Figure 4-7).

In close-coupled pumps (See Figure 4-9), the impeller is mounted directly on the motor shaft. However, cradle-mounted units with oil-lubricated bearings must be used horizontally. Although use of close-coupled pumps virtually eliminates alignment problems, their service is limited by temperature. The motor on frame-mounted pumps can be changed and repaired without breaking into the piping.

4.4 Multistage Diffuser Pumps

The multistage diffuser pump essentially escapes both ends of the trap. It has the same head curve as the typical in-line circulator, except that it can develop high head pressures. It squeezes off with virtually no low-flow hydraulic turbulence at the same time that its diffuser construction chokes off potentially dangerous high-flow rates. This keeps it out of destructive cavitation.

An example of this pump-type is the industrial vertical turbine pump equipped with mounting barrel and floor flange. It may also be equipped with a single mechanical shaft seal, replaceable without removing pump or motor. Solidly coupled, typically to a vertical hollow-shaft motor, this pump can be standardly built to develop head pressures in excess of 500 psi. Its cost level is about the same as a conventional two-stage horizontal split-case pump.

For modest duty, multistage diffuser pumps are available at relatively low costs in a horizontal design for either 1750 or 3500 rpm operation. Generally, a higher speed is more economical in first cost and is physically smaller in size, since impeller diameter for a given duty is much smaller.

4.5 In-line Pumps

In-line centrifugal pumps (Figure 4-10) often are used in plumbing services. They are available in capacities up to 1000 gpm (63.1L/s) at 450 feet (137.2 m) of head, and in a wide range of construction materials.

Initial and installation costs are approximately 30 to 40 percent lower than comparable conventional volute pumps since they require less space, need no permanent foundation and are easily maintained.

The in-line pump consists of a flanged head mounted directly in the piping, usually with a close-coupled drive positioned above

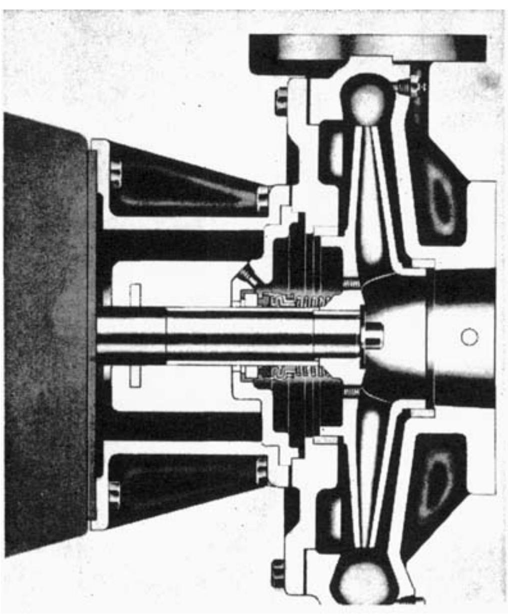


Figure 4-9. Typical Close-Coupled Pump.

the pump casing. Although smaller in-line pumps, similar to (Figure 4-11), are often mounted directly in horizontal or vertical piping, larger pumps usually require a rest plate to prevent undue strain on the piping (Figure 4-10). For this reason, larger pumps are usually mounted in the vertical position.

The simplicity of the connection between the drive and pump (four to eight studs) permits rapid disassembly. To repair the pump, the studs are merely loosened and the entire motor-impeller unit is lifted clear of the pump casing. As piping remains intact and realignment is unnecessary on reassembly, downtime is minimal.

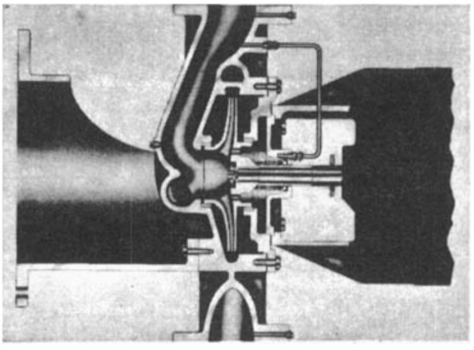


Figure 4-10. Typical In-line Pump (Larger Unit).

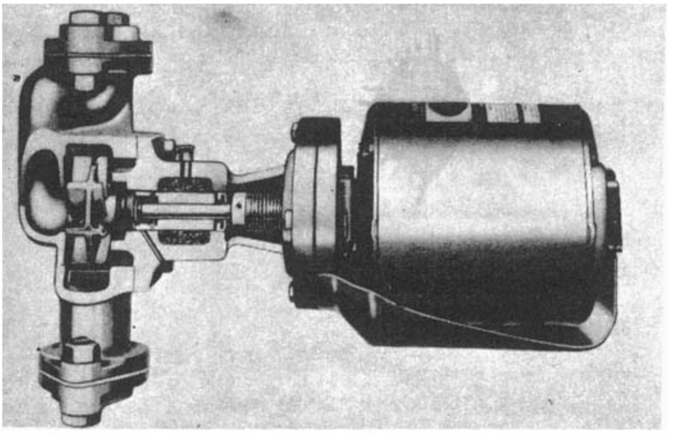


Figure 4-11. Typical In-line Pump (Smaller Unit).

4.6 Sewage Pumps

Sewage pumps are designed to transport liquid-solid mixtures without clogging, and are usually equipped with large suction and discharge ports free of obstructions (See Figure 4-12).

Non-clogging pumps usually are equipped with single- or two-port impellers designed to pass materials that would otherwise build up and clog other pump types. The impellers have rounded edges and large vane clearances.

Narrow repelling vanes on the outer shroud operate to dislodge solids caught between the shroud and the casing wall. The shaft is usually kept clear of the suction port and chamber to avoid clogging prior to reaching the impeller eye. End clearance impeller and casing wearing rings are usually provided to maintain optimum clearance for efficient operation.

4.7 Grinder Pumps

A grinder pump is a special class of sewage pump with an outstanding solids handling capability. This capability is achieved by incorporating an integral inline grinder as a part of the pumping machinery. The grinder pump is capable of handling both normal sewage solids and a wide variety of foreign objects often found in sewage.

By reducing the solids to tiny particles, the pump's passages and discharge piping can be much smaller than conventional sizes. This results in a higher efficiency and makes possible a high head, low flow pump for residential applications using 1 HP (745.7W) motors. Since the piping is smaller, scouring velocities are achieved at much lower rates.

Grinder pumps are applicable in completely pressurized sewage collection systems, as well as in lifting sewage from a residence into a nearby gravity sewer at a higher elevation. Grinder pumps are available as both semi-positive displacement (screw) pumps and modified centrifugals.

Centrifugal grinder pumps are made with steep characteristic curves. Since parallel operation of centrifugal pumps into a common pressure header may drive some of them to heads above shut-off, pumps with a nearly vertical curve should be specified for such system applications.

The positive displacement pump discharges at a nearly constant rate, even over extremely wide fluctuations in head. This means that a single-model pump can work equally well anywhere in a complex system. Its performance is predictable and consistent, whether it operates alone at a particular moment, or sharing the pipeline with several other pumps.

The key factor in pipe sizing for pressure sewer systems is the design flow. This is based on a maximum number of pumps operating simultaneously. Based on mathematical analysis and empirical relations, the recommended design flows have been developed for a pressure sewer system using positive displacement pumps. These flows are shown in Table 4-2 for various numbers of pumps connected.

Table 4-2. Maximum number of grinder pump cores operating daily

Number of grinder pump cores connected	Maximum daily number of grinder pump cores operating simultaneously
1	1
2 - 3	2
4 - 9	3
10 - 18	4
19 - 30	5
31 - 50	6
51 - 80	7
81 - 113	8
114 - 146	9
147 - 179	10
180 - 212	11
213 - 245	12
246 - 278	13
279 - 311	14
312 - 344	15

Design Flow = Maximum number of pumps running simultaneously x 11 gallons per minute (0.69 liters per second).

4.8 Vertical Pit-Mounted Pumps

Vertical volute pump impellers discharge radially and horizontally against that casing section known as the bowl. The resulting pressure forces liquid up the vertical discharge column.

The pumping chamber of the wet-pit mounted pump is located below the liquid supply level, with the discharge line usually elevated to floor or grade level in duplex configuration (Figure 4-13). Bottom suction units are most common, but side suction units are also available.

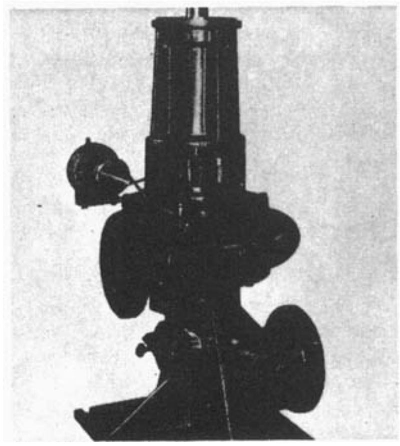


Figure 4-12. Typical Sewage Pump.

By contrast, the pumping chamber of dry-pit mounted vertical volute pumps is located above the liquid supply level (Figure 4-14).

Some vertical pumps have cantilevered-impeller shafts to avoid support problems with bearings, bushings and sleeves. Shafts are supported by two points at their top, with the remainder of the shaft running completely free. The shaft must be large in diameter and short, to prevent excessive vibration, rarely extending more than 6 or 7 feet (1.8 or 2.1 m) below the bottom outboard steady bearing. For greater lengths, center support bearings are placed at approximately 4 feet (1.2 m).

In some wet-pit installations, long shafts can be avoided by using a close-coupled, totally submersible motor located beneath the pump chamber (Figure 4-15). The motors are filled with either oil or water.

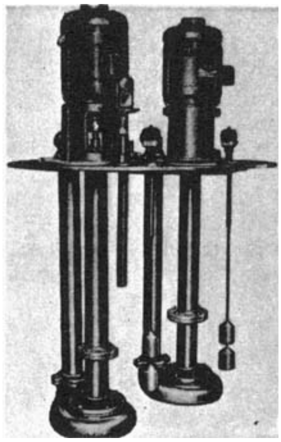
Oil-filled units have hermetically-sealed stator windings and a pressure-equalizing diaphragm to compensate for expansion and contraction under varying temperatures. A mechanical seal prevents solid particles from entering the motors area. Oil-filled motors are completely isolated from the pumped liquid.

A double mechanical seal assembly prevents oil from leaking out or foreign material from entering the motor. A moisture sensor probe indicates seal failure. Submersible motor volute pumps require less space at ground level and use less power than comparable long shaft units.

4.9 Regenerative Pumps

In a regenerative pump (commonly called a turbine-type pump), the liquid does not discharge freely from the tip (Figure 4-16). Instead, it circulates back to a lower point on the impeller diameter and recirculates several times before leaving the impeller. This recirculation enables these pumps to develop heads several times that of volute pumps with the same impeller diameter and speed.

Because of the close clearances between the impeller and casing, only clear liquids can be pumped. Regenerative turbine-type pumps are also available in multistage units, in capacities from 1 to 200 gpm (0.06 to 12.6L/s). Single-stage units generate heads up to 500 feet (152.4 m); multistage units (e.g., 5-stage) can generate heads up to 2500 feet (762 m).



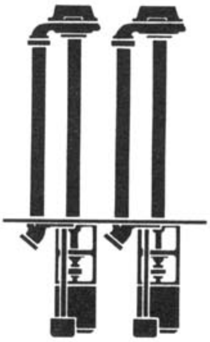


Figure 4-13. Typical Wet-pit-mounted Vertical Pump.

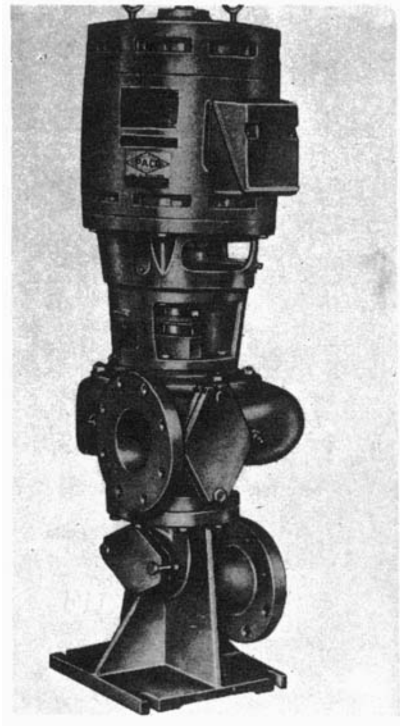


Figure 4-14. Typical Dry-pit-mounted Vertical Pump.

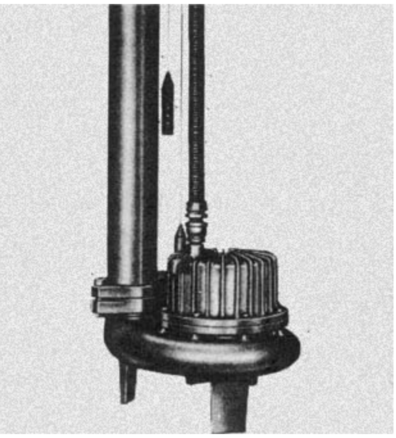


Figure 4-15. Typical Submersible Pump.

Vertical turbine pumps may be water or oil lubricated. The drive shaft of water lubricated units is located directly in the path of the discharged process liquid which lubricates the shaft bushings, packing and stabilizers. This pump type is used for raising water to the surface from deep wells and is often called a deep-well pump (Figure 4-17).

Water lubricated pumps are prone to wear at the shaft packing and bushings. When used with abrasive or corrosive liquids, the pump will be subject to excessive maintenance problems, and oil-lubricated pumps are used for the latter service. However, these pumps tend to contaminate the fluid being pumped.

In oil-lubricated pumps, a tube enclosing the drive shaft is filled with oil for the bushings and sleeves. Thus, the internal parts do not come into contact with the liquid pumped.

Oil and water lubricated vertical deep-well pumps often use a screen or strainer in the suction line to prevent clogging. To permit satisfactory solids disengagement, the screen area should be about four times the eye area of the impeller.

A short version of the vertical turbine pump, sometimes called the short coupled pot pump (similar to Figure 4-18), is often mounted in a sump with the motor above the ground. It is used widely in fire protection systems, industrial waste handling, pipe line booster services and for pumping water from lakes or slumps.

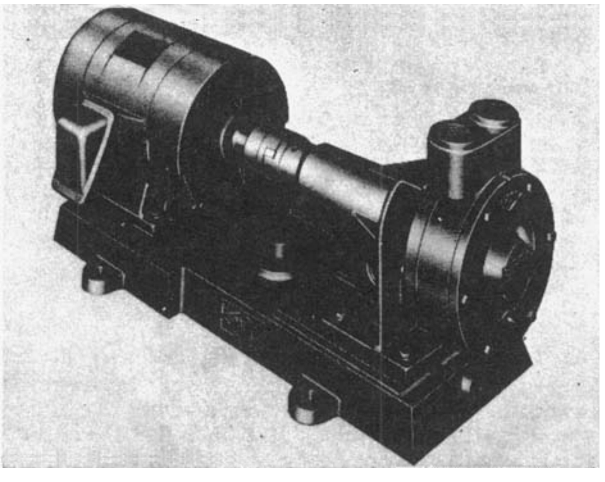


Figure 4-16. Typical Regenerative (Turbine Type) Pump.

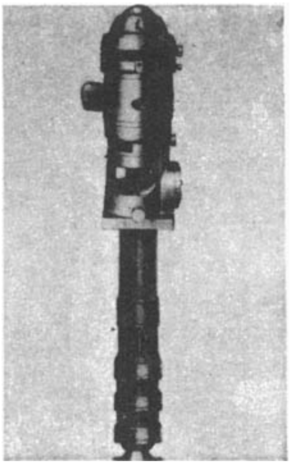


Figure 4-17. Typical Vertical Turbine.

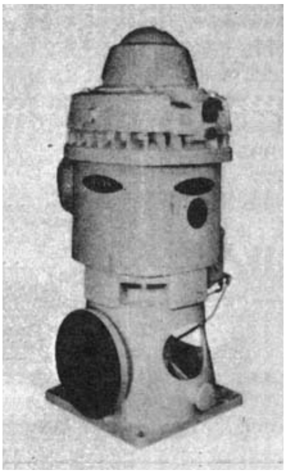


Figure 4-18.
Typical Pot
Type Turbine.

4.10 High Pressure Service Pumps

Single-stage pumps can be designed for high pressure service (i.e. fire or booster service) by increasing the impeller diameter or the rated speed. Both these methods offer certain undersirable features. The larger diameter pumps may be less efficient, and high speed pumps may not readily be matched to the driver.

To give high pressures, two or more impellers and casings can be assembled on one shaft as a single unit, forming a multistage pump. Output of each prior stage becomes the input to the next stage, so that the heads generated by each impeller are culmulative.

The pump capacity is the rating in gallons per minute of one stage; the pressure rating is the sum of the pressure ratings of the individual stages, minus a small head loss.

Both horizontal and vertical multistage pumps can be developed for high pressure service.

4.11 Fire Pumps

Fire pumps often are used to supplement supplies available from public mains, gravity tanks, reservoirs, pressure tanks or other sources.

The first modern fire pumps were the wheel-and-crank reciprocating-type, belt-driven from mill machinery. If plant operations were stopped during a fire, the pump could not operate. At best, these pumps were inadequate.

Better water supplies became necessary as automatic sprinkler systems became more common, and mill pumps were replaced by rotary displacement pumps driven by friction forces from the horizontal water wheels supplying power to the plant. As steam supplanted water power, the reciprocating driven pump was adopted for fire protection.

For many years the Underwriter duplex, double-acting, direct steam-driven pump was universally accepted as the "standard" fire pump.

Today the centrifugal fire pump is standard. Its compactness, reliability, easy maintenance, hydraulic characteristics and a variety of available drivers (electric motors, steam turbines and internal combustion engines) made the Underwriter pump obsolete, although not entirely extinct.

An outstanding feature of a horizontal or vertical centrifugal pump is the relationship of discharge to pressure at constant speed, causing pressure head to increase and discharge flow to reduce. With displacement pumps, however, the rated capacity can be maintained against any head if the power is adequate to rotate the pump at rated speed and if the pump, fittings and piping can withstand the pressure.

Horizontal and vertical fire pumps are available with capacities up to 5,000 gpm. Pressure ratings range from 200 psi for horizontal pumps and 75 to 280 psi for vertical turbine pumps. It is anticipated larger capacity fire pumps will be listed in the future.

The size of a horizontal centrifugal pump is generally the diameter of the discharge outlet. However, it is sometimes indicated by both suction and discharge pipe flange diameters. The "size" of a vertical turbine pump (Figure 4-19) is the diameter of the pump column.

Vertical
Fire Pumps

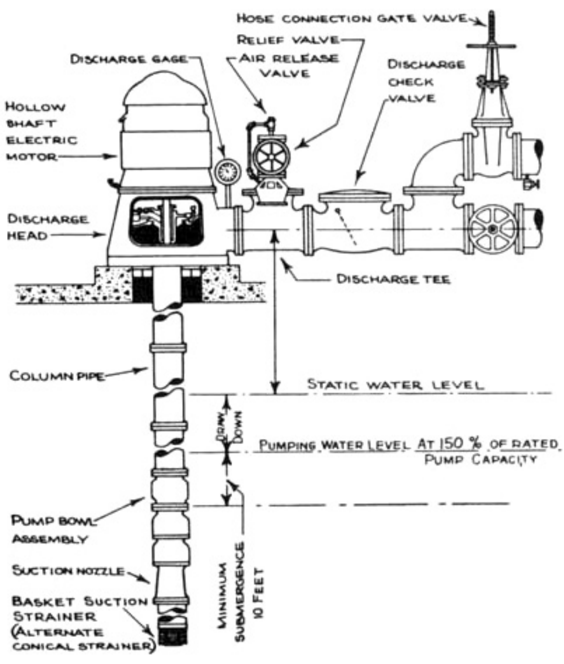


Figure 4-19. A Vertical Turbine Fire Pump.

These are fire pumps taking suction from the public water mains, industrial systems or power penstocks. (In a mechanical sense, all pumps are booster pumps.) As a prelude to installation, available fire flow in the area is obtained by testing.

Full overload capacity of the pump plus probable flow drain from hydrants in the area by the fire department are calculated. The pressure in the water mains should not be allowed to drop below 20 psi. Head rating of the pump should be sufficient to meet all pipe friction in the connection, plus pressure demand.

Vertical turbine-type pumps were originally designed to pump water from bored wells. As fire pumps, they are recommended in instances where horizontal pumps operate with suction lift. An outstanding feature of vertical pumps is their ability to operate without priming. (See the NFPA Pumps Standard for required submergence).

Vertical pumps may be used to pump from streams, ponds, wet pits, etc., as well as to booster service. Suction from wells is not recommended for fire service, although it is acceptable if the adequacy and reliability of the well is established, and the entire installation is made in conformance with the NFPA Fire Pump Standard.

In many instances, the cost of a deep well fire pump installation is prohibitive, especially if the pumping level at maximum rate is more than 50 ft below ground level (200 feet is the limit.)

If the yield from a reliable well is too small to supply a standard fire pump, low capacity well pumps can be used fill conventional ground level tanks or reservoirs for the fire pump supply.

A typical vertical fire pump consists essentially of a motor head or right-angle gear drive, a column pump and discharge fitting, an open or enclose drive shaft, a bowl assembly (containing the impellers), and a suction strainer.

The principle of operation is comparable to a multistage horizontal centrifugal pump. Except for shutoff pressure, the characteristic curve is the same as for horizontal pumps.

Vertical pumps have the same standard capacity ratings as horizontal fire pumps. Pressure ratings are not standardized. By changing the number of stages and the impeller diameters or both, the pump manufacturer can provide a specific total head at rated speeds.

For electrically driven pumps, hollow shaft motors are used. Diesel engines or steam turbines can be used by means of right angle gear heads.

4.12 Prefabricated Systems

Water pressure boosting systems are available in either knockdown (field erected) or completely prefabricated. Knockdown systems require the contractor mount equipment, pipe the system and install controls. This can add up to considerable piping and wiring.

The prefabricated system is a substantial convenience to everyone, especially if the system is tested after fabrication. Testing should include a complete running test with actual suction and discharge pressures set at the factory. All alarm conditions should be simulated.

Where the manufacturer has the facility to plot the system pressure versus flow rates, the designer or owner can get a visual print-out of the exact performance of the system. This complete test is to the advantage of the manufacturer as well as the contractor, designer and owner.

Should any on-site conditions indicate an improper performance, the problem should be readily diagnosed and corrected by minor adjustments.

The prefabricated system is advantageous from an operational point of view. The installed cost is usually less. Assembly is done by experienced technicians. Furthermore, almost without exception, the prefabricated system will require less floor space than will a knockdown-type system.

Prefabricated systems also enable the mechanical contractor to minimize his involvement with other building trades. In other words, the electrician only needs to bring power to the control panel before the system can be operational.

4.13 Modified Centrifugals Used in WFI Systems

A standard, commercially available, sanitary designed centrifugal pump can be modified for use in water for injection systems. Additional design requirements must be incorporated to provide specialized requisites for pharmaceutical applications, i.e., provisions for sterilization by in-place steam sterilization. The pump and seal must be designed to permit full penetration of steam under pressure to achieve and maintain sterilizing conditions. Condensate must be removed from pockets which require revisions of the standard design. The volute casing was modified to provide drainage by welding a sanitary fitting to the lowest point of the volute casing. Seal flush piping was designed to permit drainage. (See Figure 4-20).

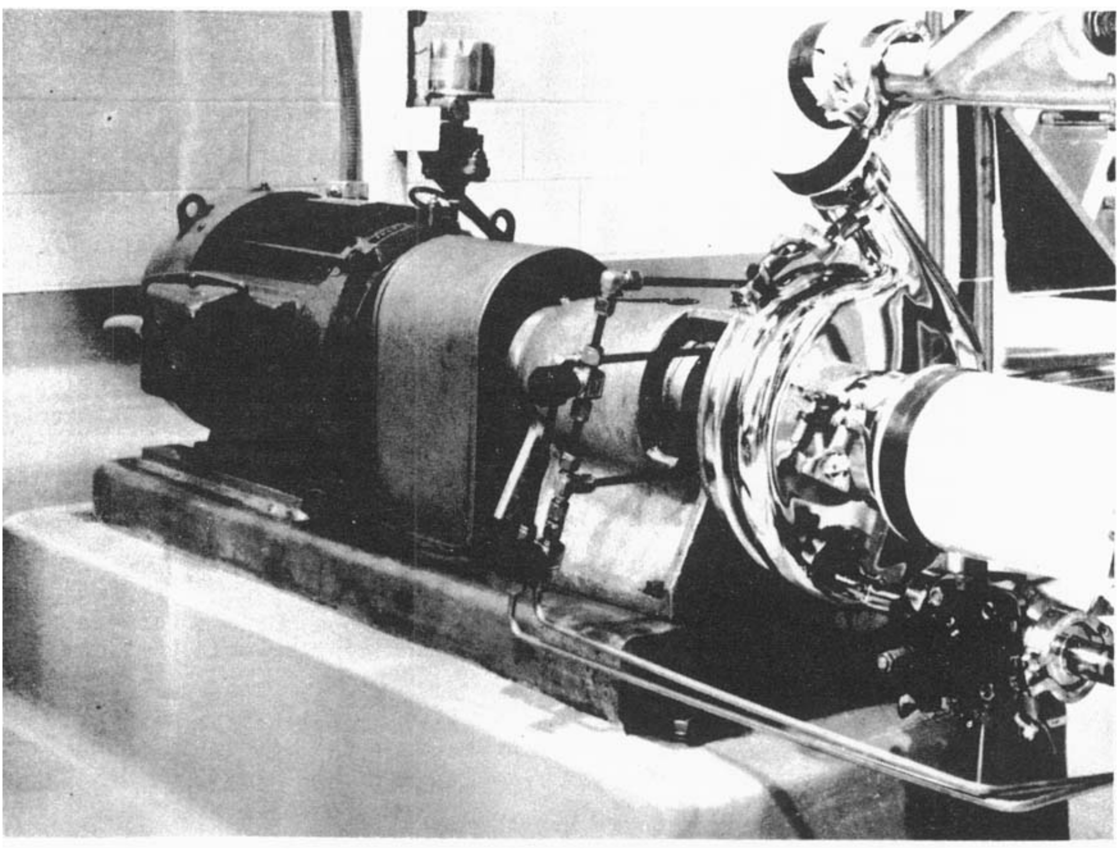


Figure 4-20. A Specially Modified Sanitary Centrifugal Pump Used For WFI Systems.

4.14 References

1. "Centrifugal Pumps in Plumbing Service - Part I," by Milton Meckler, P.E., 1975/76 ASPE Data Book, Chapter 12, pages 1-5. Reprinted with permission.

2. "Centrifugal Pumps in Plumbing Service - Part II," by Milton Meckler, P.E., 1975/76 ASPE Data Book, Chapter 12, pp. 6-10. Edited by permission.

3. "Fire Pumps," by Robert M. Hodnett, Fire Protection Handbook, 1981, Section 16, Chapter 6, pg. 65, 66-68, 69, 70. (Figs. 16-6A, 16-6C to 16-F.

CHAPTER 5

WATER PRESSURE BOOSTING SYSTEMS AND COMPONENTS

5.1 INTRODUCTION

The water pressure boosting system is one of the most complicated parts of a building's plumbing system. In light of the little information available, the designer is faced with many decisions attempting to apply those commercially available items to design the water pressure boosting system.

In an attempt to offer a cohesive document that points out the key aspects to consider in making this design selection, the following discussion is presented to list some of the choices that the designer has, as well as some of the fundamental problems that the designer must consider prior to designing or specifying.

5.2 Primary Equipment

The typical pump-motor unit of water pressure boosting systems can be classified into three categories:

- Constant speed motors and pumps
- Constant speed motors and variable speed pumps
- Variable speed motors and pumps

Constant speed motor and pump units include those utilizing close-coupled pumps or frame-mounted flexible-coupled pumps. The close-coupled configuration can be either the standard horizontal arrangement with an end-suction centrifugal or vertical-mounted end-suction centrifugal (i.e. a vertical inline.) Frame-mounted flexible-coupled units include end-suction centrifugal frame-type pumps or split-case double-suction frame-type pumps. The split-case can be furnished either the horizontal shaft arrangement or a vertical shaft arrangement. The vertical multi-stage diffuser-type pump and motor are independent but are rigidly coupled.

The constant-speed motors with variable-speed pump units normally utilize the frame-mounted pumps and a variable-speed drive. The most popular of these is the fluid coupling unit. The variable-speed motor and pump units usually utilize frame-type pumps and a high-slip variable-speed motor.

The constant-speed motor and pump unit offers the most efficient operating assembly because there is not energy lost in the driving mechanism. Generally, the other choices are made in order to gain control of the discharge pressure and flow.

5.3 Centrifugal Vs. Diffuser Impeller Pumps

The centrifugal impeller has inherent limitations, but it is generally the less expensive unit. The centrifugal impeller is basically the same regardless of what type mounting and shafting arrangement is selected. Closed-coupled pumps mount the impeller directly to the motor shaft and the shaft and bearings take the shaft loads generated by the impeller. The frame-mounted end-suction utilizes the standard foot-mounted or vertical-mounted motor and the frame contains the pump shaft and bearings. This eventually eliminates the need for the motor to withstand the pump thrust loading, but it adds a separate set of bearings that must be maintained. It also creates the problem of maintaining coupling alignment. The split-case double-suction pump gains the advantage of lower axial shaft loading but adds significantly to the cost of the pumping unit.

Multi-stage diffuser-type pumps are primarily used in the vertical configuration. The pump discharge head acts as a motor stand with a register fit allowing the coupling alignment to be held closely. The coupling can be bolted or threaded directly to the motor shaft. The diffuser-type pump is a modified centrifugal with significant differences.

To allow operation at higher speeds, the impeller diameters are smaller for selection used in booster systems. Pumps are usually furnished with multiple stages, and each stage includes a diffuser which smoothly diverts water to the next stage. The bearings are water lubricated and require no maintenance. Pumps can operate at reduced flows because of a close dimension between bearings. Mechanical seal can be repaired easily without pump disassembly or motor removal. (See Figure 5-1).

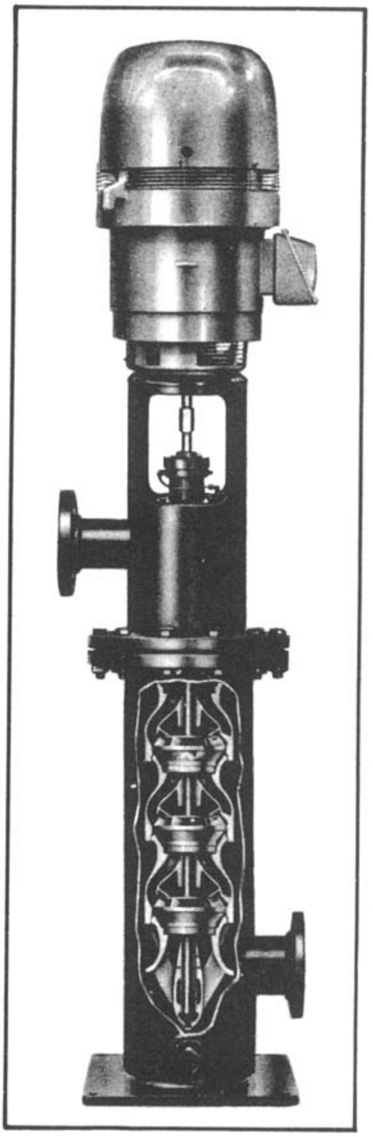


Figure 5-1. Vertical Multi-Stage Diffuser Pump With Hollow Shaft Motor.

5.4 Sizing

The most widely used sizing technique for water pressure boosting systems has been the calculation of fixture units and graphs or charts which relate fixture units versus flow rate in gallons per minute. This curve is called Hunter's curve, and it is accepted in many model codes as the basic criterion for estimating flow rate.

Almost every practicing engineer realizes that Hunter's curve is grossly oversized for the usual office building, apartment building, or hospital; but in the absence of current information a designer is reluctant to select pumping equipment with less capacity than is shown on this curve. Many studies have been made, but the results of these studies have never been put into a nationally accepted code which engineers can point to as a new industry standard. A new design guide or standard should be developed through extensive testing by independent industry committees or non-profit research organizations.

After the total flow rate has been established, the designer must establish a capacity for each pump in the pumping system. As a general practice, the designers will select either a three-pump system with a small lead pump or a two-pump system with a device to shut down operations during periods of low flow. Apartment buildings, nursing homes, office buildings, sports complexes, schools, factories, etc., often experience periods of low flow. However, hospitals and large hotel/motel complexes are less likely to have these long periods of low flow.

If a three-pump system is selected, the designer has to choose the pump "splits." The most popular split is 20/40/40, depending on whether or not to have some standby capacity. If a two-pump system is selected, the split is usually 65/65. For the systems that require the utmost in economy, the split is 50/50. After the split has been determined, it becomes a relatively simple process to select the pumps to meet the design capacity. In head calculations, consideration must be given to static head, friction loss, and pressure required at the highest outlet fixture. After the total discharge pressure or system pressure is calculated, the suction pressure must be deducted to get the boost required by the pumping system.

5.5 Pressure Control

The typical pressure controls used on water pressure boosting systems are variable-speed devices and pressure regulating valves.

The concept of variable speed is to increase or decrease the speed of the pump impeller to maintain a constant discharge pressure at varying flow rates. Figure 5-2 illustrates the result of varying speed in pump discharge pressure and flow. In general, the steeper the pump curve the more advantage there is in varying the pump speed and maintaining a constant system pressure to save horsepower. Variable-speed systems are normally using a centrifugal-type pump, and the shape of the centrifugal curve will only allow for minor variation in speed. (A more steeply sloping curve would result from a significant variation in speed.) Variable-speed pump selections rarely allow more than a ten percent variation of pump speed while maintaining system pressure at constant suction.

Pressure regulating valves on the discharge of each pump provides a simple way for maintaining constant system pressure. A quality regulating valve can be very dependable and operate for many years while maintaining a constant system pressure. As pump discharge pressure varies, the control valve modulates to maintain system pressure.

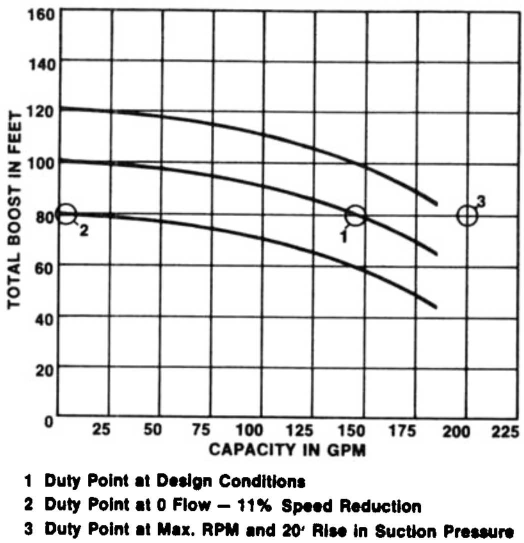


Figure 5-2.

A globe pattern valve with a soft seat for tight shut-off is a typical regulating device called a pilot-operated pressure-regulating valve (see Figure 5-3). A small amount of water is diverted from the inlet into the diaphragm chamber, as well as through the pilot regulator. The pilot will allow more water to enter the discharge, and therefore less to the diaphragm which opens the valve. If less water is required, the pilot diverts less to the discharge and the valve closes.

5.6 System Location

Typically, a water pressure boosting system is located in the mechanical room of a building's basement. However, location in other parts of the building should be considered if there are advantages.

Normally, suction pressures can accommodate the lower floors of a building. In the case of a motel complex, this can mean a substantial savings in water pressure boosting equipment because the major water-using equipment is normally located on the lower floors. Even if the booster system must be located in the basement, a separate supply line can be used to feed these heavy water-using fixtures. The boosted water is then simply expressed to the first floor, where the boosted pressure is required.

Hot and cold water pressure should be equalized in a building. Therefore, it is necessary to carefully consider hot water aspects when locating the booster system. The hot water storage tank can be a lower design pressure vessel if located on the top of a building, but power or gas/steam lines might be more costly.

Several methods can be used to get water to the required floors in high-rise buildings. Expert advice is usually available to the engineer in arriving at the method most suited for a particular job. Equipment costs, energy requirements and future

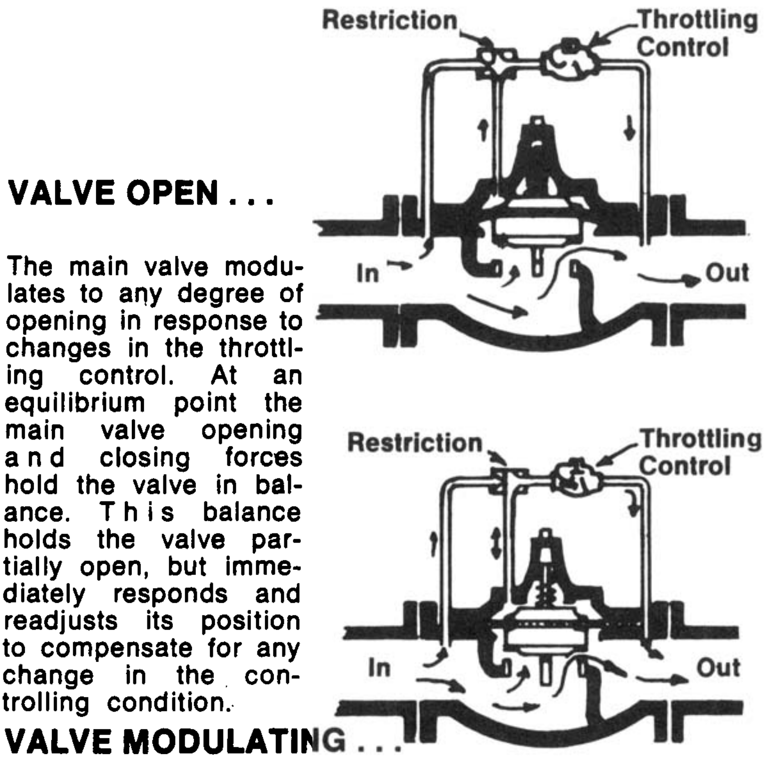


Figure 5-3.

maintenance costs can vary considerably with different systems.

In extremely tall buildings, zoning of domestic water usually is advantageous. Here, a portion of the building's boosted water can be expressed to a specific zone to be served. Or, a lower pressure booster system can supply a second system located at some higher elevation.

Pressure regulating valves can then be installed at each floor, or group of floors, as required. Regulators at each floor offer the best in regulations, and subject only one floor to the possibility of inconvenience if maintenance or repair requires shutdown of the water supply.

5.7 Flow Sequencing

Several types of flow sequencing devices are utilized in water pressure booster systems. These include pressure transducers, pressure switches, current sensing relays, flow switches, time clocks and component combinations.

As long as system pressure is being maintained by the pump(s) currently in operation, there is no reason to bring an additional pump on line. Only when operational equipment has reached its safe flow capacity should additional pumps be programmed on.

A pressure transducer is normally used with a variable speed system. It monitors system pressure and varies the pump speed in response to changes in flow or demand. Normally it increases the speed of the lead pump to its limit. When pressure falls below system requirements, it locks the lead pump on line and brings on the next pump.

A disadvantage of this method is that the system demand will have already exceeded the lead pump capacity and allowed system pressure to drop slightly before another pump is brought on line.

If demand continues to increase, a further drop in system pressure will occur as the second pump comes up to speed. Add to this the inherent momentum of a rotating assembly, and it can be seen that variable speed pressure transducers result in variable system pressures due to hunting on the pump impeller curve.

If suction pressure increases, the flow may reach a point that causes the pump to cavitate or overload the motor. The pressure variations during this type of sequencing are less severe with flat head-capacity curve pumps and with gradual changes in flow requirements. Smaller capacity units normally produce the greatest pressure swings when using transducer sequencing.

Sequencing employing pressure switches to maintain control must also permit variations in system pressure. This is because each pressure switch must maintain a differential to prevent short cycling. In addition, minimum run timers must be used. Pumps run, sometimes, even when not required by the system.

Another approach to pressure switch sequencing is to bring the additional pump on with a drop in pressure and shut it off with a flow switch. This is a complicated approach made more difficult by varying suction pressures.

5.8 Sensing Relays

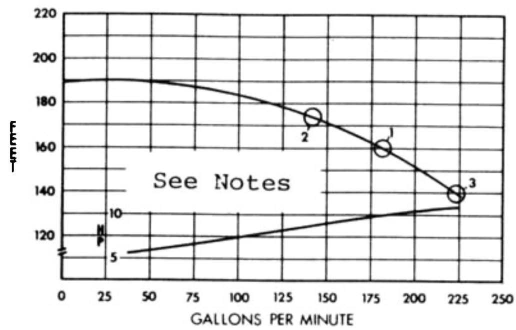
Current sensing relays monitor current draw to the motor in service and bring another pump on line when draw has reached a pre-determined maximum. This device includes a coil that pulls in a relay when current reaches a preset point. After the relay is energized, a time delay relay must keep the added pump on line to avoid short cycling. The device also can have a dash-pot to prevent current surges from energizing the relay.

Since all motors have slightly different characteristics, and incoming voltages vary, current sensing devices are quite sensitive and should be finely tuned under actual load conditions for good operation. Any variation in incoming voltage can have a substained effect on motor amperage draw. This can result in pumps running unnecessarily, or losses in system pressure. A 10 percent variation in incoming voltage results in a 25 percent change in the pump sequencing point (Figure 5-4).

A dependable flow switch is the best method for sequencing pumps. Since this device directly monitors pump flow in the pipe, it is a simple mechanism to bring another pump on line when the flow reaches the limits of the pump(s) currently running. A switch should have a differential or time delay relay to prevent unwanted pump cycling. Flow switches can be the paddle-type, or the primary element and visual readout type. The latter is usually a rotometer and orifice plate arrangement. The orifice plate creates a differential pressure that by-passes water to a tapered tube containing a floating indicator. As the indicator reaches a predetermined point, additional pumps are brought on line to handle the flow. The orifice plate and rotometer arrangement require time clocks to prevent short-cyling.

5.9 Low Flow

Excessive amounts of energy are wasted by domestic booster systems if pumps are permitted to run when the flow is small or non-existent. A building with periods of low flow should prompt the designer to consider devices that allow the pump system to shut down completely. Such devices include hydro-pneumatic tanks, HydroCumulator tanks and low flow controls. Since no plumbing system is without some leakage, any device that stops all pumping equipment without a storage provision for water under pressure should not be considered.



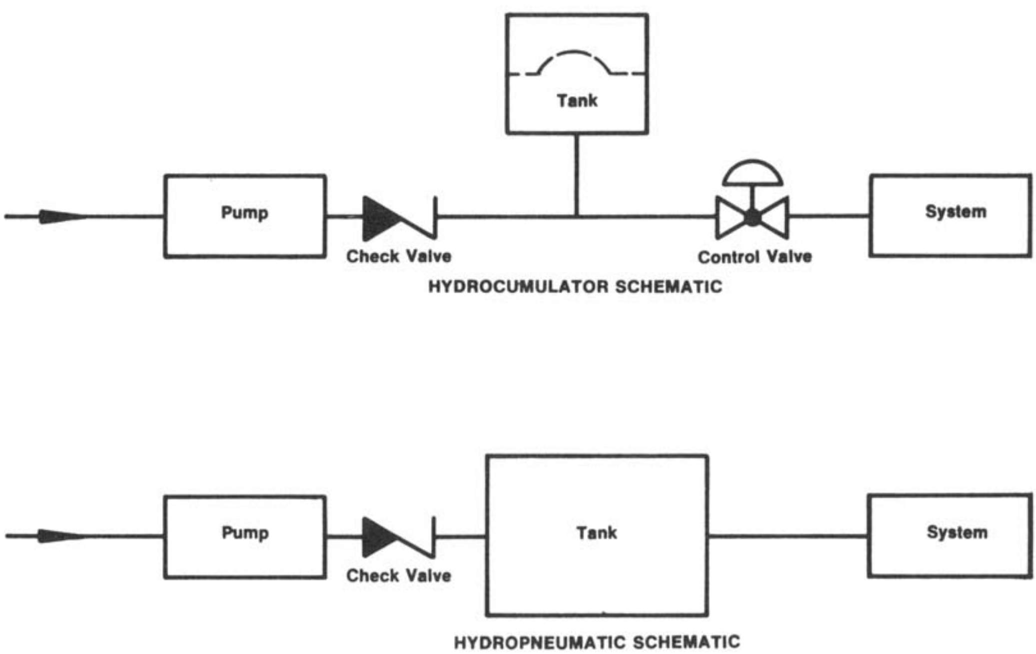
Notes: 1-Switch Point at 230 Volts
2-Switch Point at 207 Volts
3-Switch Point at 253 Volts

Figure 5-4.

A hydro-pneumatic tank is used as a jockey on the system, and a HydroCumulator as a jockey on the inlet to the control devices. Figure 5-5 illustrates the piping differences. The hydro-pneumatic tank, when jockeyed directly on a system, requires system pressures to vary somewhat in order to store water in the tank. A 1,000 gallon hydro-pneumatic tank installed on a 125 psig system, and a control sequence allowing system pressure to vary only 5 psig, could store a maximum of 30 gallons. However, to store more water, a HydroCumulator tank is installed between pump discharge and pressure regulating valves to permit an increase in pump pressure as it approaches the no-flow point. The increase in pump pressure results in a significant increase in the amount of water capable of being stored in the smaller HydroCumulator vessel.

Figure 5-5

Piping Schematics — Shut Down Systems



5.10 Motor Sizing

A typical centrifugal impeller requires greater current draw as the impeller passed its best efficiency point to the right of the curve. The motor should, therefore, be sized for the maximum draw of the impeller noted on the curve. Of course, non-overloading type impellers may be selected so that the horsepower draw is maximum prior to reaching the maximum flow and minimum head on the pump curve. Vertical turbine type pumps are typical of the non-overloading design in which their horsepower normally at a peak near the best efficiency point. Open drip-proof motors are entirely adequate for most water booster applications. Even outdoor installations do not normally require totally enclosed motors. It should be remembered that totally enclosed motors do not have the 15 percent service factor found in open units.

The designer may choose vertical hollow shaft motors or vertical solid shaft motors for vertical multi-stage diffuser pumps. Each type is available with the high thrust bearings. One consideration for using the hollow shaft motor is that the hollow shaft design enables simple adjustment of impellers and easy seal maintenance without the use of expensive spacer type couplings and higher motor stands.

5.11 System Controls

The basic control components, including starters, overload blocks, and control circuit transformers, should be a part of the package furnished by the booster system manufacturer. Other controls that should be considered are as follows:

1. Individual pump disconnect switches
   Through the door disconnects installed in the control panel supplied by the pump manufacturer enable maintenance personnel to completely isolate any one pump, while allowing the system to continue to supply water to the building.
2. Low suction pressure shutdown
   This is usually a pressure switch, or a float switch in the case of break tank installations. As the name suggests, it prevents the pump from running and destroying itself should there be insufficient water in the suction manifold. A low suction pressure shutdown is usually required by plumbing codes.
3. Low system pressure switch
   This device monitors system pressure, and should low pressure be detected, would bring another pump on the line by by-passing the normal pump sequencing controls. A low system pressure switch should require a manual reset to alert maintenance personnel of the problem.
4. High system pressure switch
   This device also monitors system pressure. Should the pressure rise beyond a predetermined point, it would shut down all pumping equipment. When system pressure falls, the low system pressure switch should bring on the lag pump to maintain system pressure.
5. Control power switch
   This is a simple on-off or key operated on-off power switch, usually furnished with a light to indicate to operating personnel that the system is powered.
6. Pump running lights
   These are convenient for maintenance personnel and indicate that the pump is receiving power.
7. Alarm
   Any alarm condition should be signaled to operating personnel via an audio-visual system. An excellent method of signaling alarm is to sound the audible alarm for any

condition, and energize a light to indicate the specific condition. An alarm silencing button should be used to silence the horn and a separate button to reset the device. Some alarm conditions should not be automatically reset since they could create problems in the plumbing system.

8. High suction pressure by-pass
   This device should be used when suction pressure rises to a pressure high enough to accommodate the building without further boosting. A pressure switch senses suction pressure. Should pressure rise and remain high for a predetermined time, the complete system would then be shut down and water bypassed directly to the discharge manifold, under control through the pressure regulating devices.
9. Other controls
   Running time totalizers, strip chart recorders, voltage and amperage meters, are all devices that can be used in water pressure boosting systems. They can give the owner good input information to determine the true conditions in his system.

5.12 System Head Curves and Areas

Evaluation of pumping requirements for plumbing systems usually consists of determining maximum pump capacity and head. Often pumps are selected without consideration of operation at minimum or part-load conditions.

Proper selection of pumping equipment requires development of a system head curve, which is defined as the pump head required by a water system (of any type or use) from minimum to maximum flow.

The system head curve usually is parabolic in form, increasing from minimum to maximum flow and head. The system head curve consists of independent and friction heads. Independent head is the actual increase in elevation, the true static head difference and, on closed systems, includes the difference between residual pressure at the end of the system and supply pressure. Friction head includes losses in piping, pipe fittings, valves, meters and back flow preventers.

There are five distinctly different pumping systems used in public buildings. These five systems are shown in Figures 5-6, 5-7, 5-8, 5-9 and 5-10. The Type 1 system consists of lifting water out of a sump, or receiver, to a sewer at a higher elevation. (See Figure 5-6.)

Typical applications are sewage ejectors and storm or underground water pumps that lift water from a basement into a storm sewer. The system's total head consists of the static head (the difference between the water level in the sump and the level of the connection to the sewer), plus the piping friction from the sump to the sewer.

The Type 2 system shown in Figure 5-7 is similar to the Type 1 system in that water is lifted from a reservoir or tank into another tank. In this case, water is delivered to fixtures between the pump and the upper tank, making this system typical of potable water in a building with a suction tank or reservoir and a roof tank. It is also typical of an industrial plant where water is pumped from a ground storage tank to an elevated tank. The system head curve consists of the static elevation between the suction tank and the elevated tank, plus piping friction.

The Type 3 system shown in Figure 5-8 is similar in that water is pumped from a suction tank or reservoir into a potable water system. However, there is not a roof tank and a minimum pressure is maintained at the top of the building or at the end of the potable water system. This system is typical of a potable water system in a public building with a suction tank, or an industrial water plant, taking water from a ground storage tank and and pumping it throughout a building complex. The system head curve consists of static elevation between the top building fixture and the suction tank, plus the delivery pressure at the top of the building and piping friction.

The Type 4 system in Figure 5-9 represents a potable water system for a high rise building receiving water from a city water main. A roof tank is provided for storage or fire fighting purposes. The system head curve consists of independent head (static head minus city water main pressure) and friction head, including head losses in the water meter, backflow preventer and piping.

The Type 5 system shown in Figure 5-10 consists of potable water being pumped from a city water main through a water meter and backflow preventer into a building containing no major storage tanks. A minimum pressure is maintained at the top of the building or at the end of the main. This system is typical of potable water systems in public buildings or industrial plants that receive water from city mains. The system head curve consists of the static head difference between the city water main and the top of the building, or the end of the building water system plus the difference between the pressure required at the top of the building and the minimum water main pressure and the friction in the water meter, backflow preventer and piping.

It is obvious from the five examples of public building plumbing systems that the system head curve is not the same for all installations. For each of these systems, the generation and demonstration of the system head curve should be reviewed for different building types.

Figure 5-11 describes a classical system head curve with uniform flow in the water systems. Uniform flow means all fixtures in the system are flowing water at the same percentage of design flow. It is imperative that uniform flow is stressed in the development of the system head curve. It will be demonstrated later that nonuniform flow causes a system head area or band to be generated, not a simple curve. As shown in Figure 5-11, a system head curve consists of an independent head plus a friction head. The independent head is the building's static head or system plus any difference between a minimum supply pressure at the highest fixture and water supply pressure in a city main.

Friction head varies with water flow in the system. Independent head varies separately from the system flow and is subject to other conditions, such as elevation of water in tanks or supply water pressure in the city water mains. Friction head is caused by resistance to flow in piping and equipment. It increases as flow increases and decreases when flow subsides. The Williams-Hazen formula, extensively used for calculating pipe friction, demonstrates that flow varies to the 1.85 power of system flow.

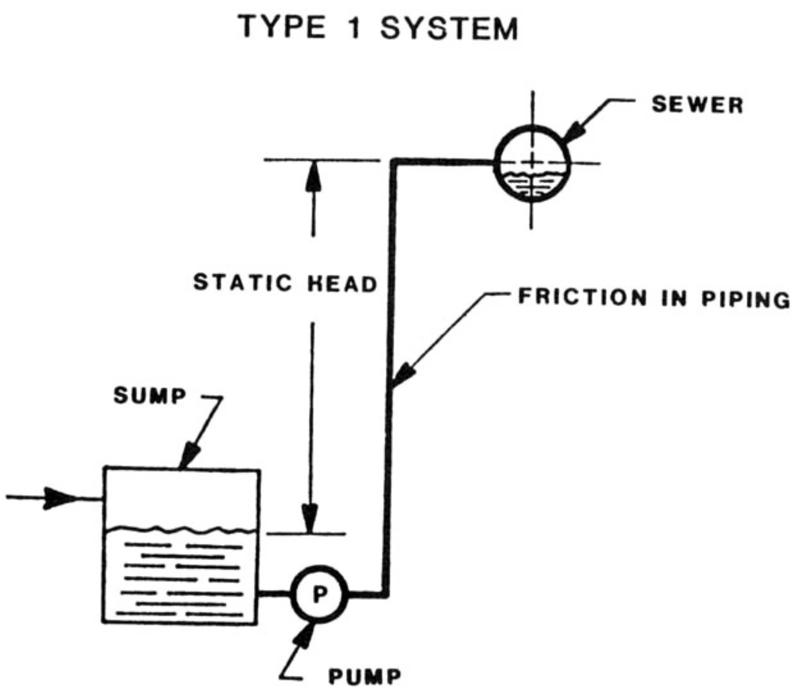
Formula 5-1 illustrates this:

$$H_1 = H_2 \times (Q_1/Q_2)^{1.85}$$

H_1 is the head at any system flow (Q_1) and H_2 is the head at the design flow (Q_2) for the building.

The system head curve shown in Figure 5-11 is useful in generating a means to indicate the approximate head for a pumping system at any given system flow. In actuality, very seldom does uniform flow occur in a water system. Rather, non-uniform flow occurs most of the time and generates a system head band or area as shown in Figure 5-12.

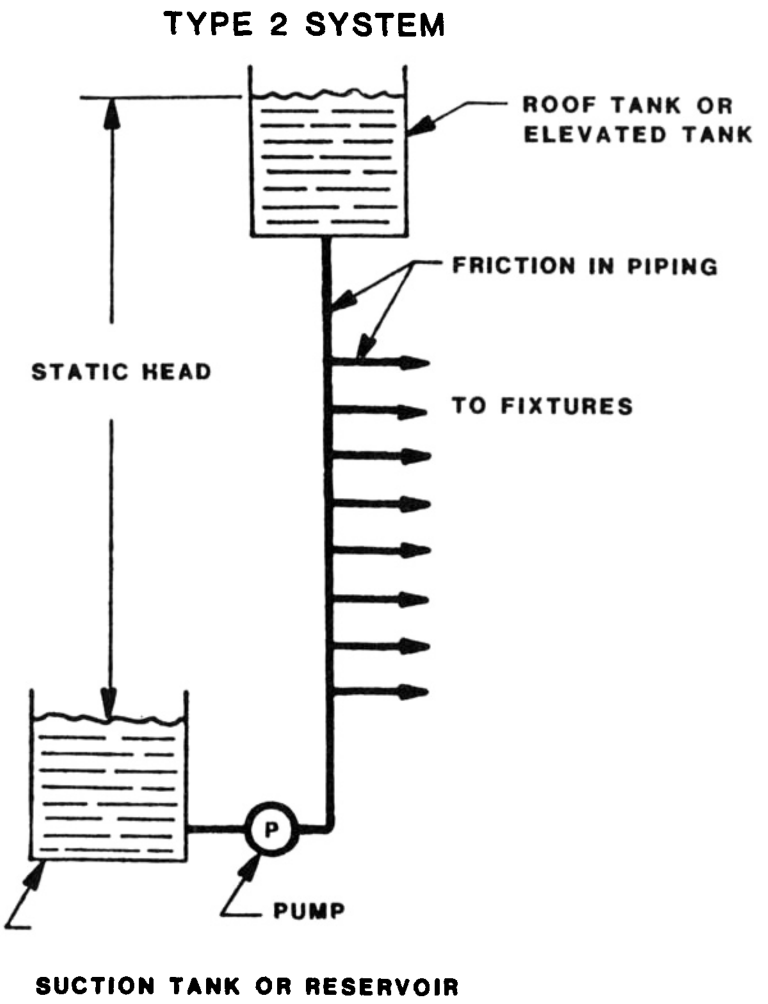
If the loads are active near the pumps, the actual friction head will be less than that shown by the uniform head curve. Likewise, if the loads are active far from the pumps, the friction will be greater than that of the uniform system head curve. It is easy to prove the existence of an area or band (See Figure 5-12) by developing friction in the system as various loads are added or subtracted. The system head area concept was dis-



Typical Applications:

1. Sewage Ejectors
2. Storm or Underground Water Sumps

Figure 5-6. **Type 1 System**

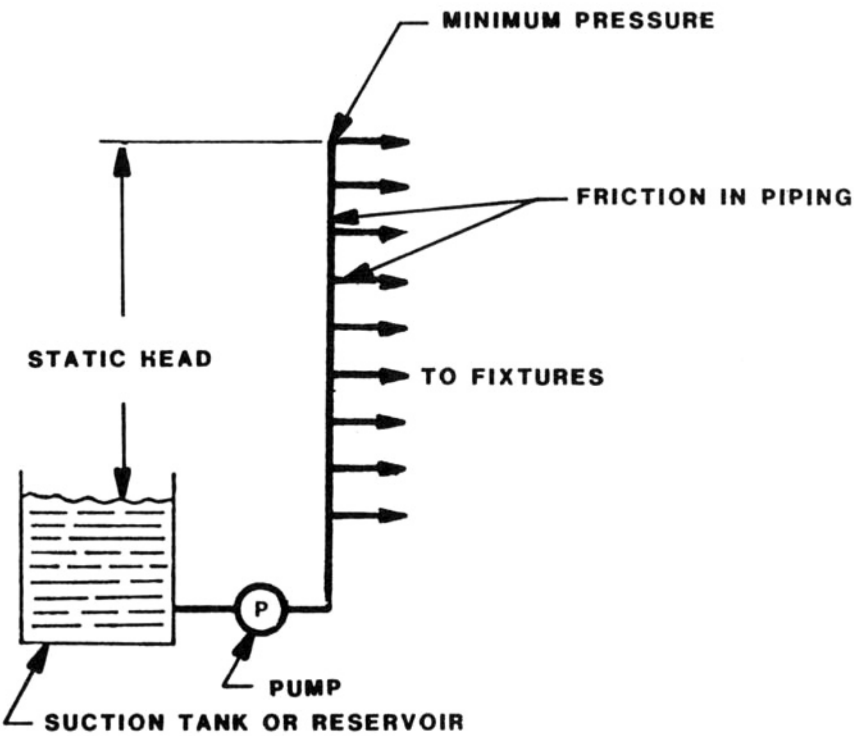


Typical Application:

1. **Potable Water in Building with Suction Tank and Roof Tank**
2. **Industrial Plant with Water Supply Tank and Elevated Tank**

Figure 5-7. Type 2 System

TYPE 3 SYSTEM



Typical Application:

1. **Potable Water in Building with Suction Tank**
2. **Industrial Plant with Water Supply Tank**

Figure 5-8. **Type 3 System**

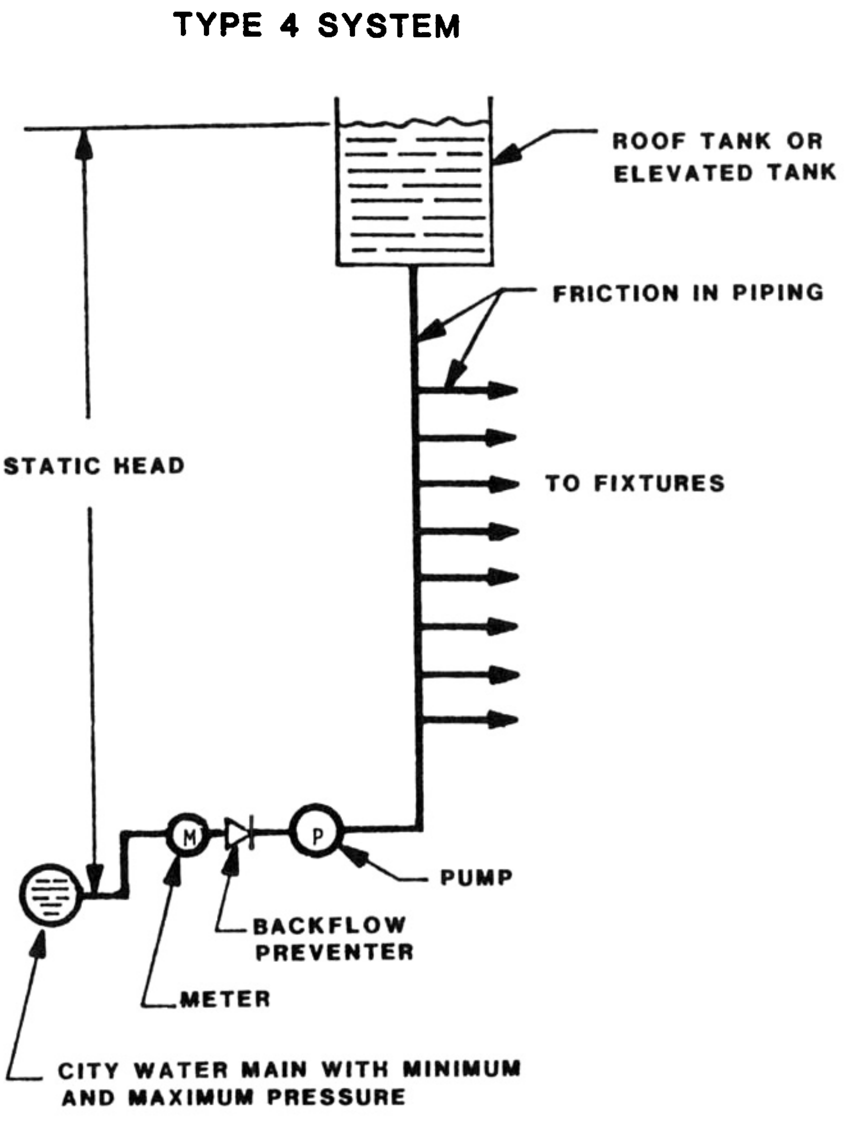
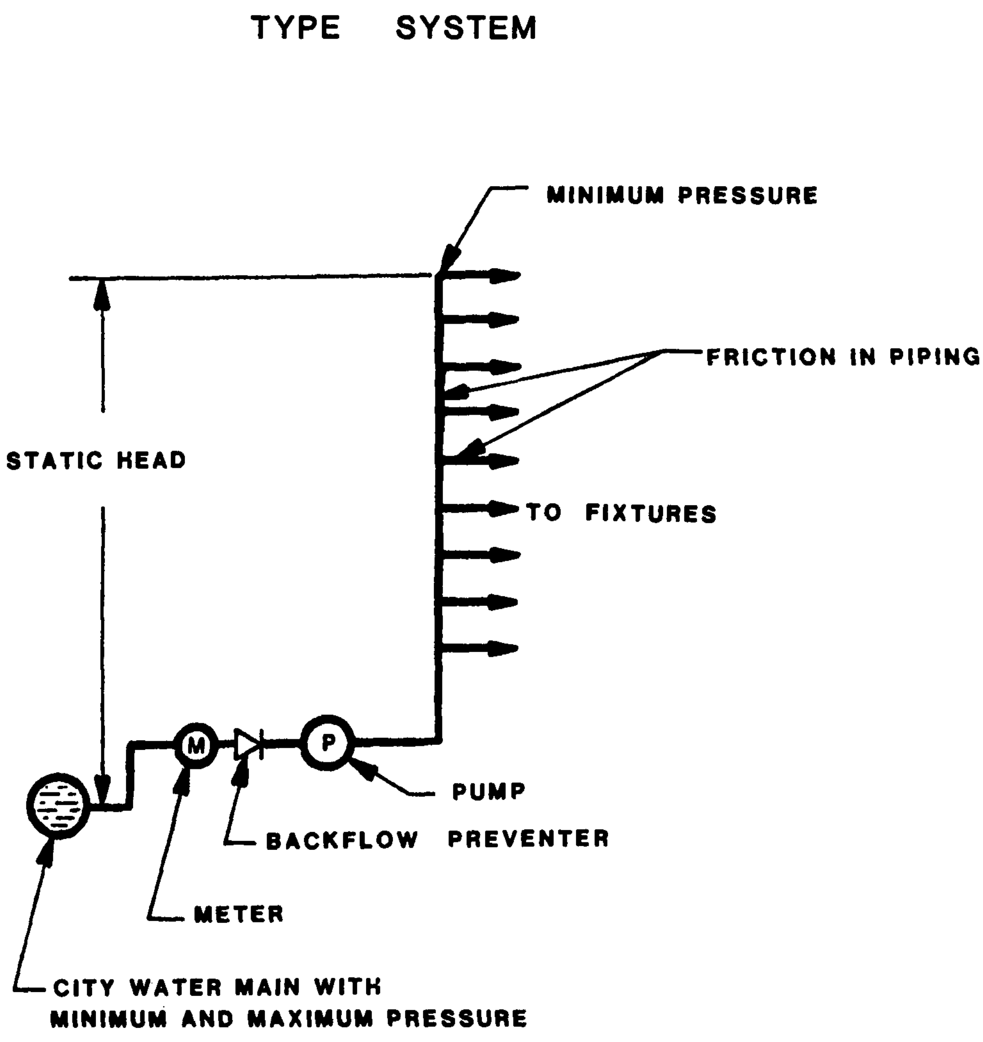


Figure 5-9. Type 4 System

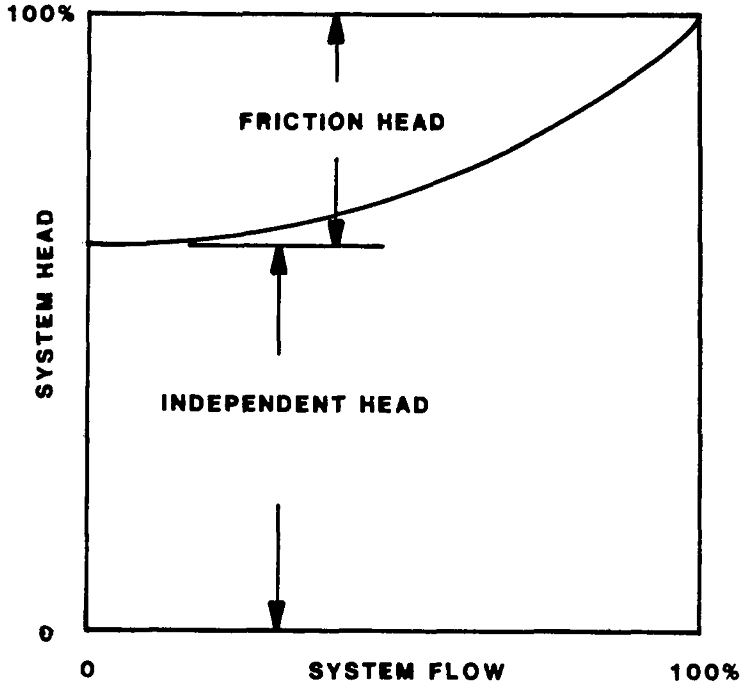


Typical Application:

1. Potable Water in Building Receiving Water from City Water Main
2. Industrial Plant Receiving Water from a City Water Main

Figure 5-10. Type 5 System

covered originally in municipal water systems where a great variation in loads occurs at different elevations, causing a broad band of pressure requirements by the pumps.



Friction Head:

Friction Head Caused by Resistance to Flow in Piping and Equipment; it Increases as Flow Increases:

For Piping : $H_1 = H_2 \left(\frac{Q_1}{Q_2}\right)^{1.85}$ (Williams-Hazen Formula)

Independent Head:

Consists of Heads That Do Not Change with Flow, Such as Static Head and Minimum Pressure at End of System

Figure 5-11. Classical System Head Curve (Uniform Flow in Water System)

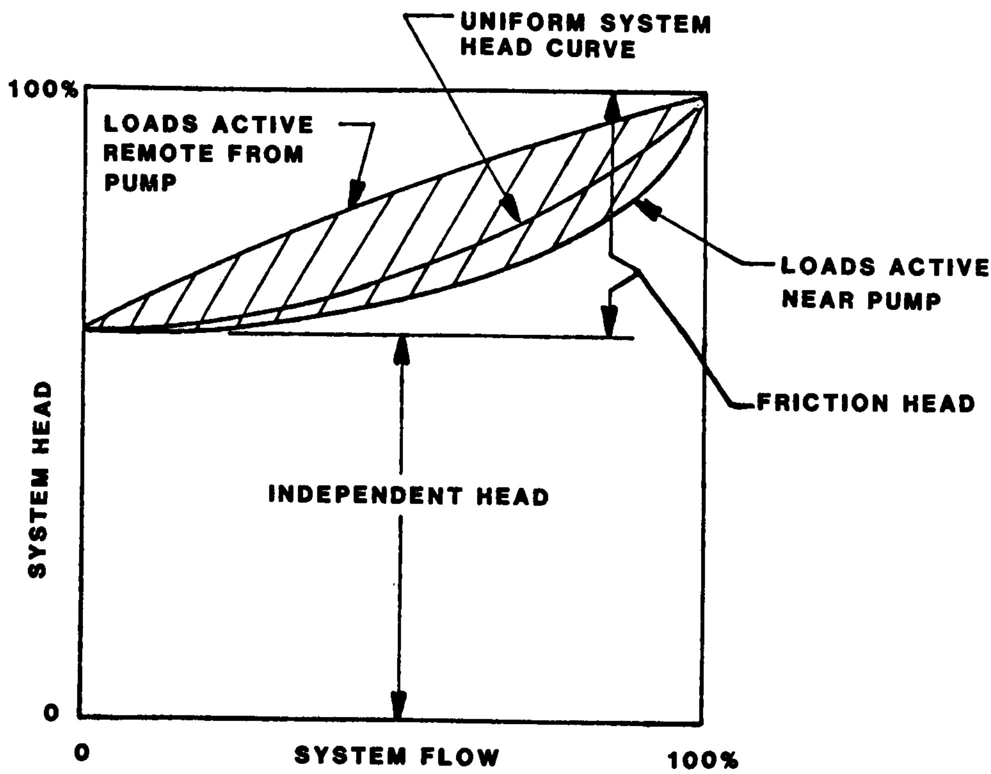
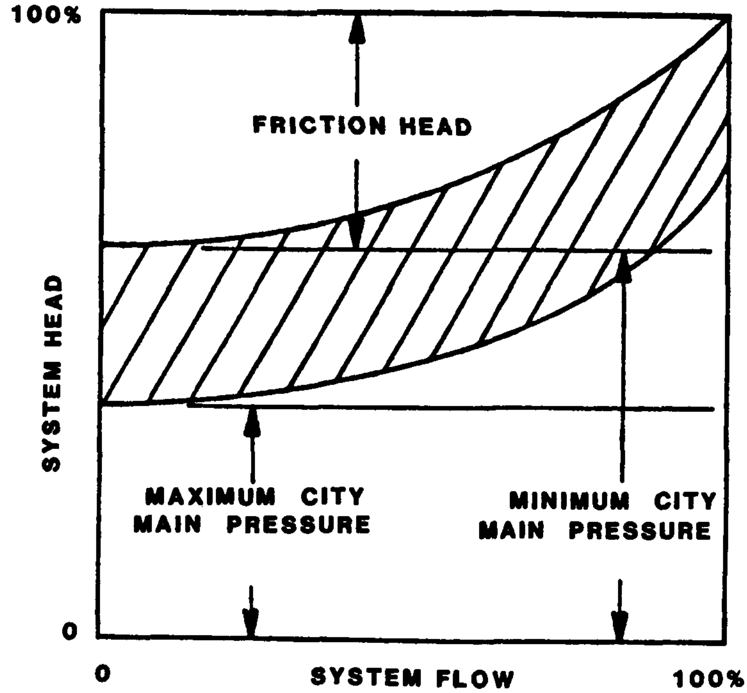


Figure 5-12. **System Head Band Caused by Non-Uniform Flow in Building**

A variation in the system head curve can also be caused by a fluctuating water supply pressure that changes the independent head and, therefore, raises and lowers the uniform system head curve. This system head area or band is shown in Figure 5-13. A variation in the independent head, along with nonuniform flow in a building, can result in a system head area as shown in Figure 5-14.

This description of system head curves and areas will be applied to typical buildings after a review is made of pumps and their head-capacity curves. Generally it is believed that steep curved pumps are superior to flat curve pumps on potable water systems for public buildings, particularly on high rise installations. In actuality, the steep curve pump is an inefficient pump when compared to a flat curve pump. Figure 5-15 describes this inefficiency. The shaded area in this figure represents the energy wasted or lost by using a steep instead of a flat curved pump. Rather than assuming that a steep curved pump always is needed, pumps with a rising characteristic to shutoff or no-flow condition should be used, not one with a looping or falling characteristic whereby maximum pump pressure exists at some point other than at the no-flow or shutoff condition.

In the past, a steep pump curve has been selected to provide pressure for pumping hydropneumatic or diaphragm tanks in building basements. This is a terribly inefficient method of providing storage water for a public building. The kw per 1000 gallons of water pumped is extremely high, and can be as much as



Independent Head = Static Head + Minimum
Pressure - City Main Pressure

Figure 5-13. System Head Band Caused by Variation in Water Supply Pressure

twice that for normal pumping conditions in the building.

This leads to another misconception that the hydropneumatic or diaphragm tank should be located in the basement for providing low-flow conditions that occur at night. Again, the hydropneumatic tank should not be located in the basement, but should be installed at the highest point in the system to provide the most efficient means of water storage for low-flow conditions. The tank located on top of the building can provide twice as much storage at 25 to 50 percent of the cost of the basement tank. No advantage is achieved by locating the tank in the basement. Likewise, the tank should be sized to serve low-flow conditions in the building, rather than on some nebulous leakage rate that may exist in the building at night. Actual tank size is contingent on the characteristics of the system itself. For example, the tank size for a hospital building will differ than that for a public office building. Another advantage of installing the tank at the top of the building is that the tank acts as an excellent shock absorber for removing shock loads caused by solenoid valves on washers or on make-up lines to cooling towers.

The pump type selected for a plumbing system is contingent on the head and capacity required, as well as overall efficiency and ease of maintenance. Recently, there has been great emphasis on the use of vertical turbine type pumps on all public buildings.

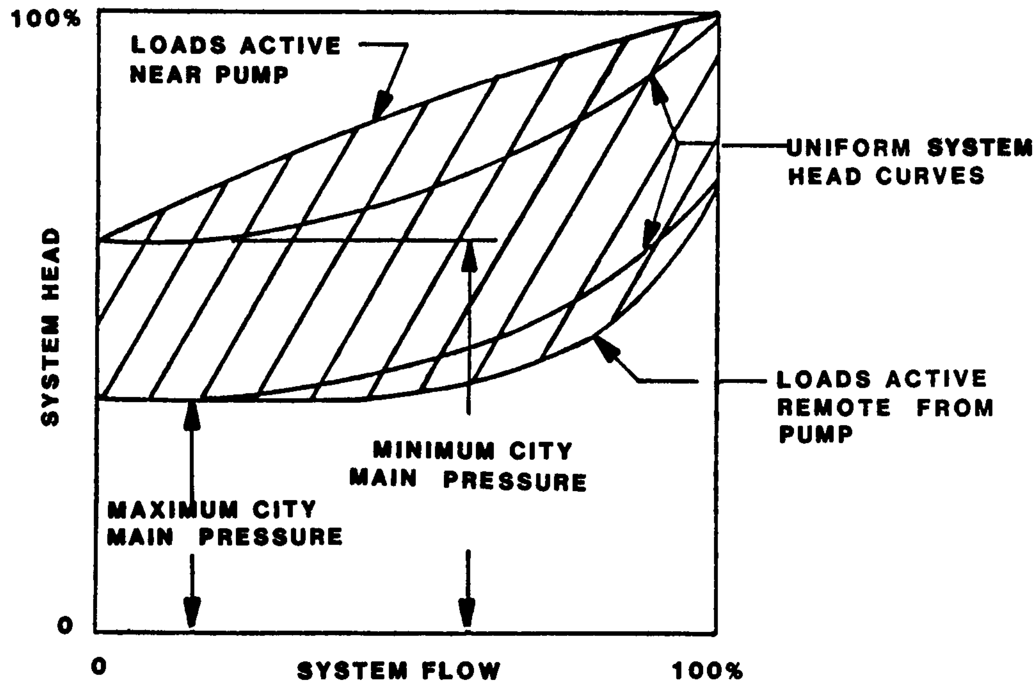


Figure 5-14 . **System Head Band Caused by Variation in Water Supply Pressure and Non-Uniform Flow in Building**

This is a misapplication of that particular pump. In many cases a horizontal split case or end suction pump will be much more efficient and provide a lower level of maintenance and technical capability for repairs.

The evaluation of the following five pumping system types will demonstrate there is no single pump-type that is best for all potable water systems.

Another misconception is that all pumps should be constant speed or all pumps should be variable speed. It will be shown that the use of variable speed is dependent on the system head curve or area and type of building. It is not a panacea for all buildings. The following discussion demonstrates the pump needs for various types of buildings.

Figure 5-16 describes a typical Type 1 system, which could be a sump pump or sewage ejector. In this case, a sump is located about 15 feet below the connection of the pump discharge into the sewer. The piping friction is five feet, so the total dynamic head required of the pumps is 20 feet. As there is no intermediate drawoff in the piping between the pump and the sewer, it is not possible for a system head band or area to exist.

Figure 5-17 describes a system head curve for this system. If the sump level did vary greatly, it would be necessary to draw another system head curve below the one shown to demonstrate the variation in level in the sump. In actuality, the sump level

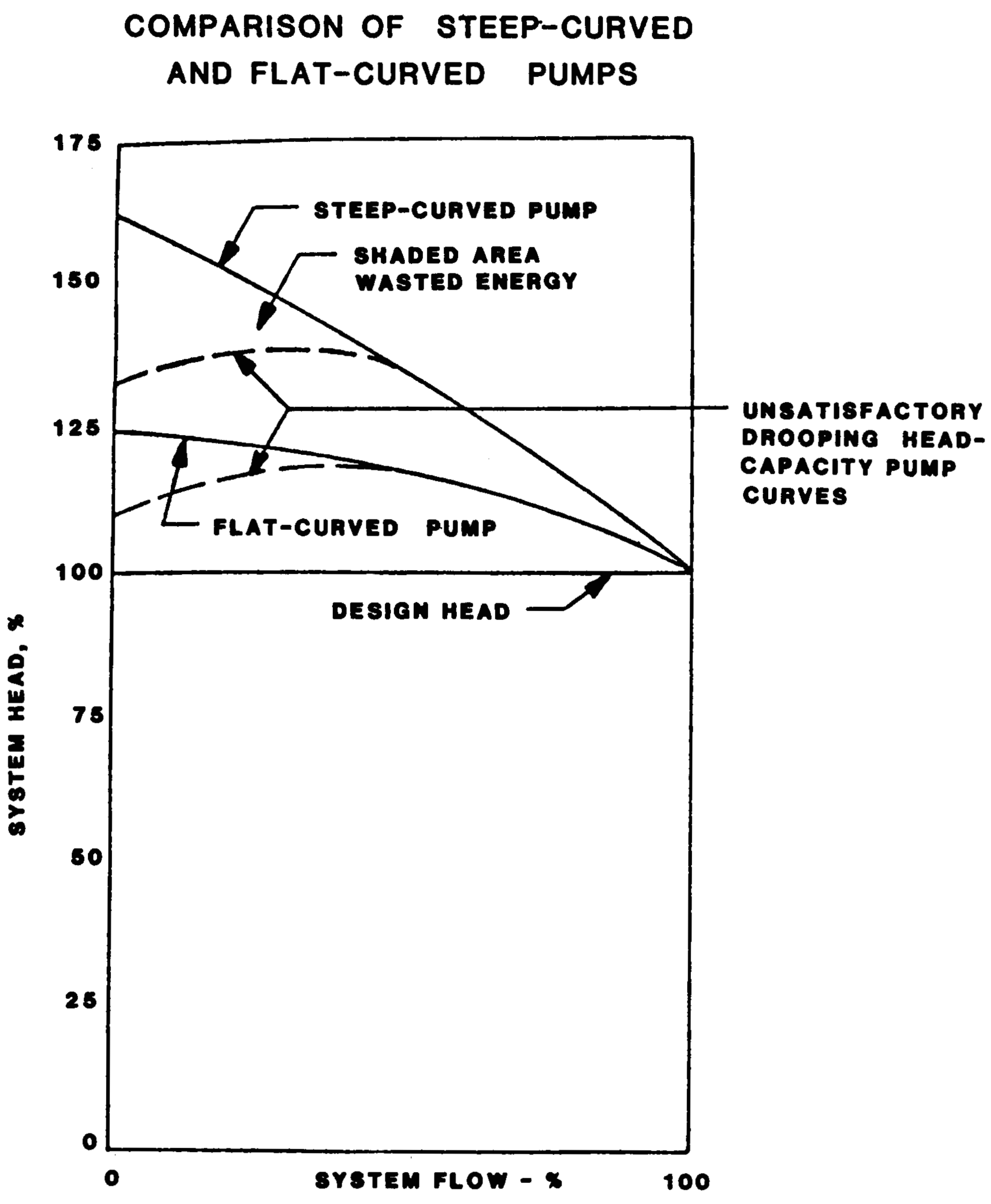


Figure 5-15. Comparison of Steep-Curved and Flat-Curved Pumps

seldom varies over a foot or two, so it is not necessary to draw a system head area for this particular system.

As this is a free flowing system, pumps will operate at the intersection of the sytem head curve and the pump capacity curve. If, as shown in Figure 5-17, two pumps each with a capacity of 50% of the system are selected for this system, when one pump is operating, it will not operate at 50% of system flow, rather it will operate at approximately 70% of system flow. Two pumps operating together should, obviously, operate at the design condition of 100 percent flow at 20 feet of head.

Since the system head for this type of system is easy to calculate, the selection of pumps is also not difficult. Pumps for such systems are usually centrifugal, either of the vertical centrifugal, submersible, or self priming type. Pump efficiency should be evaluated with consideration for ease of maintenance.

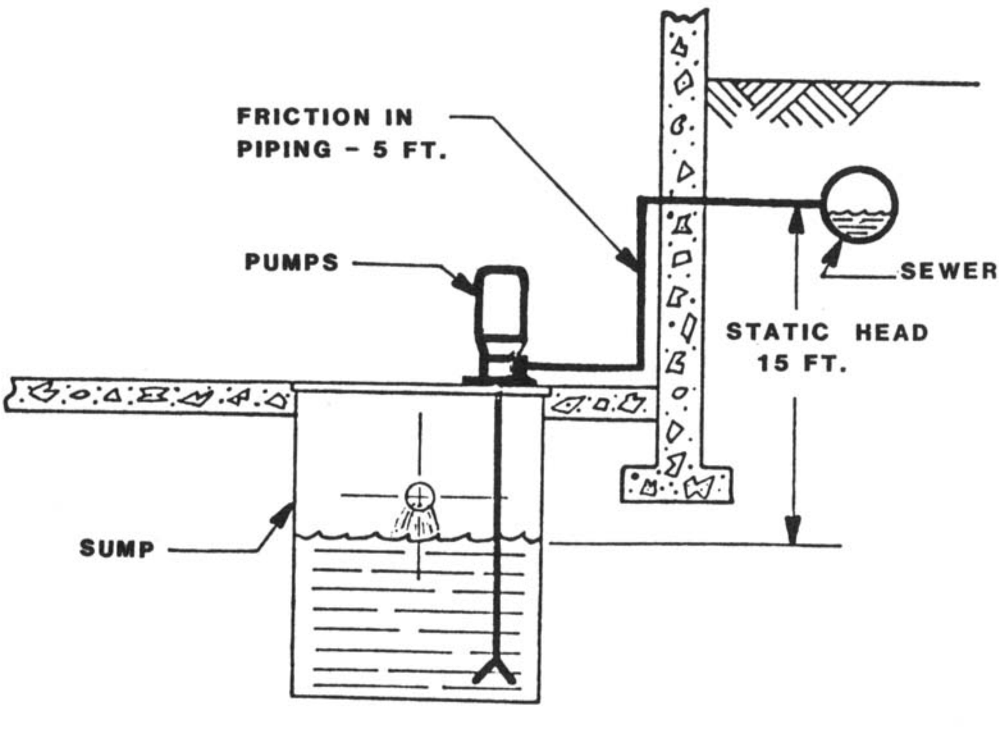


Figure 5-16. **Type 1 System Sump Pump or Sewage Ejector**

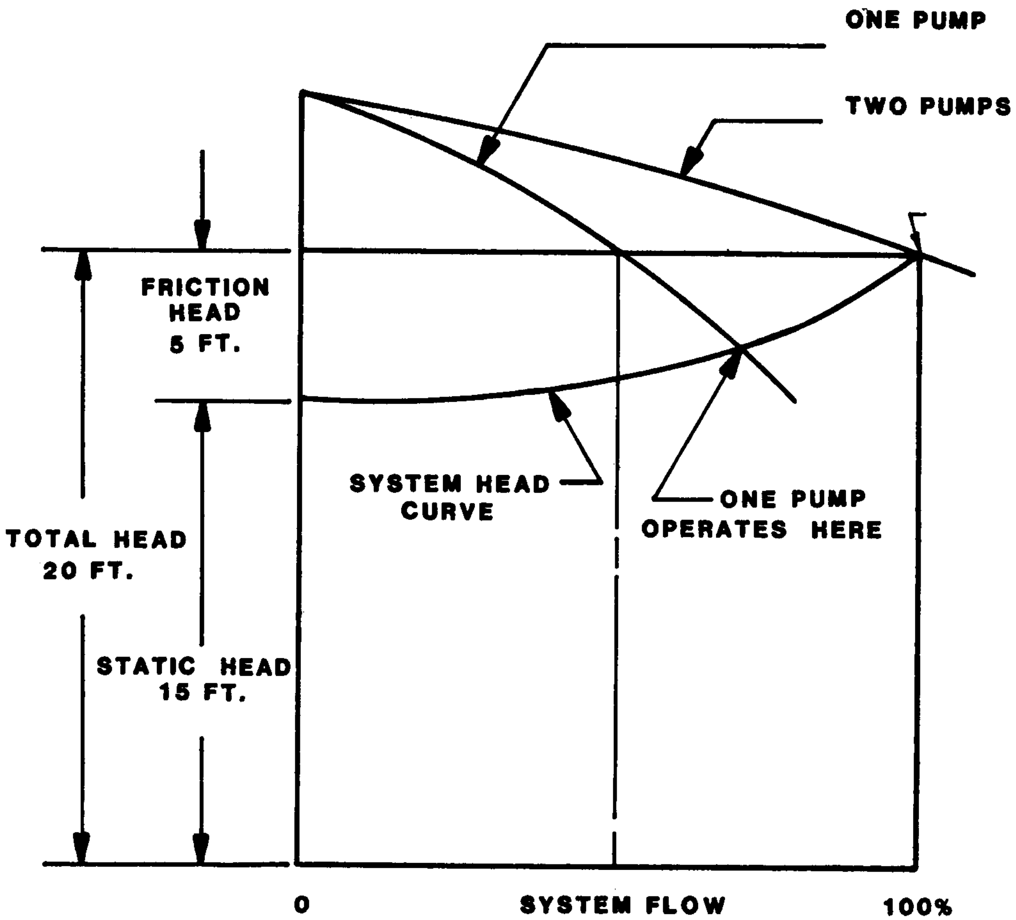


Figure 5-17. System Head Curve for Type 1 System

Figure 5-18 describes a Type 2 system, namely a potable water system in a high-rise public building. As shown, static rise from the suction tank level to the top fixture of the building is 200 feet, with an additional 46 feet up to the roof tank level. This provides a minimum of 20 pounds pressure at the top fixture. The piping friction is only ten feet and, since water meters and backflow preventers do not exist in such a system, a very flat system head curve results. The system head curve is shown in Figure 5-19. The independent head of 246 feet consists of 200 feet of static and 46 feet of minimum pressure. If there is a significant change in water level either in the roof tank or the suction tank, a parallel system head curve will result. In actuality, the water seldom varies so that the system curve will be that shown in Figure 5-19. Since there is a very small amount of

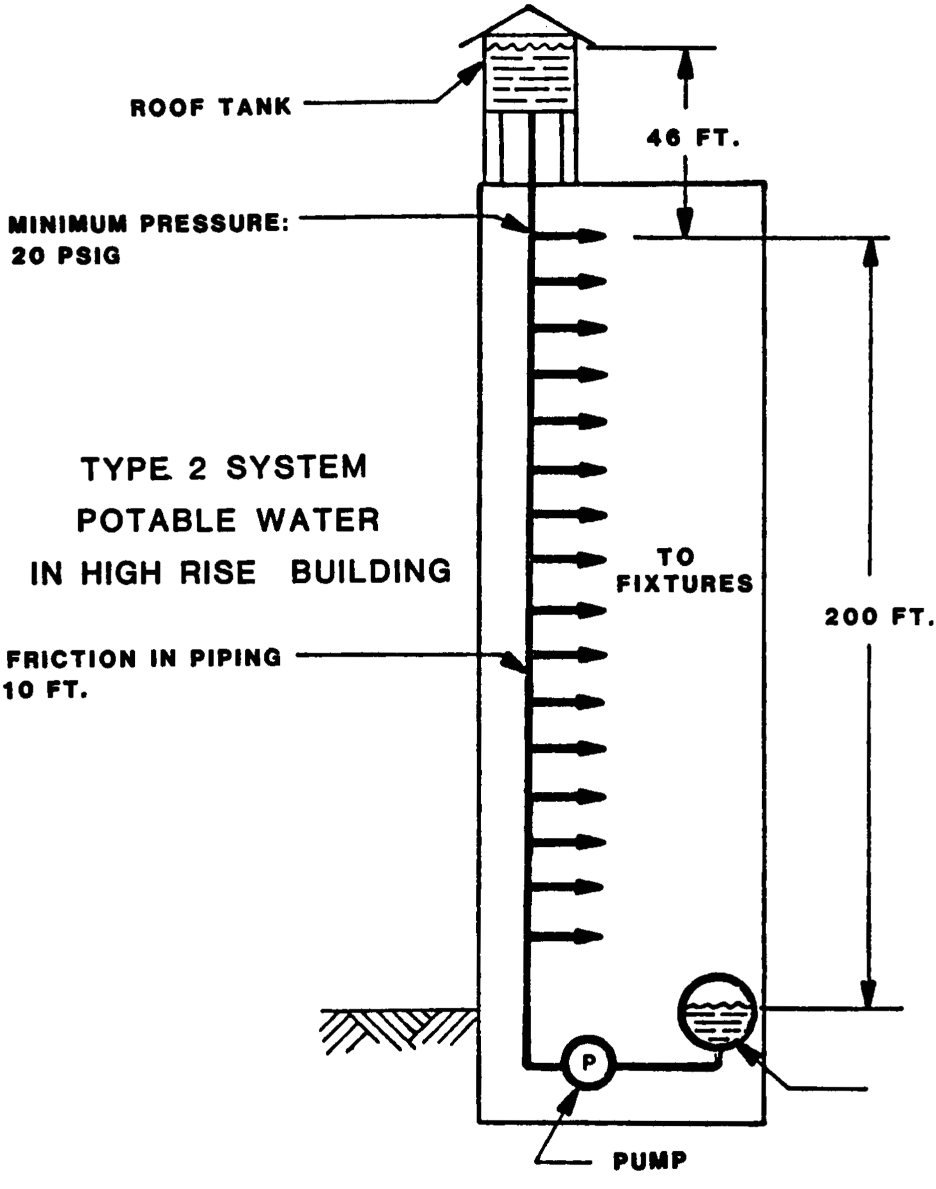


Figure 5-18 . **Type 2 System (Potable Water in High Rise Public Building)** friction, a system head area or band will not exist.

Flat curved pumps without pump control valves are excellent for this type of system. The pumps will not operate at any point but at the intersection of the pump head capacity curve and the system head curve. Therefore, there is no need for pressure regulating valves on the pump discharges; likewise variable speed is of no value on this system. Rather, the emphasis should be on maximum pump efficiency at the design condition. The type of pump can be either vertical turbine, horizontal split case or end suction. The important factors are pump efficiency and ease of maintenance.

Figure 5-20 shows a Type 2 system where water is taken from one tank and transferred to another. The system is potable water in an industrial complex where water is supplied from a ground storage tank and delivered to an elevated tank at the far end of the system. The static difference is 135 feet in the elevated tank minus the 20 feet in the ground storage tank. Since this is a distributive system, the friction in the piping can be appreciable and it is assumed in this case to be 50 feet. The system head curve for this industrial complex is shown in Figure 5-21 with a static head of 115 feet and a friction of 50 feet for a total system head of 165 feet at design flow. Because 50 feet of friction exists in the system, there is a possibility a small system head band exists due to non-uniform flow in the buildings. If the loads in the first building are active, the system head will be below the system head curve. Likewise, loads active in the far building could create system heads greater than the uniform system head curve.

In actuality, the pumps will only operate along a narrow band so that most of the system head area can be ignored. Figure 5-21 shows the pump must be capable of running at the far right condition for single pump operation, which will be about 70 percent system flow rather than at 50 percent system flow. Horizontal split case and end suction types of pumps usually offer the best selection for this type of system, particularly on large flows where efficiency is of utmost importance.

Figure 5-22 describes a Type 3 system which, in this case, is potable water in a high rise building where water is taken from a suction tank and delivered to a water system without a large or open storage tank at the top of the building. The diaphragm tank shown at the top of the building is for minimum flows, not for general storage. The system head curve is shown in Figure 5-23 with the independent head being 246 feet (200 feet of static head plus a minimum pressure of 20 pounds at the top of the building or 46 feet). Again, the friction in this system, like the Type 2 system, is very small, only 10 feet for the entire system. Since this is a closed system with no open storage providing free flow for the pumps, pumps will operate on the pump head-capacity curves, not at the intersection of a pump head capacity curve and the system head curve.

This results in over pressuring, as shown by the shaded area on this figure. Flat curved pumps are a necessity here to prevent over-pressuring of the system. If steep curved pumps are used, it may be necessary to limit their pressure by installing pressure regulating valves on each pump discharge. Careful selection of a flat curved pump can be much more efficient by reducing the amount of overpressuring caused at reduced flows.

As with other high rise buildings, pump selection should be contingent on pump efficiency and ease of maintenance. The pump must be selected to serve the system at the specified flow and head. A large system like this should include a small jockey pump to handle low flows in the building, which can cause exorbitant use of electrical energy. The jockey pump can be used to pump the diaphragm tank at the top of the building and provide

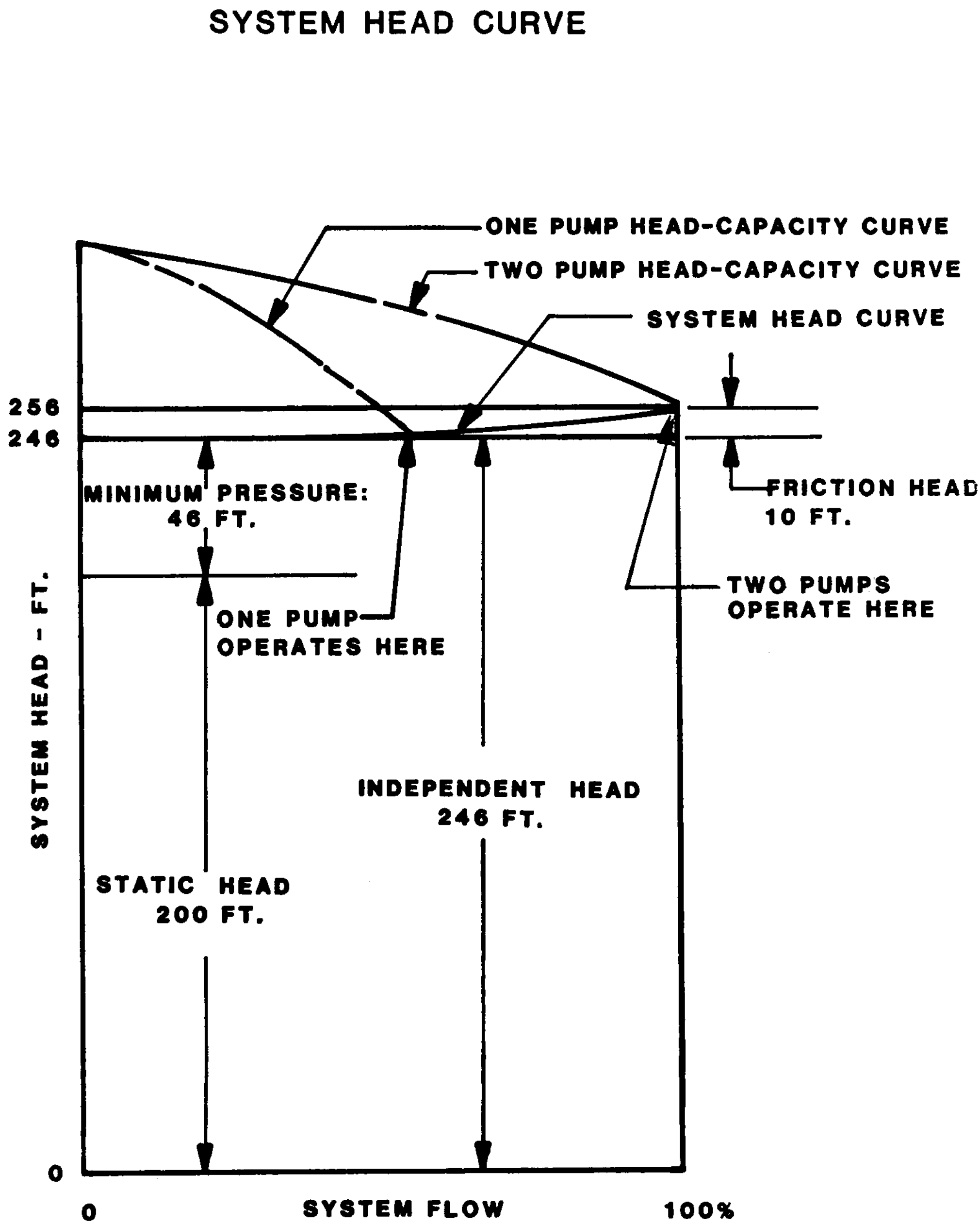


Figure 5-19. System Head Curve for Building Shown in Figure 5-18.

TYPE 2 SYSTEM
WATER SUPPLY IN AN
INDUSTRIAL COMPLEX

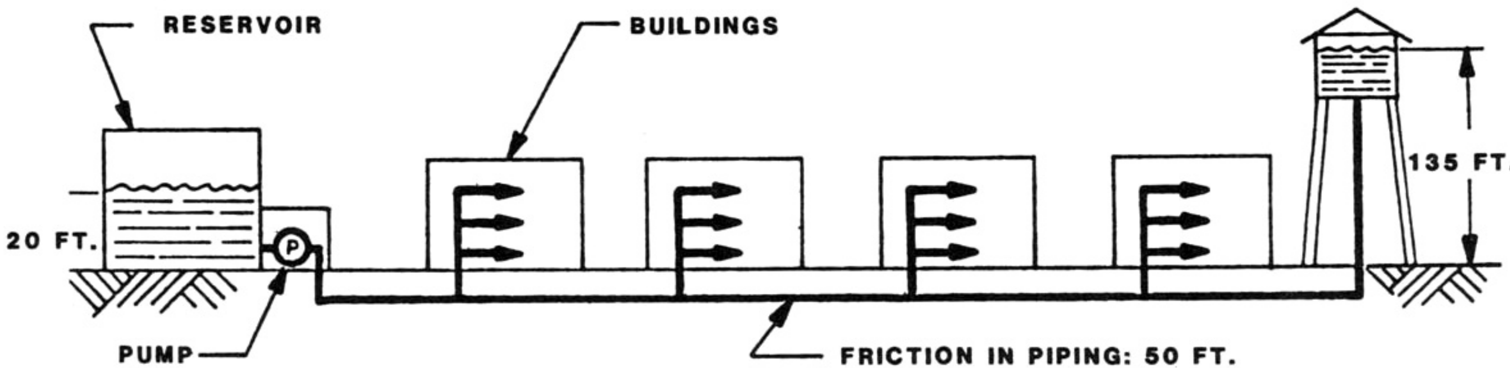
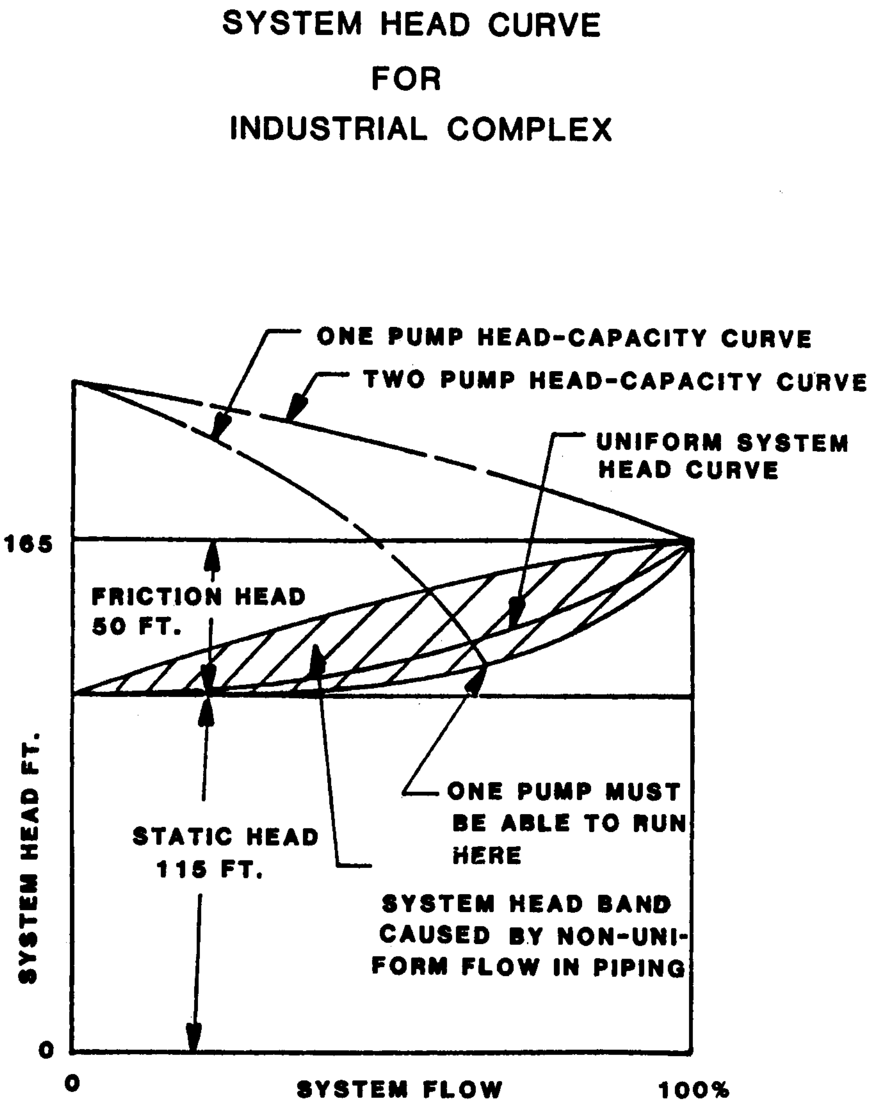


Figure 5-20. Type 2 System (Water Supply in an Industrial Complex)



Net Static Head= Elevated Tank Height (135 ft.)
Less Ground Storage Tank Height
(20 ft.)
=115 ft.

Figure 5-21. System Head Curve for Industrial Complex Shown in Figure 5-20.

efficient operation at very low flow.

The industrial complex as shown in Figure 5-24 is also a Type 3 system where water is taken from a ground storage tank and delivered though the system, providing a minimum pressure of 20 pounds, or 46 feet of head, at the top of the far building.

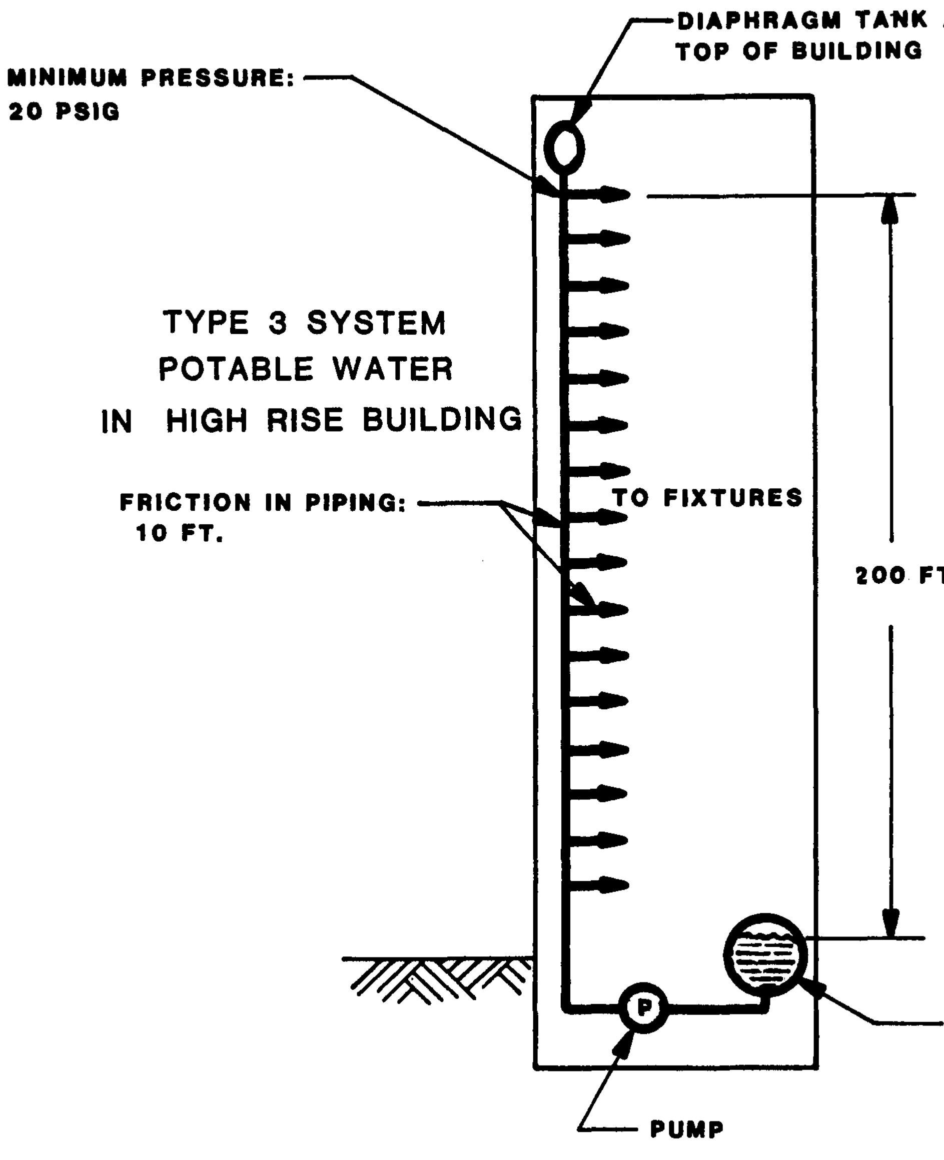


Figure 5-22. Type 3 System (Potable Water in High Rise Building).

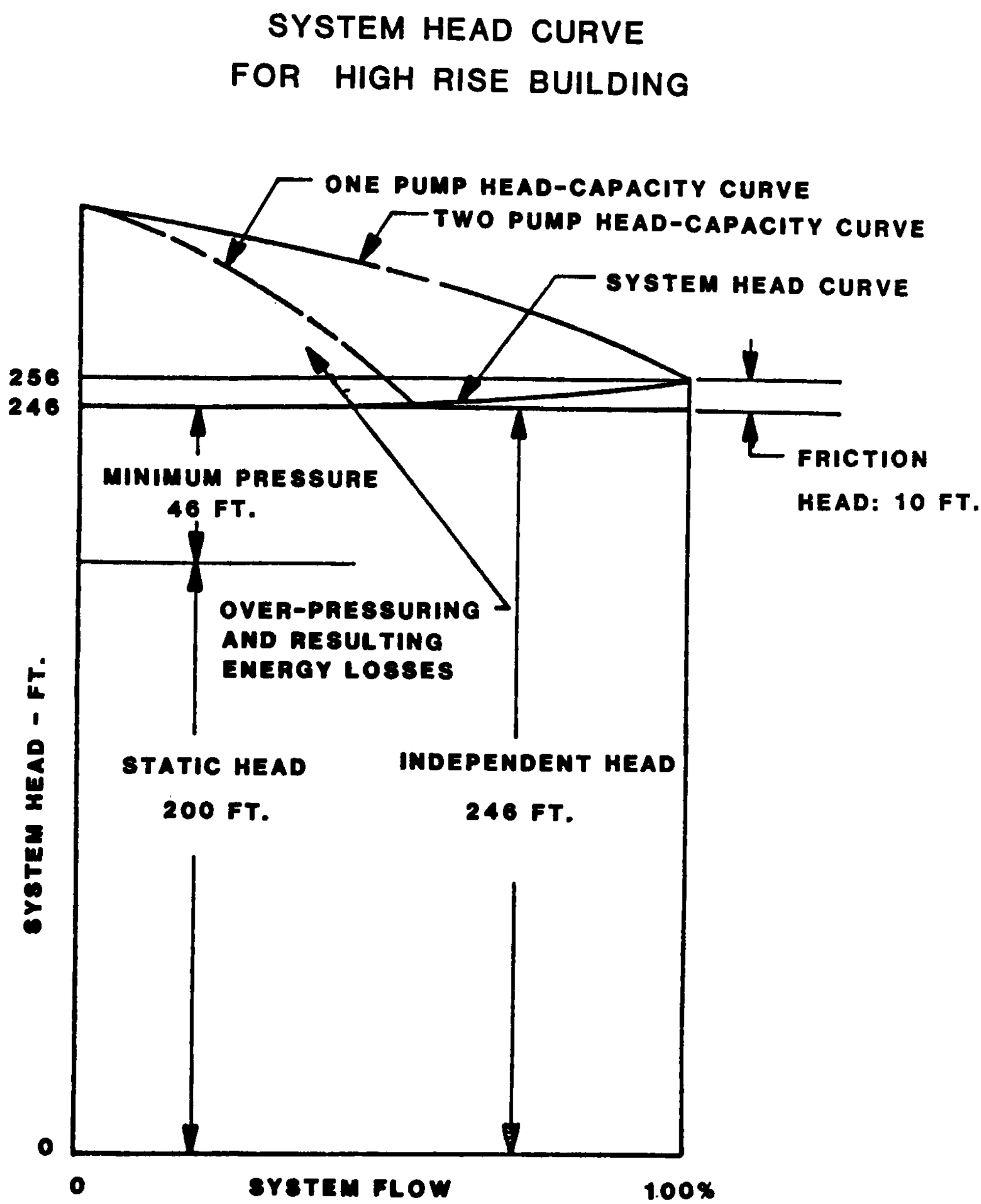


Figure 5-23. **System Head Curve for High Rise Building (shown in Figure 5-22).**

If the top of the building is 20 feet and the storage tank has an elevation of 20 feet above ground level, then no static head exists and the independent head will consist merely of the minimum pressure required at the top of the far building (i.e. 46 feet). The system head curve for this industrial complex is shown in Figure 5-25. Since the friction head is 50 feet, or greater than the independent head, a sizeable rise in head occurs between minimum flow and maximum flow. If the water flow can vary among the four buildings, there is the possibility of a sizeable system head area due to uneven flow in the buildings. The shaded area in Figure 5-25 represents the over pressuring caused by constant speed pumps. In this case, great care must be made in selecting the pumps. If the pumps each with 50 percent capacity are selected, a single pump operation can be as high as 80 percent of the system flow. It is imperative the pump be checked at the far right operating point to insure that cavitation does not occur or, with vertical turbine pumps, that upthrusting does not occur and damage the vertical motor. This system could be an excellent application for variable speed pumps to reduce the overpressuring caused by constant speed pumps. An evaluation for variable speed pumps should be made to determine the time required to amortize their additional costs over the use of pressure regulating valves on the pump discharges.

To secure adequate pressure control, it may be necessary to install a remote sensing pressure transducer to vary the pump speed and prevent undue over-pressuring. This system is an excellent application for horizontal split case or end suction pumps. Normally, vertical turbine pumps are not considered for such an application.

The Type 4 system shown in Figure 5-26 describes a potable water system in a high rise building taking water from a city water main. The building is equipped with an open roof tank for storage or fire fighting purposes. The system head area for this system is shown in Figure 5-27. The system head area will be dependent on the pressure variation in the city water main, in this case from 30 to 50 PSIG. The friction head will consist of the losses in the water meter, backflow preventer and piping, for a total of 33 feet at maximum flow. The type of pump selected for this building could be vertical turbine, horizontal split case or end suction. Emphasis should be placed on pump efficiency and ease of maintenance. The pumps must be able to operate at the far right condition without cavitation, or thrusting in vertical turbine pumps. Variable speed should be considered if there is a sizeable variation in city water pressure so that energy savings can justify the cost of variable speed equipment.

The Type 5 system shown in Figure 5-28 consists of potable water in a high rise building, receiving water from a city water main. The warer main pressure could vary from 30 to 50 PSIG pressure. Static rise is 200 feet of head and minimum pressure at the top of the building is 20 PSIG. A diaphragm tank is shown at the top of the building as was the case of the Type 3 high-rise building. The friction in the piping in only 10 feet, but there is a pressure loss in the water meter and back flow preventer of 5 PSIG each or 10 PSIG total.

The system head curve for this building is shown in Figure 5-29. It is a fairly flat system curve due to the fact that only 33 feet of friction exists. In actuality, system head curves in this case will not be as is shown because the pressure loss curve for water meters and back flow preventers are not necessarily proportional to flow through them. As shown, a system head band has been generated by the variation in the suction pressure to the building. At maximum, the independent head will be 177 feet, with 30 PSIG in the water main. At minimum, it will be 131 feet, with 50 PSIG in the water main. Since this is a

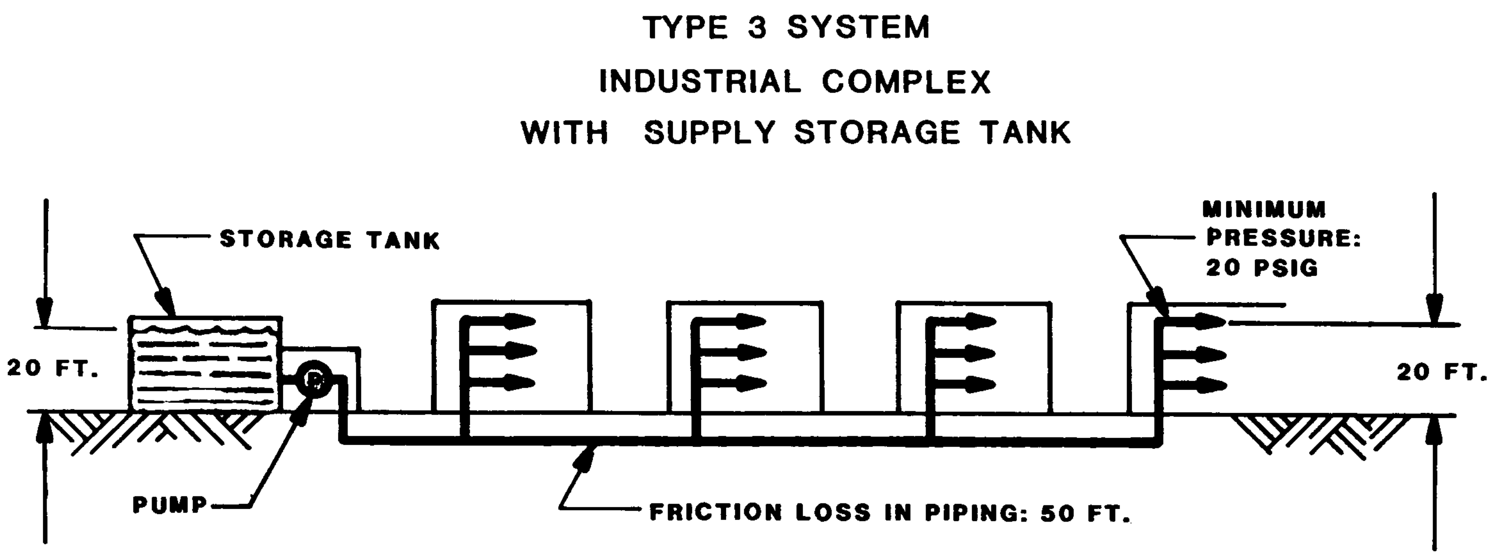


Figure 5-24. Type 3 System (Industrial Complex with Supply Storage Tank)

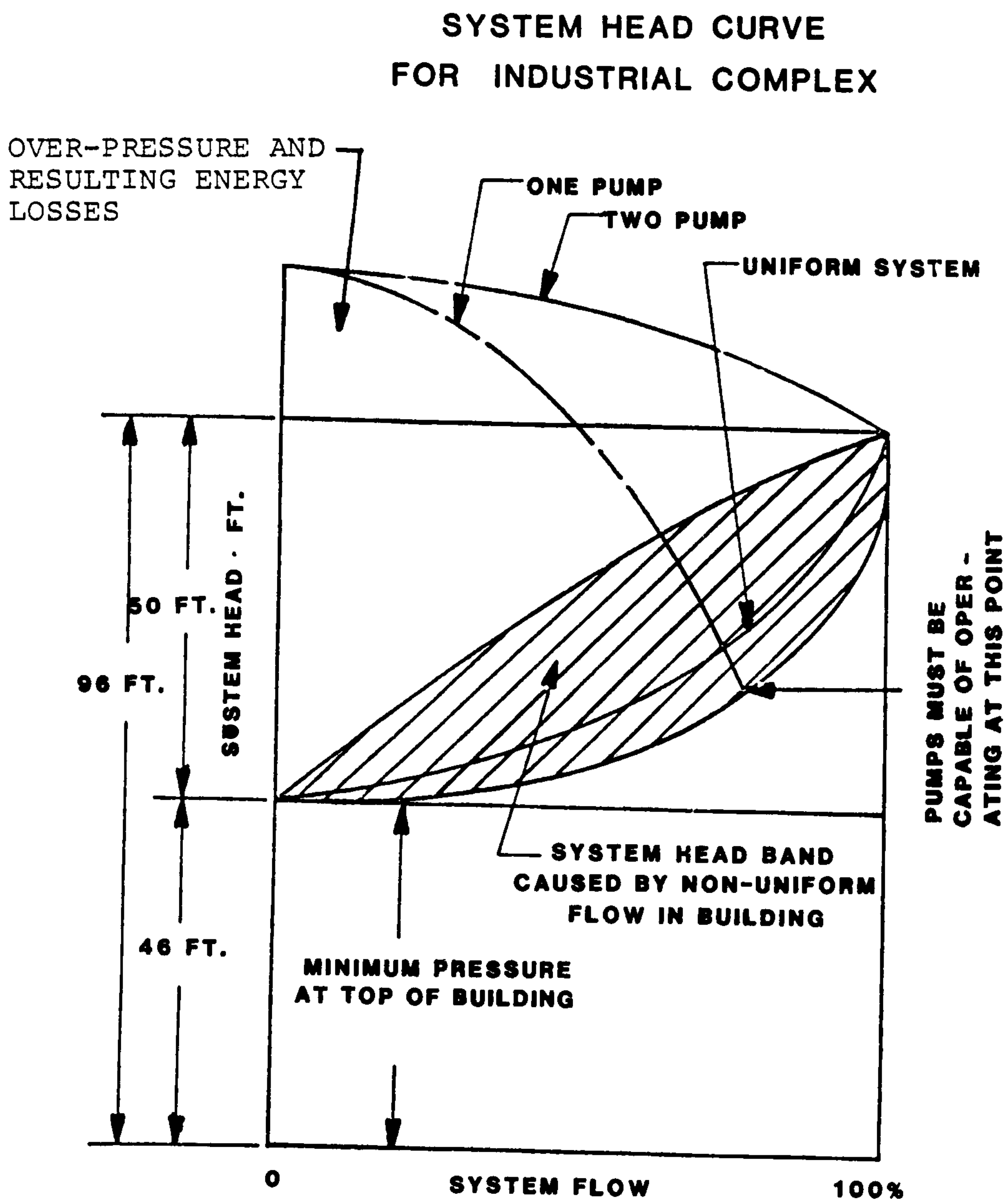


Figure 5-25. System Head Curve for Industrial Complex (shown in Figure 5-24).

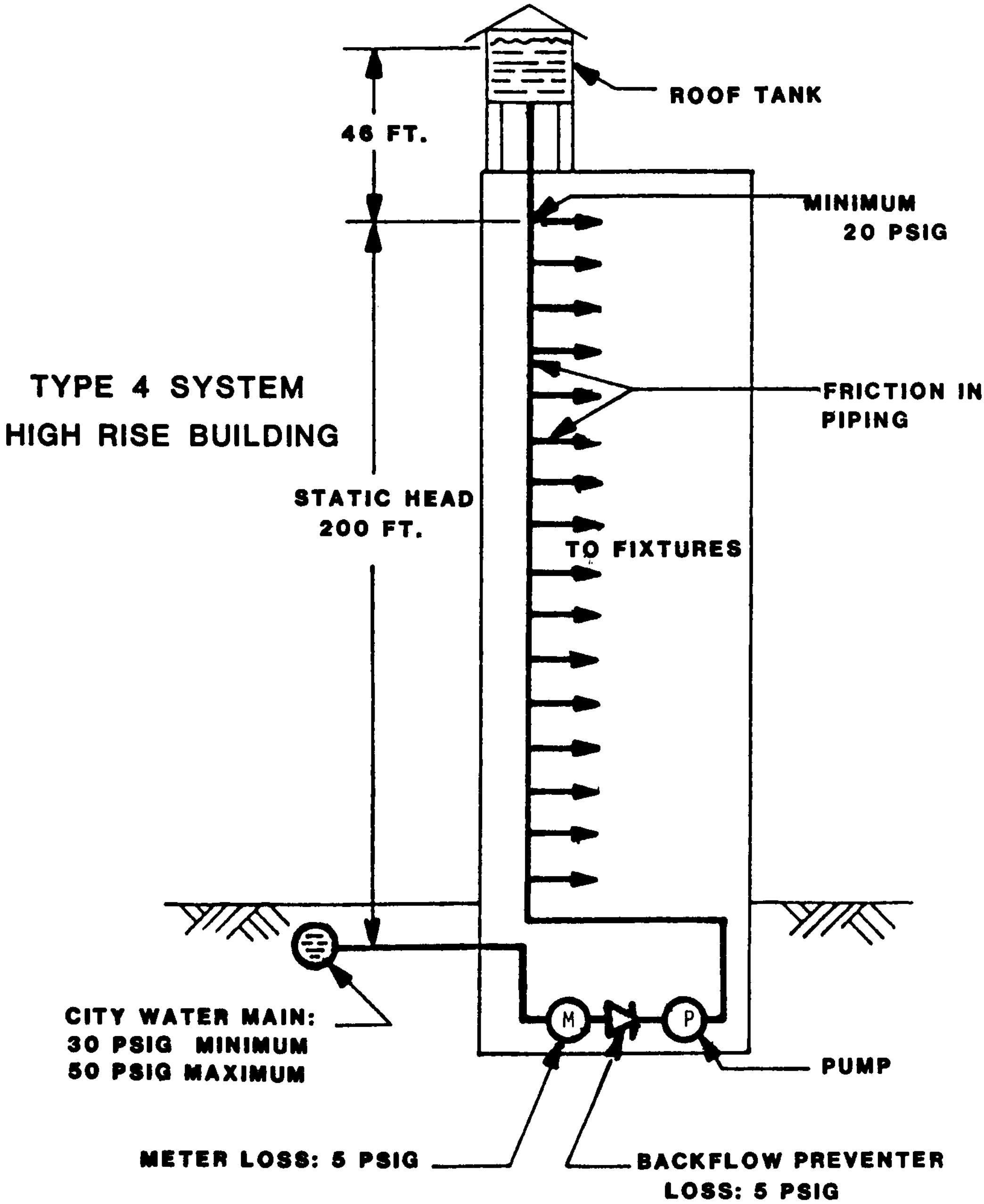


Figure 5-26 . Type 4 System (High Rise Building)

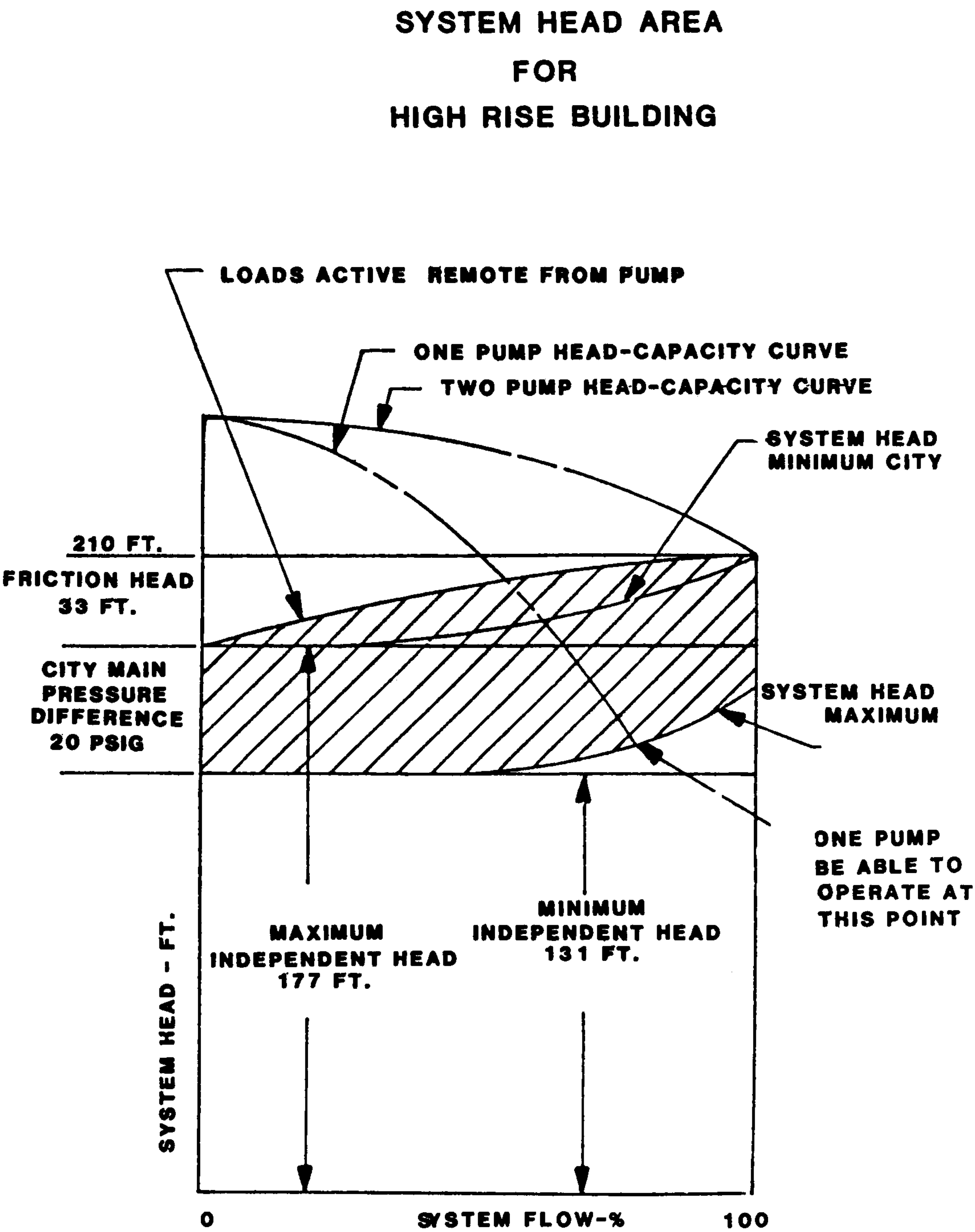


Figure 5-27 . System Head Area for High Rise Building (shown in Figure 5-26) .

closed system, the operation point on the pump curve will depend on the total flow in the building, and will not be at the intersection of the system head curve and the pump head capacity curve. Single pump operation must be checked at the far right condition when maximum pressure occurs in the city water main. This building can be a candidate for variable speed if there is sufficient variation in street pressure. In any case, either variable speed or pressure regulating valves must be installed on the pump to prevent over-pressuring in the building. The important factor here is efficiency, both at the far right condition and at the design condition, as the pumps will operate up and down the curve. The type of pump could be vertical turbine, horizontal split case or end suction, depending on actual system design flow.

Like the high rise building for the Type 3 system, this building should be equipped with a jockey pump to provide water at low flow rates in the building. The jockey pump can be used to maintain pressure in the diaphragm tank and eliminate large motor operation on low building flows.

Figure 5-30 describes a Type 5 system for an industrial complex receiving water from a city water main and delivering it to the building, maintaining a pressure of 20 PSIG at the far building. If the fixtures at the far building are at an elevation of a maximum of 36 feet, the static head will be 36 feet plus 4 or 40 feet above the city water main. The independent head consists of static head plus the minimum pressure required at the top of the building minus the minimum street pressure, 30 PSIG, or as shown in Figure 5-27, a maximum of 17 feet.

When the street pressure rises to 50 PSIG maximum, the independent head actually becomes negative, -30 feet, as shown in Figure 5-31. The reason is street pressure is greater than static head and the pressure required at the end of the system (i.e. 20 PSIG). The maximum friction of 73 feet results from the pressure drop through the water meter, back flow preventer and piping. A combination of the variation in the friction and in the street pressure creates a very broad system head area, or band, as shown in this figure. The dotted cross-hatched area at the bottom of the system head area indicates conditions under which no pumps are required. Therefore, a bypass should be provided around the pumps (See Figure 5-30) to allow the city water pressure to feed the building under low flow conditions. The shaded area represents over-pressuring and resulting energy loss that can be caused by using constant speed pumps. With two pumps servicing this building, it is obvious that pump selection must be a very careful procedure. First, a single pump will carry out far to the right to provide 90 percent of the flow under some conditions. Therefore, the pump must be able to operate without any cavitation at this high flow rate.

Variable speed pumping is a must for this type of system. Energy losses through constant speed pumping are so high that it should be easy to amortize the basic cost of the variable speed pump. The pumps should be operated by a pressure transducer located at the far end of the building.

At 20 PSIG the pumps will run at the desired speed to maintain this pressure, regardless of the flow and head variations in building. If the suction pressure rises to the point where it is capable of pumping the building, a transducer located on the pump suction header can program the pumps off and allow the city pressure to maintain building supply until the load in the building again rises to the point where pumps are required.

Horizontal split case or end suction pumps are the only kinds of pumps that should be considered for this application. Variable speed drives preferably should be of the variable frequency type to secure maximum efficiency.

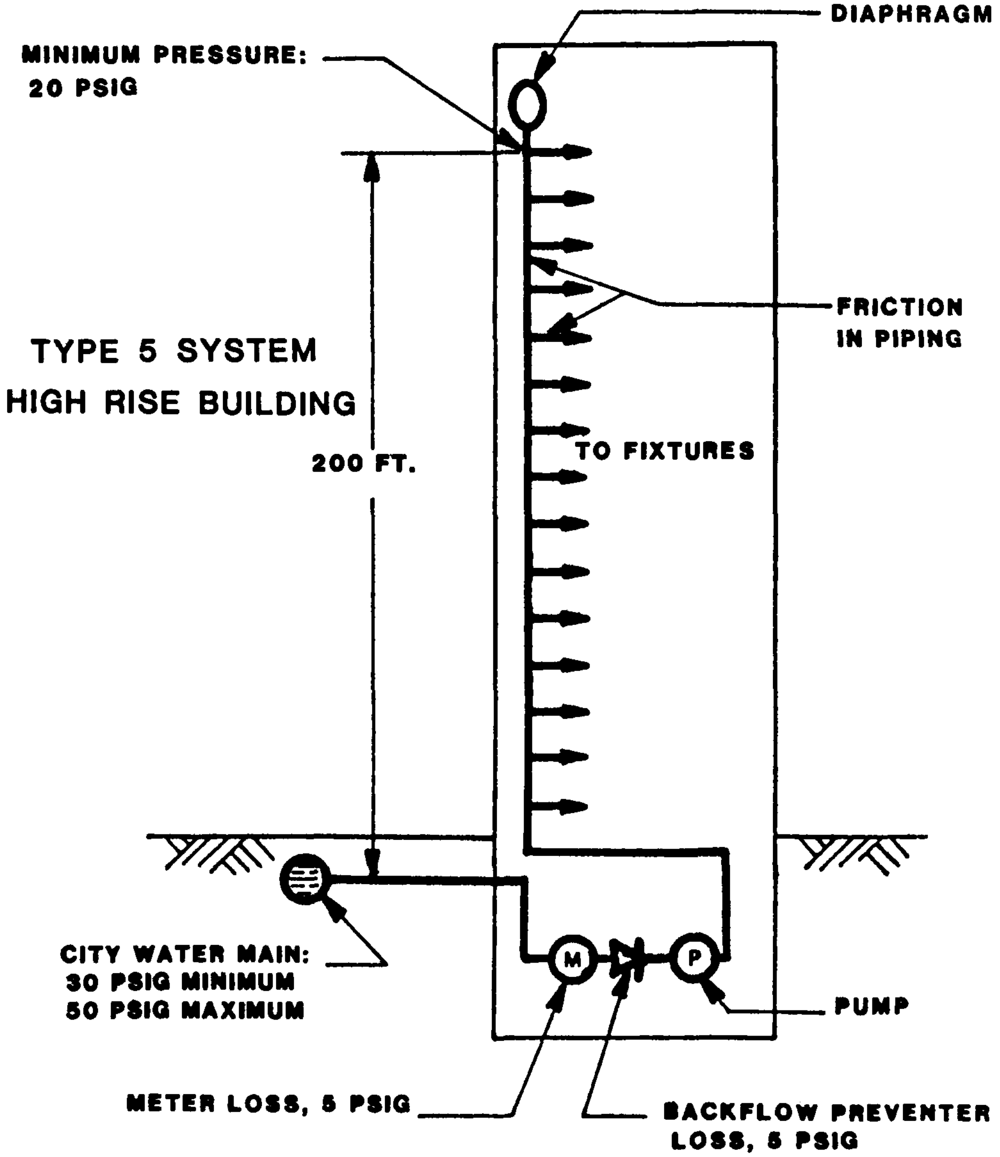


Figure 5-28. Type 5 System (High Rise Building).

SYSTEM HEAD AREA FOR HIGH RISE BUILDING

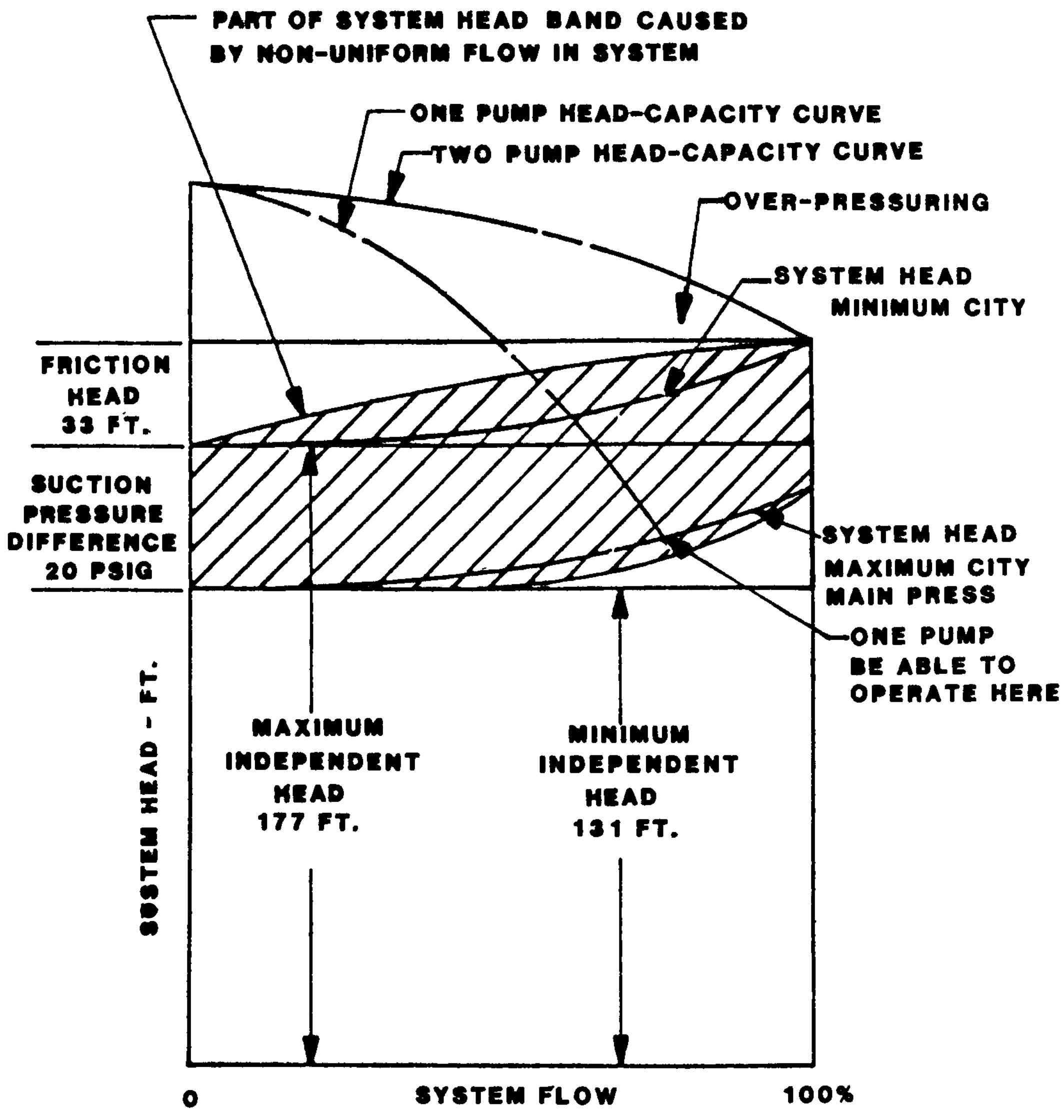


Figure 5-29. System Head Area for High Rise Building (shown in Figure 5-28).

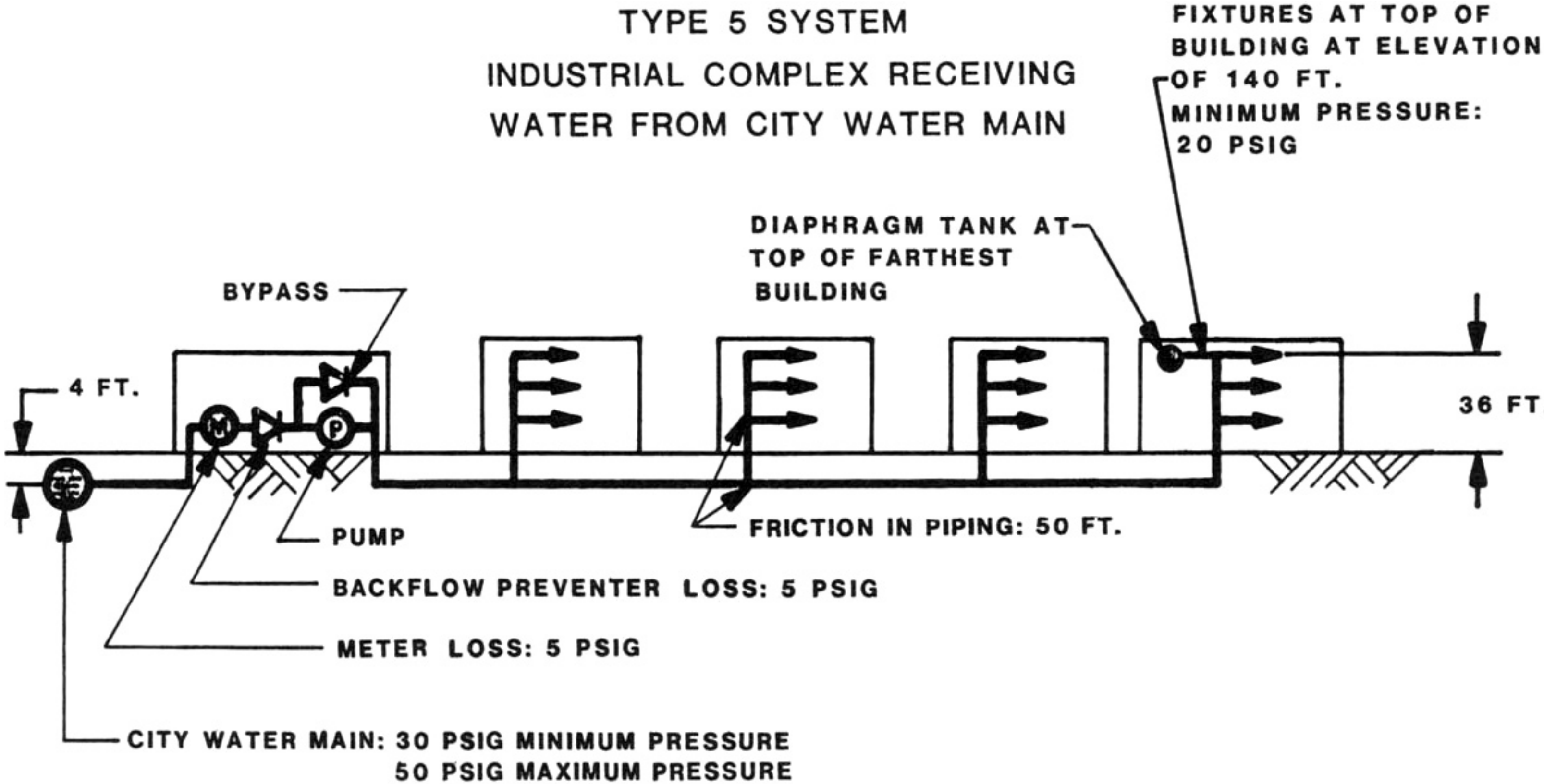


Figure 5-30. Type 5 System (Industrial Complex Receiving Water from City Water Main)

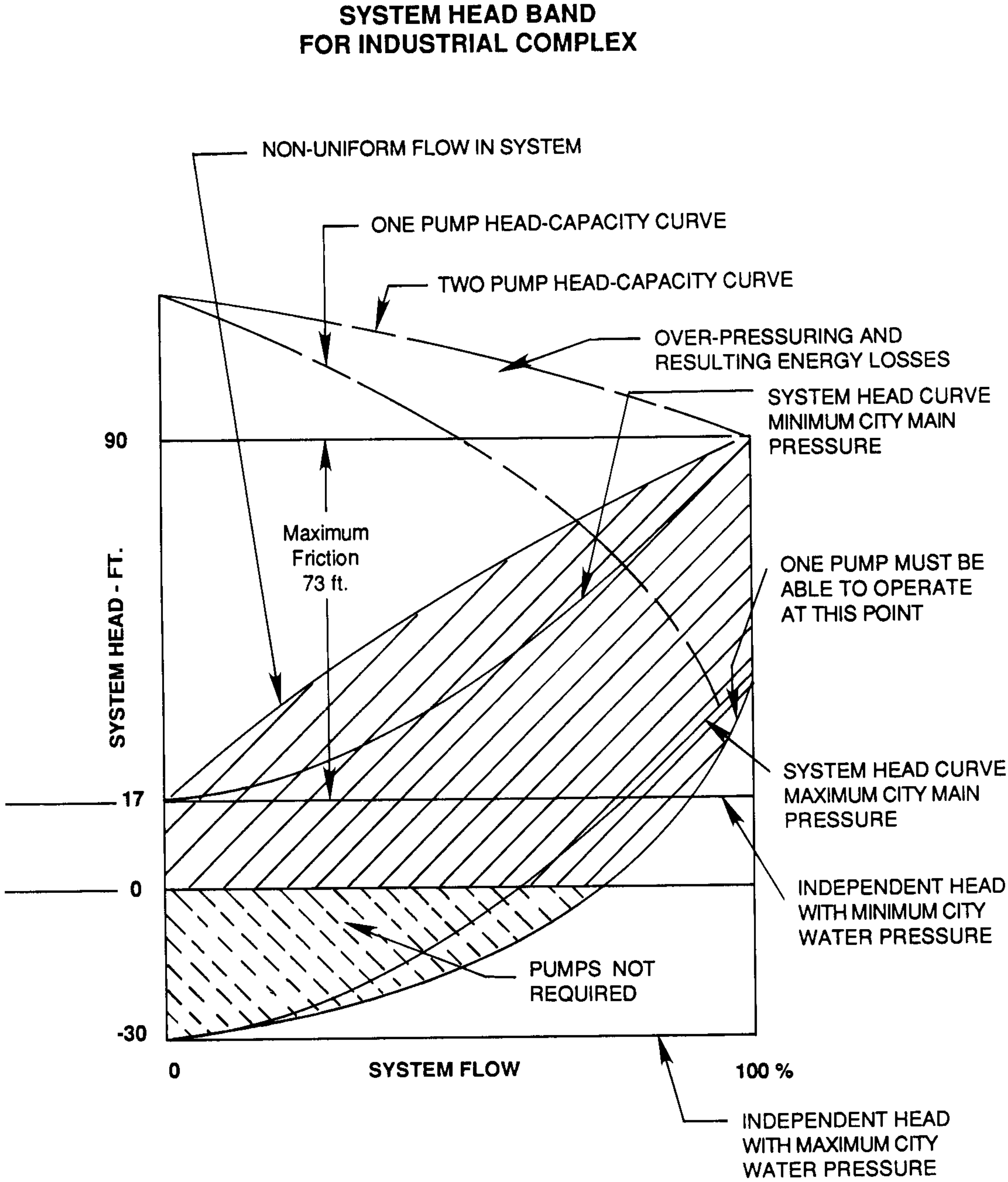


Figure 5-31. System Head Band for Industrial Complex (shown in Figure 5-30).

Direct current drives and motors should never be used for these applications due to low brush life achieved on DC motors applied to variable torque loads, such as centrifugal pumps. Eddy current - and fluid coupling-type drives can provide a reliable means of variable speed for such an application, but they are far less efficient than the variable frequency drive.

Previous discussion of the five types of plumbing systems reveals the following:

- A water system requiring pumping in plumbing should be evaluated from a minimum flow with the generation of a system head curve or area to prove the needs for various types of pumps and drives
- No one pump-type can be used on all pumping systems in public buildings. For equal efficiencies and pump motor horsepowers, horizontal split case and end-suction pumps should be preferable to vertical turbine pumps
- Flat curve pumps should be used instead of steep curve pumps to avoid over-pressuring and resulting energy losses
- Jockey pumps should be provided on high rise buildings to eliminate operation of large pumps during low flow periods, which can occur day and night
- Hydropheumatic tanks for low system flow should be installed at the top of the building, not in the basement, to achieve greater use and less costly tank installations
- Steep curve pumps should not be used to pack hydropneumatic tanks because of the very high cost of energy created by such operation
- A careful evaluation of a water system's pump head and energy requirements from minimum to maximum flow, will result in more efficient pumping

5.13 References

1. "Criteria for the Selection of Domestic Water Booster Pumps for High Rise Buildings," by David Hanson, Plumbing Engineer, Jan./Feb. 1980, pages 12-16 (Figs. 1-12). Edited with permission.

2. "System Head Curves and Areas in Plumbing Systems," by James B. Rishel, ASPE 8th Biennial Convention, 1982, (Figs. 1-26). Edited with permission.

3. "Water Pressure Boosting Systems," (Part I and II), by David Hanson, Plumbing Engineer. May/June 1975 and July/Aug. 1975, (Figures 1, 2 (Pt. II). Edited with permission.

CHAPTER 6

DESIGNING DOMESTIC WATER BOOSTER SYSTEMS TO CONSERVE ENERGY

6.1 Introduction

Considerable amount of energy is wasted by water booster systems as a result of improper sizing, poor pump selection, continuously running pumps, and outdated standards and codes. Each of these parameters has an impact on system design, operation, maintenance, and energy projection. Engineers must carefully study the existing systems to come up with the best possible pumping system design, based on actual observed usage and proven equipment performance. This chapter will discuss potential methods to conserve energy in designing water booster systems.

6.2 Proper Sizing

Proper sizing is not an easy task, since most designers do not know the exact demand for a given system. Estimation of water demand in buildings is often based on the fixture unit to gpm conversion known as Hunter's Curve (see Figure 6-1).

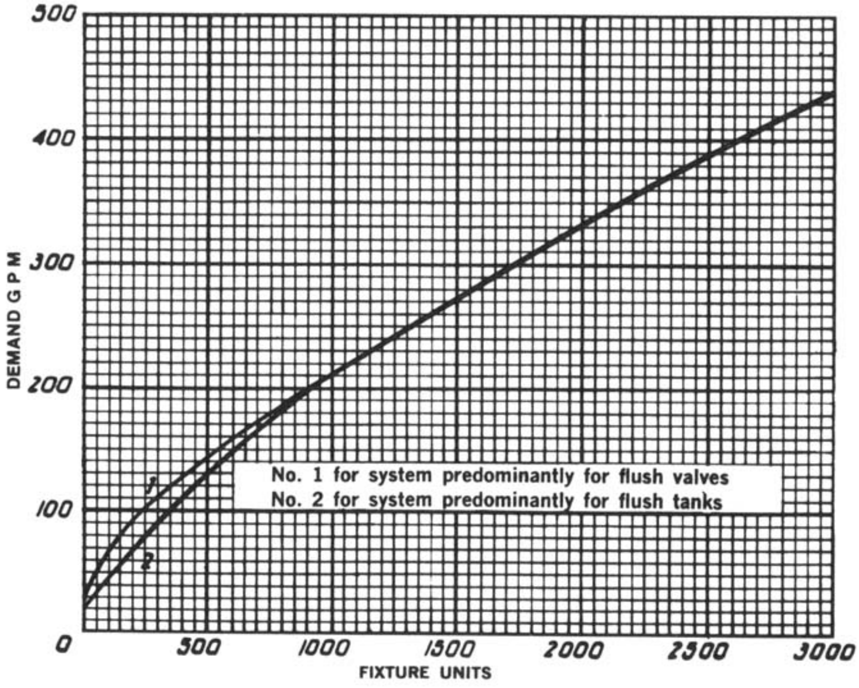


Figure 6-1. Hunter's Curve.

This curve was based on a probability analysis done by Hunter at the National Bureau of Standards in 1940. This design procedure is the framework of present design method in both the United States and British Standards. In recent years, practicing engineers have questioned the accuracy of Hunter's method and the resulting product, Hunter's Curve, which has been proven to be as much as 100 percent inflated in some instances. The acknowledged overdesign has a significant effect not only on the initial cost of the building but also on operating costs including excess energy costs as a result of oversized inefficient pumping systems.

When pumps are oversized and do not operate at or near best efficiency, the flow capacity being delivered by the pump is at a very inefficient point on the pump head capacity curve. Studies have shown that domestic water booster systems operate at less than 25% of capacity 75% of the time. This indicates that the pump selection based on the Hunter's Capacity would operate at poor efficiencies.

6.3 How Many Pumps?

Multiple pump systems save energy by running a smaller lead pump to maintain system pressure during low flow periods. Figure 6-2 illustrates a typical apartment building water demand over a 24-hour period.

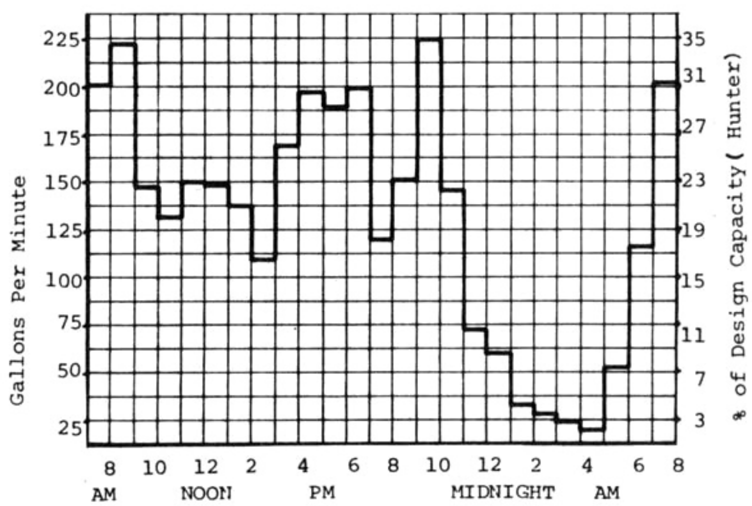


Figure 6-2. Hourly Average Demand Chart for Low Income Housing (480 units).

Hourly average demand chart has shown that capacity split is important. In general, for projects requiring less than 200 gpm it is not likely that energy savings can be achieved by using more than 2 pumps. However, for systems larger than 200 gpm a small lead pump or jockey pump will usually save significant amount of energy. When a jockey or small lead pump is used, it should be sized for a minimum of 50 gpm if the project system contains flushometers. This is because the fixture flow rate of a flushometer is as much as 50 gpm.

Table 6-1 is a guide for capacity splits for various size and type of buildings. After the capacity and pressure booster of each pump has been selected, the designer should consider power consumption of the components of the system.

Table 6-1
Suggested Capacity Splits

TYPE OF OCCUPANCY	Peak Load (gpm) 0-250	251-500	501-1000
APARTMENTS	65-65	20-40-40	20-40-40
OFFICE BUILDING	20-40-40	30-40-40	30-40-40
SCHOOLS	30-40-40	25-50-50	25-55-55
HOTELS	50-50	20-40-40	20-40-40
MOTELS	65-65	30-40-40	30-40-40
	20-40-40	25-55-55	25-55-55
HOSPITALS	65-65	30-40-40	30-40-40
	30-40-40	25-55-55	25-55-55
	30-70-70	30-70-70	30-70-70
INDUSTRIAL	50-50	20-40-40	20-40-40
	20-40-40	30-40-40	30-40-40

The selection of optimum efficiency pumps can be best achieved with an open mind about pump type. The designer should examine both centrifugal and vertical turbine selections to fit project flow and head conditions. The designer also should consider both 1,750 and 3,500 rpm speeds. Some duty points are better in efficiency at 1,750, and others at 3,500 rpm. Often motor efficiency is higher at 3,500 rpm.

6.4 Shutdown Systems

Excessive amounts of energy are wasted by booster systems if pumps are permitted to continuously run when the small or zero flow occurs. These long periods of low flow should prompt the designer to consider devices that allow the pumping system to shut-down completely.

All pumping systems have some leakage. Any device that shuts down all pumping equipment without provisions for water storage under pressure must not be considered. Since water is an incompressible fluid. Small leaks would immediately drop system pressure and cause the system to restart.

There are two types of devices that do allow long down times – the HydroCumulator type system and the modified Hydropneumatic type system. The Hydropneumatic tank is utilized as a jockey on the system, and the HydroCumulator acts as a jockey on the inlet to the pressure control devices.

In general, the same capacity HydroCumulator or Hydropneumatic tank located high in the building would store twice the volume of water providing twice the pump off time. Figure 6-3 is a typical pressure - volume storage table for different initial pressures.

The designer must recognize that these type systems should not be used on building applications with high base loads. Occupancies such as hospitals, large convention type hotels, etc. would not likely experience long periods of low flow or no flow and should not consider the Hydropneumatic or HydroCumulator type systems. Other structures such as office buildings, smaller hotels and motels, apartment buildings, recreational facilities, restaurants, etc. with long periods of low flow should consider this type of system.

The potential energy savings with the Hydropneumatic or HydroCumulator type systems depend on the horsepower of the pump being shutdown, the cost of power, and the lead load. Table 6-2

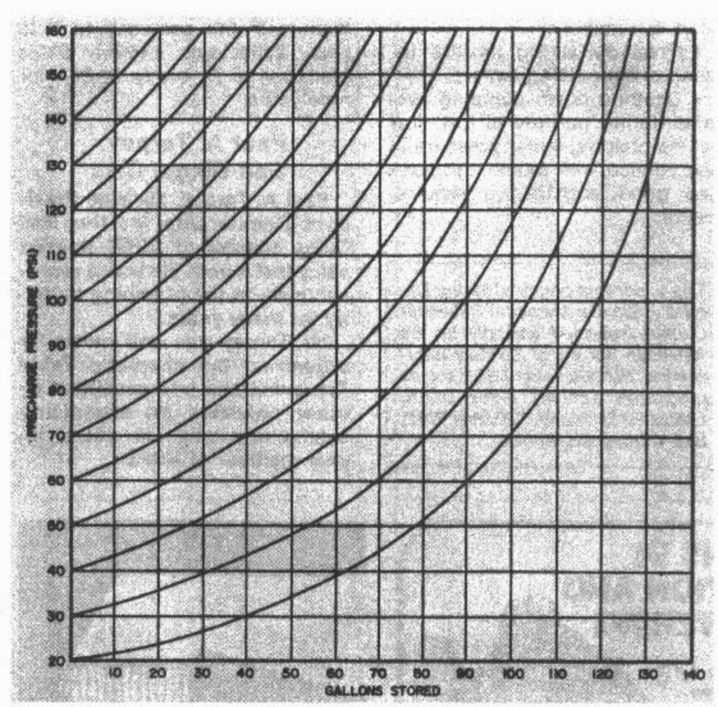


Figure 6-3. Pressure-Volume Storage Graph.

tabulates some typical yearly savings for various horsepowers. This table is for an office building experiencing night time and week-end low flows, and recognizes that even under low flow conditions the pump must recycle occasionally to refill the tank. The potential dollar savings are quite significant.

Table 6-2
Potential Yearly Savings

Motor Horse Power	Low-Flow Horse Power	Annual Savings When Your Per Kilowatt Hour Cost Is:			
		.04	.06	.08	.10
10	6.6	$ 892	$ 1,338	$ 1,784	$ 2,230
15	10.0	1,338	2,007	2,676	3,345
20	13.3	1,784	2,676	3,568	4,460
25	16.6	2,230	3,345	4,460	5,575
30	20.0	2,676	4,014	5,352	6,690
40	26.6	3,568	5,352	7,136	8,920
50	33.3	4,460	6,690	8,920	11,150
60	40.0	5,352	8,028	10,704	13,380
75	50.0	6,690	10,035	13,380	16,725
100	66.6	8,920	13,380	17,840	22,300

6.5 Energy Conservation Tips

While it is true that "design" pump efficiency increases with increased flow, it is important that every effort be made to specify gpm to the lowest possible flow rate. This is because the decrease in BHP through reduced flow is much more important than the BHP change caused by decreased efficiency in the smaller pump.

Decreased pump head specification, on the other hand, provides a compound advantage; pump BHP is reduced by head reduction, while pump efficiency is also increased - further reducing pump BHP needs.

The actual "pay-out" cost of pump operation will be influenced by motor drive efficiency. There is need for introduction of a new term - Utility Horsepower (UHP) which is simply BHP divided by drive efficiency. Most constant speed electric motors operate at about 85% efficiency, so BHP as shown on the pump curve should be divided by .85 to obtain UHP before converting to KWH. Table 6-3 shows the cost of operation for various time periods per BHP and based on 85% motor efficiencies at various KWH costs.

Table 6-3

Pumping cost per 1 BHP based on 85 percent motor efficiency

	$ per kilowatt hour					
Operating time	0.025	0.03	0.035	0.04	0.05	0.06
1 hr	0.022	0.0264	0.0308	0.0351	0.0438	0.0525
12 hr	0.263	0.316	0.368	0.42	0.524	0.628
24 hr	0.526	0.632	0.735	0.84	1.048	1.256
30 days (1 month)	15.80	19.00	22.10	25.20	31.40	37.60
6 months	95.20	116.00	134.00	154.00	192.00	230.00
9 months	143.00	175.00	202.00	232.00	289.00	346.00
1 yr	193.00	232.00	270.00	308.00	384.00	460.00

Table 6-3 shows the importance of pumping energy conservation — especially when it is considered the present national average electric cost of 0.060/KWH is shortly expected to double. The basic information described can be utilized in terms of reducing pump power draw for pressure boosting systems.

6.6 Single Pump - The Base Example

It should be noted that pump efficiency is under control of the pump design engineer, while the terms ft. hd. and gpm are under control of the system design engineer. Overall system pump operating efficiency (operating cost) is established more by engineers concerned with system design and pump specification (ft. hd. - gpm), than by pump design engineers.

To illustrate this point, consider a high-rise building (1,000 F.U. at 210 gpm) of about 200' height equipped with flush valve closets. Facilities and other operations on the lower floor require 70 gpm. A roof top cooling tower and boiler need about 20 gpm at 20 psi. Total flow need is stated at 300 gpm.

Street pressures may vary from the maximum of 60 psi to the minimum of 40 psi in this building. For safety, a low street pressure of 30 psig is used. Future street loading may further reduce street pressure.

The basic design example is shown as Figure 6-4.

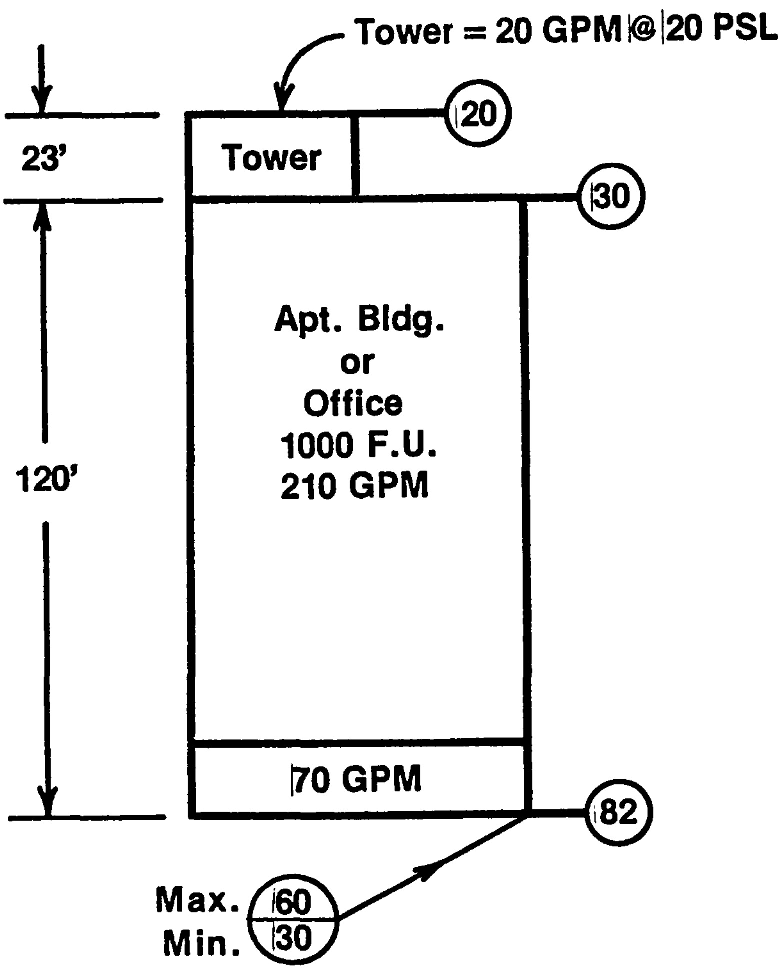


Figure 6-4. The Basic Design Example.

Either variable or constant speed pumps can be used for the base example. Given that either pump will be selected for 300 gpm, the pump head selection points will be defined as follows:

Variable Speed Selection

Static Hd. = 82 psi - 30 psi = 52 psi; 52 x 2.3 = 120'
Pipe Friction Flow Hd. Loss at 1'/10'; 120 x 1'/10' = 12'
TOTAL HEAD = 132'

Pump selection based on 300 gpm at 132'

Constant Speed Selection

Constant speed pump application often includes a pressure reducing valve (PRV) to reduce pressure as it is delivered to the building when street pressures are expected to be highly variable (as in this example). To retain control, the PRV must be selected for a relatively high order head loss. Valve selection head loss will vary between 10 and 20'. A PRV head loss of 15' is used in this example.

Static Hd. = 82 psi - 30 psi = 52 psi; 52 x 2.3 = 120'
Pipe Friction Flow Hd Loss at 1'/10' = 120' x 1'/10' = 12'
Pressure Reducing Valve (PRV) Head Loss = 147'

Pump Selection Based on 300 gpm at 147'

Noting that usage of the PRV introduces more head loss (more pump power consumption) for the constant speed pump with PRV, as compared with the variable speed pump - or for a constant speed pump without the PRV.

Methods for reducing power consumption for both the variable speed and the constant speed pressure booster pumps follow. Constant speed energy conservation methods will be examined first - principally because the constant speed pump provides the simplest introduction to the basic methods. It should be understood that the basic methods for constant speed pressure boosting energy conservation also apply to variable speed pumps.

At the point of 300 gpm at 147 ft. head, a pump selection can be made by referring to pump curves.

A pump as illustrated on Figure 6-5 is selected with a 6-5/8" impeller.

The selection point at 300 gpm and 147' shows a pump efficiency of 70%. The 40% efficiency line crosses the 6-5/8' impeller curve at 88 gpm and 182'.

$$\text{at 300 GPM;} \quad \text{BHP} = \frac{300 \text{ gpm} \times 147 \text{ ft. hd.}}{3960 \times .70} = 16 \text{ BHP}$$

$$\text{at 88 GPM;} \quad \text{BHP} = \frac{88 \text{ gpm} \times 182 \text{ ft. hd.}}{3960 \times .4} = 10.1 \text{ BHP}$$

Given the illustrated pump head and efficiency points, a plot of BHP usage vs. GPM can be made for the 6-5/8" impeller as in Figure 6-6.

Pressure booster pumps will operate across a wide flow draw range (and BHP range) as shown in Figure 6-5. The pump in this case will draw 16 BHP at full flow draw (300 gpm) and 8 BHP at low flow draw (shutoff).

The BHP curve in Figure 6-6 was derived for the base example and will be used as a reference for evaluation of pump energy conservation methods for pressure booster pumping systems.

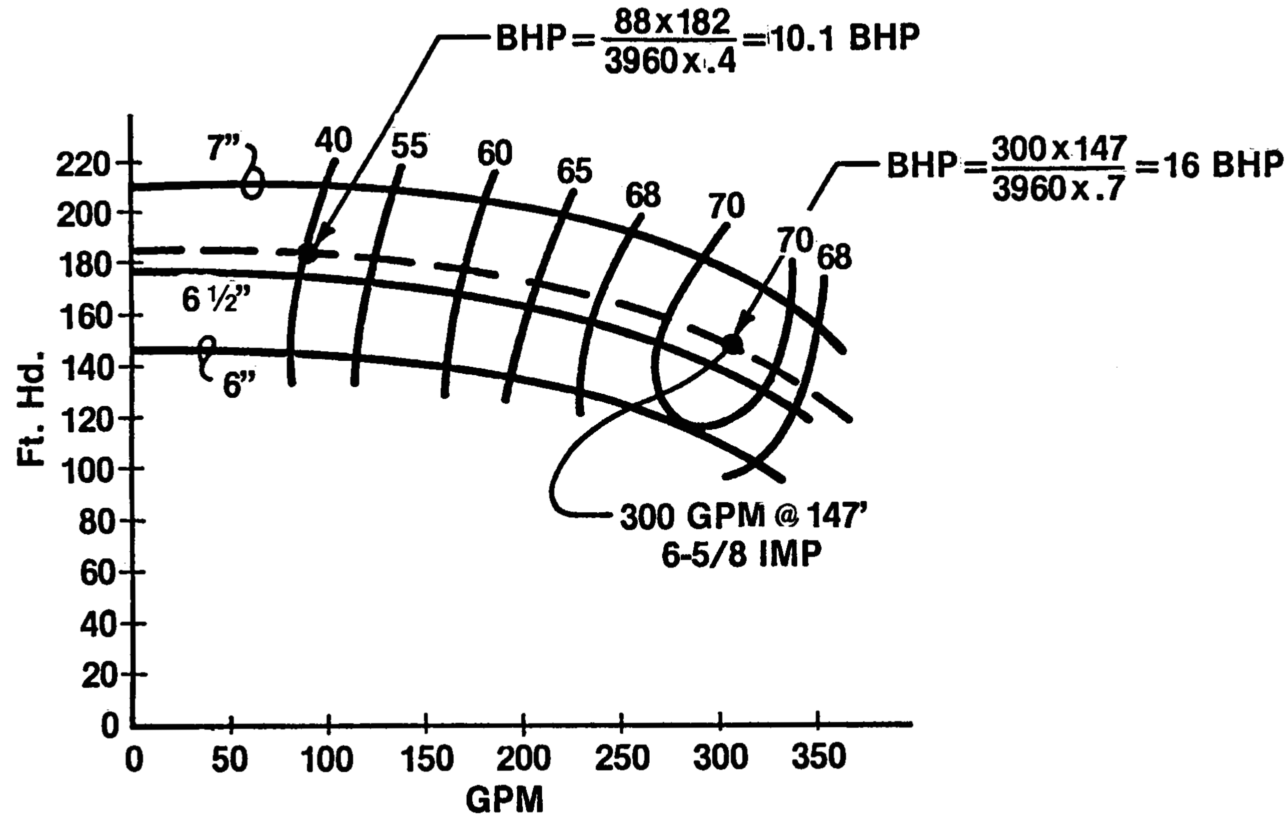


Figure 6-5. Pump Selection for 300 GPM at 147'.

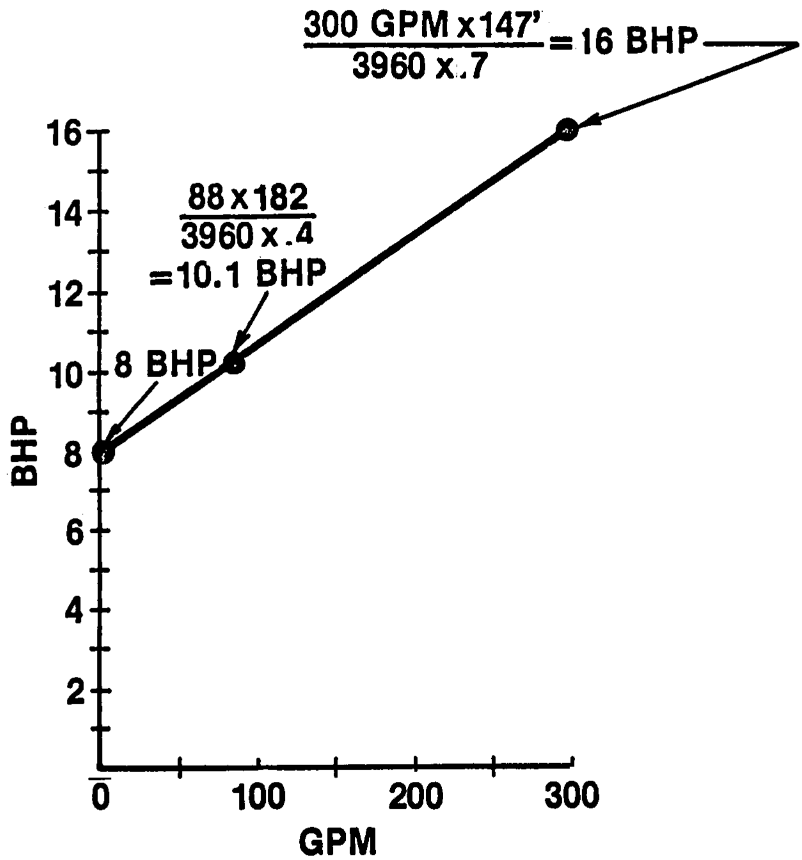


Figure 6-6. Constant Speed BHP Variation.

6.7 Energy Conservation Methods And Evaluation Procedures

The basic approach to booster pump energy conservation is simple:

Specify the pump or pumps for the lowest possible ft. hd. and gpm.

There are several important points that should be remembered in terms of evaluation:

- Booster pumps will operate across a wide flow range. It is important to assess low flow BHP because low flow operation occurs for long time periods.

- Recent studies indicate maximum probable flow will only be about .6 that shown as probable by the Hunter F. U. curve.

In order to assess operating costs for Booster pumping systems, some correlation between building flow draw demand as an hourly function should be established. Table 6-4 shows hourly flow demand as a percentage of peak demand obtained directly from Hunter's F. U. probability flow.

Table 6-4

Hourly Flow Demand

APARTMENT BUILDING		OFFICE BUILDING	
Hrs. at % Flow	% Flow*	Hrs. at % Flow	% Flow*
4	25	12	0
6	30	1	20
4	35	4	30
4	40	6	40
2	45	1	60
3	50		
1	60		

*% Flow is % of Hunter F. U. Probability Flow

It should be recognized that Table 6-4 is not authoratative and can only be considered for reasonable estimation of hourly flow draw.

For evaluation purposes, full load pump BHP is defined by:

$$\text{Full Load BHP} = \frac{\text{Specified ft. hd. x Specified gpm}}{3960 \times E_p}$$

Pump efficiency (E_p) can be calculated by knowing the ft. hd., gpm, and BHP.

In order to strike the pump BHP usage curve as a function of pump or system flow, shut-off BHP must be known. For the purpose of this discussion, shut-off BHP will be one-half full load BHP.

6.8 Pump Energy Conservation by Multiple Pump Use

The basic approach to energy conservation is specifying pumps to the lowest possible gpm and ft. hd. If two pumps in parallel were specified for the base condition shown in Figure 6-7, each pump would be specified for one-half total flow (150 gpm at 147' rather than 300 gpm at 147').

Paralleled multiple pumps operate in stages. At low flow, one pump operates. When flow increases to maximum capacity of the single pump, normally a control signal starts the second pump. Any number of stages pumps can be used.

The control signal can be developed from motor amp draw (current relay) or from flow meters.

Given a selection at 150 GPM and 147', Figure 6-5 shows that each pump will operate at about 63% efficiency; a decrease from the 70% available for the larger pump. Given the conditions, full load and shut-off BHP can be determined as follows:

$$\text{Full Load BHP} = \frac{150 \times 147}{3960 \times .63} = 8.9 \text{ BHP}$$

$$\text{Shut-off BHP} = 8.9/2 = 4.45$$

The BHP plot for the multiple pump system as compared with the single pump is shown on Figure 6-8. At flows beyond 150 GPM multiple small pumps draw more power than the single large pump. The important point, however, is that at low flow (less than 150 gpm) only one small pump will operate and its power draw will be significantly less than the single large pump.

An operating cost comparison of BHP usage for the single and parallel multiple pumps can be made, considering the base example as either an apartment building or office building. The methodology follows:

Table 6-4 provides hourly operation at an estimated flow which is the percentage of the Hunter probability flow assessed in this case at about 300 gpm. For the apartment building Table 6-4 shows that the building will operate for 4 hours at 300 x .25 or 75 gpm.

Refer to Figure 6-8 at 75 gpm and determine that for 4 hours, the single large pump will draw 10 BHP/Hr. - a total BHP/Hr. of $4 \times 10 = \frac{40 \text{ BHP/Hr.}}{4 \text{ Hr.}}$.

A summation of BHP/Hr. draw per day can be established and converted to utility draw (UHP Hr/Day); based on motor drive efficiency. UHP Hr./Day can be converted to UHP Hr./Yr. draw, then to KWH/Yr. and finally changed to $/Yr. by referring to KWH Cost.

Table 6-5 illustrates calculation of yearly cost for the example considered as an apartment building and for the single large pump as compared with the multiple pump system.

Table 6-5

Calculation Of Yearly Cost For An Apartment Building

			Single Pump		Multiple Pump (2)	
Hrs. at % Design Flow	% Design Flow	Actual Flow	BHP Draw	BHP Hr. Draw Per Time Period	BHP Draw	BHP Hr. Draw Per Time Period
4	25	75	10	40/4 Hr.	6.5	26/4 Hr.
6	30	90	10.25	61.5/6 Hr.	7	42/6 Hr.
6	35	105	10.5	42/6 Hr.	7.5	30/4 Hr.
4	40	120	11	44/4 Hr.	8	32/4 Hr.
2	45	135	11.5	23/2 Hr.	8.5	17/4 Hr.
3	50	150	12.5	37.5/3 Hr.	9	27/3 Hr.
1	60	180	13	13/Hr.	13.5	13.5/Hr.
			TOTALS	261 BHP Hr. / 24 Hr.		207.5 BHP Hr. / 24 Hr.

*Design Flow at 300 GPM

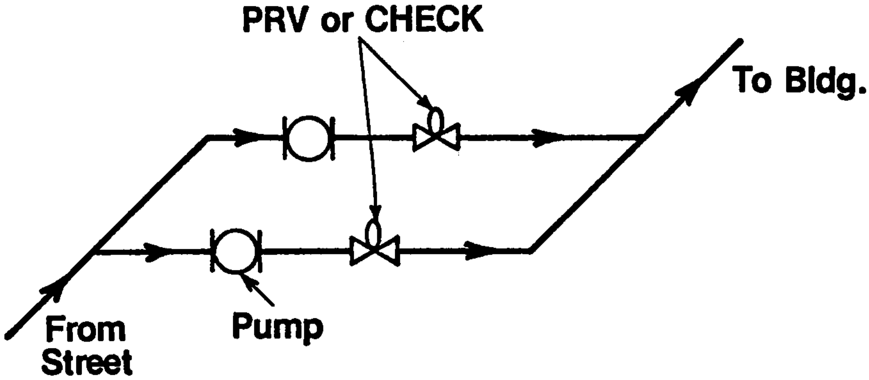


Figure 6-7. Paralleled Staged Booster Pumps.

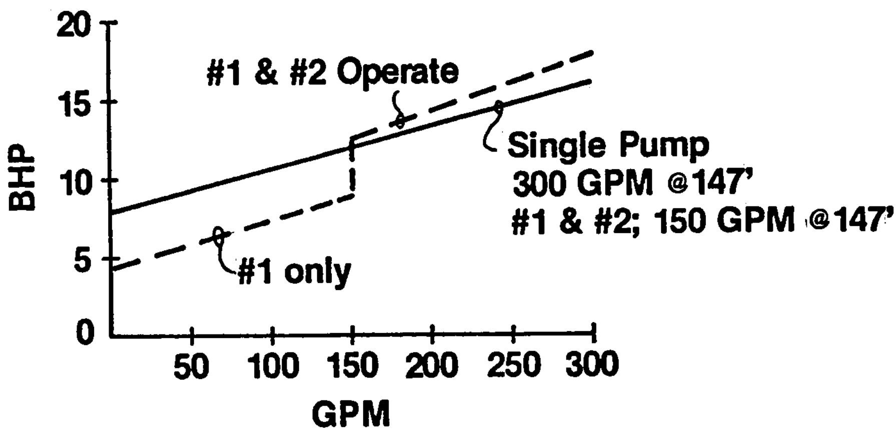


Figure 6-8. Multiple Parallel Pumps Vs. Single Large.

Operating cost/yr. will then be:

Single Pump:

$$\frac{261 \text{ BHP Hr.}}{\text{Day}} \times \frac{1}{.85} \times \frac{365 \text{ Day}}{\text{Yr.}} \times \frac{.746 \text{ KWH}}{\text{H.P. Hr.}} \times \frac{\$0.05773^{*}}{\text{KWH}} = \$4827/\text{Yr.}$$

Multiple Pump:

$$\frac{207.5 \text{ BHP Hr.}}{\text{Day}} \times \frac{365}{.85} \times .746 \times 0.05773 = \$3837/\text{Yr.}$$

*The KWH for residential buildings (without taxes) is .05773¢.

Savings will amount to about $990/Yr. for the multiple pump system. Given a doubling of electric energy cost as to $0.115/KWH, savings would amount to $1980/Yr.

A comparison should also be made for the example applied as an office building. The comparison is shown in Table 6-6.

Table 6-6
Calculation Of Yearly Cost For An Office Building

			Office Building			
			Single Pump		Multiple Pump (2)	
Hrs. at % Design* Flow	% Design* Flow	Actual Flow	BHP Draw	BHP Hr. Draw Per Time Period	BHP Draw	BHP Hr. Draw Per Time Period
12	0	0	8	96/12 Hr.	4.5	54/12 Hr.
1	20	60	9.5	9.5/ Hr.	6	6/Hr.
4	30	90	10.25	41/4 Hr.	7	28/4 Hr.
6	40	120	11	66/6 Hr.	8	48/6 Hr.
1	60	180	13	13/Hr.	13.5	13.5 Hr.
			TOTALS	$\frac{225 \text{ BHP Hr.}}{\text{Day}}$		$\frac{149.5 \text{ BHP Hr.}}{\text{Day}}$

*Design Flow at 300 GPM

Year Operating Costs:

KWH for Office Buildings (without taxes) is .05971¢.

$$\text{Single Pump} = 225 \times \frac{365}{.85} \times .746 \times 0.05971 = \$4304/\text{Yr.}$$

$$\text{Multiple Pump} = 149.5 \times \frac{365}{.85} \times .746 \times 0.05971 = \$2860/\text{Yr.}$$

Savings for the multiple pump system amount to $1444/Yr. at 0.05971/KWH or about $2888/Yr. at 0.119/KWH (future cost).

Reference to Table 6-6 shows that a percentage of operating power is consumed during the 12 hour time period of virtually no flow. A very small pump could be operated during this time period, the pump selected for the order of 50 gpm @ 147'. While pump efficiency would be low (because the pump is small) the full flow BHP would be about 4 BHP. The important point is at no flow, BHP drops to about 2 BHP. Inclusion of the small "lead" pump would change the multiple pump system from two to three pumps.

The three pump system (one small lead) would decrease operating cost/yr. to about $2286/yr. at 0.05971/KWH. Basic yearly costs are reduced to about one-half that required for the single pump system. Results are as shown on Table 6-7.

Table 6-7

Single Pump Vs. Three Pumps For An Office Building

			Single Pump		Multiple Pump(3 Pump)	
Hrs. at % Design Flow	% Design* Flow	Actual Flow	BHP Draw	BHP Hr. Draw Per Time Period	BHP Draw	BHP Hr. Draw Per Time Period
12	0	0	8	96/12 Hr.	2	24/12 Hr.
1	20	60	9.5	9.5/Hr.	6	6/Hr.
4	30	90	10.25	41/4 Hr.	7	28/4 Hr.
6	40	120	11	66/6 Hr.	8	48/6 Hr.
1	60	180	13	13/Hr.	13.5	13.5/Hr
			TOTALS	$\frac{\text{225 BHP Hr.}}{\text{Day}}$		$\frac{\text{119.5 BHP Hr.}}{\text{Day}}$

*Design Flow at 300 gpm.

It appears several tentative conclusions can be reached:

- Multiple pump systems save on operating costs for both the apartment and office building booster pumping systems. Operating cost savings are particularly accentuated for the office building
- Small lead pumps propose particularly high savings for the office building because of its long "down" time

6.9 Pump Energy Conservation by Flow Specification Reduction

In many high-rise structures, the lower floors have a relatively high water demand - and are adequately served by available street pressure. Low floor water demand can then be separated from the pressure booster system. This will require, of course, that a separate heater be provided for the separated section. Additional heater cost will have to be evaluated against booster pump operating saving. In our basic example, 70 gpm can be separated from booster pump requirements - leaving a required pump flow at 70 ft. of 300 - 70 or 230 gpm. Base pump draw conditions will then become:

<u>Single Large Pump @ 230 gpm, 147' & 68% E_p:</u>

$$\text{Full Flow Draw} = \frac{230 \times 147}{3960 \times .68} = 12.5 \text{ BHP}$$

$$\text{Shut-off Draw} = 12.5/2 = 6.25 \text{ BHP}$$

<u>Multiple Stages Pumps (2) in Parallel, Each at 115 gpm, 147' and 59% E_p</u>

$$\text{Full Flow Draw/Pump} = \frac{115 \times 147}{3960 \times .59} = 7.3 \text{ BHP}$$

$$\text{Shut-off Draw} = 7.3/2 = 3.65 \text{ BHP}$$

The comparative BHP - Flow Draw relationship is shown in Figure 6-9.

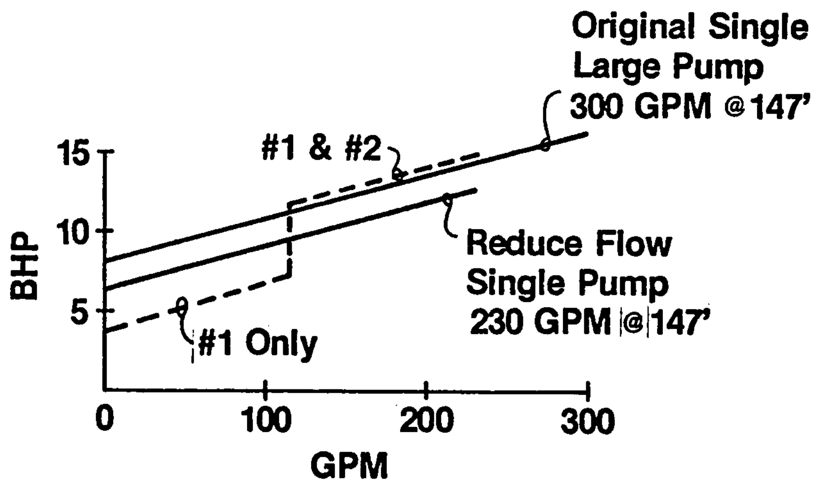


Figure 6-9. Single Vs. Multiple Pump BHP.

Application of the same technique for calculation of yearly operating cost shown in Tables 6-5 and 6-7 - but based on a lowered design flow to 230 gpm - shows year costs as in Table 6-8.

Table 6-8

Comparison of Yearly Operating Cost

APARTMENT BUILDING	BHP, Hr. / Day	Cost/Yr.
Design at 300 gpm		
Single Pump	261	4827
Multiple Pump	207.5	3837
Design at 230 gpm		
Single Pump	214.5	3967
Multiple Pump	143	2645
OFFICE BUILDING		
Design at 300 gpm		
Single Pump	225	4304
Multiple Pump	149.5	2860
Multiple Pump W/Small Lead	119.5	2286
Design at 230 gpm		
Single Pump	177	2386
Single Pump W/Small Lead	122.5	2343
Multiple Pump	123.5	2362
Multiple Pump W/Small Lead	103	1970

Separating the lower floor section from the booster pump can lower operating cost, since operating cost is a function of pump size. A decrease in specified flow will generally result in a smaller pump.

While operating cost savings would not appear to justify the extra heater cost in this instance, many large high-rise buildings require very heavy low floor draw rates. Separation of these facilities from the booster pump, in many cases, will be justified.

It is of interest to note that for the office building, the importance of the small lead pump in combination with the multiple pumps has decreased. This is because the multiple pumps themselves have become so small that they in effect have become the lead pump. The small lead pump would still be justified when used in combination with the single large pump.

6.10 Pump Energy Conservation: Pump Head Specification Reduction

Pump head specification reduction will save power, because pump BHP requirements are directly related to the ft. hd.

Figure 6-10 is stated in terms of psi reduction, as are street pressures and required top system pressures. Conservatively, about .8 BHP per 10 psi reduction will be saved for each 100 gpm flow rate specified.

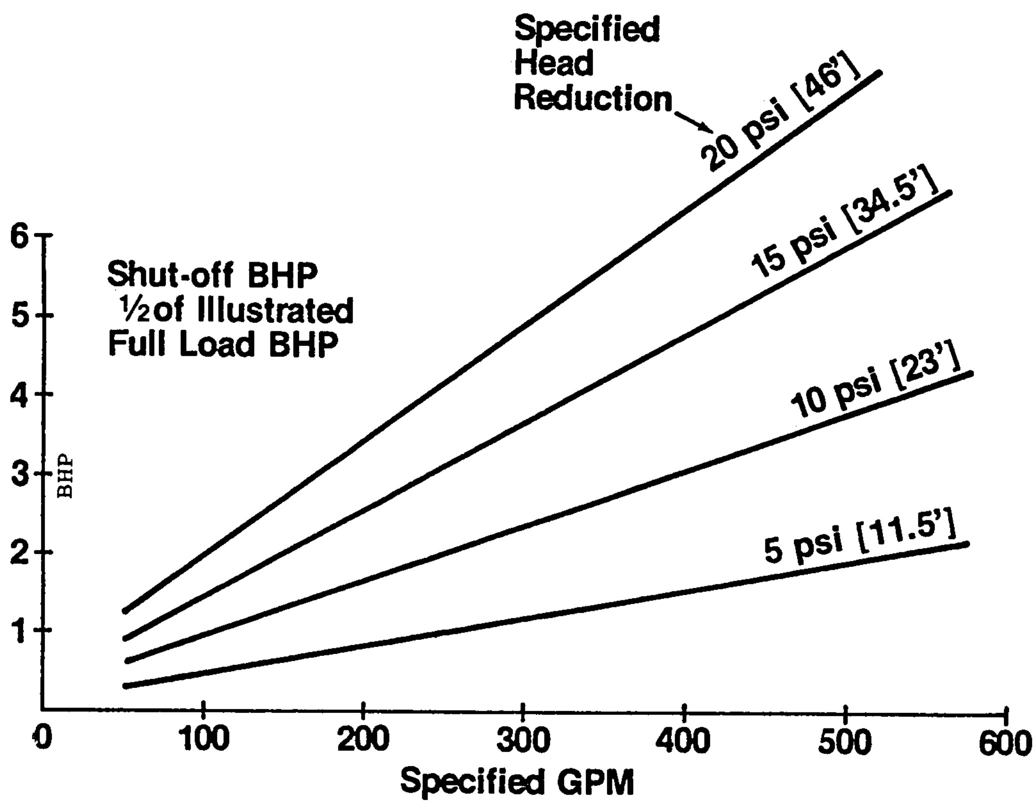


Figure 6-10. Full Load BHP Reduction for Reduction in Specified Head.

Methods for reduction of specified head can be referenced to the base example (Figure 6-4). The basic methods follow:

Reduce top floor pressure needs:

Figure 6-4 shows a roof mounted cooling tower requiring 20 gpm at 20 psi at the tower and which is elevated 23' above the top fixture. The cooling tower requirement sets the top fixture pressure at 30 psi.

The cooling tower flow and pressure requirement would be satisfied with a small separate pump drawing from the top of the main distribution riser and operating at about 1/2 BHP. This small separate pump will allow top fixture pressure requirement to establish top system pressure need.

Use of flush valves at the upper level would set a top floor pressure need to the order of 15 psi. Since the original example requires 30 psi at the "top," a reduction of 15 psi (34.5') in specified pump head would occur; the resulting BHP saving at 300 gpm of 3.6 BHP at full flow draw, and 1.8 BHP at shut-off.

Use of tank water closets at the upper floor would introduce an even greater available pump head reduction.

Thoroughly evaluate street pressure availability:

In the basic example, street pressure availability was hurriedly evaluated as ranging from 30 psi to 60 psi. A more thorough investigation (city water pressure records, etc.) reveals that present street pressure variation is from 50 to 60 psi. A future projection indicates street pressure variation will change from 40 psi to 60 psi in about 5 years.

Eliminate pressure reducing valve when street pressure variation is less than 20 psi:

A change in top pressure to 20 psi will not be serious. Aerating faucets have removed the "water hammer" problem causing broken glass or china breakage at the kitchen sink. Small PRV can be used to pressure zone the CW side to reduce valve seat wear at closets and flush valves.

With the introduction of the described head saving methods, a new diagram (Figure 6-11) can be drawn describing the change in pressure requirements for the new evaluation.

For the new condition, a single pump would be selected so that its pump casing would accommodate an impeller capable of at least 74' at 300 gpm (E_p at .75). The pump could be operated initially, however, for the 51' head requirement and with an impeller suited to the low head. Use of the smaller impeller would decrease pump efficiency to 70%. Pump BHP draws follow:

Initial Pump at 300 gpm and 51'

$$\text{Full Flow BHP} = \frac{300 \times 51}{3960 \times .7} = 5.5 \text{ BHP}$$

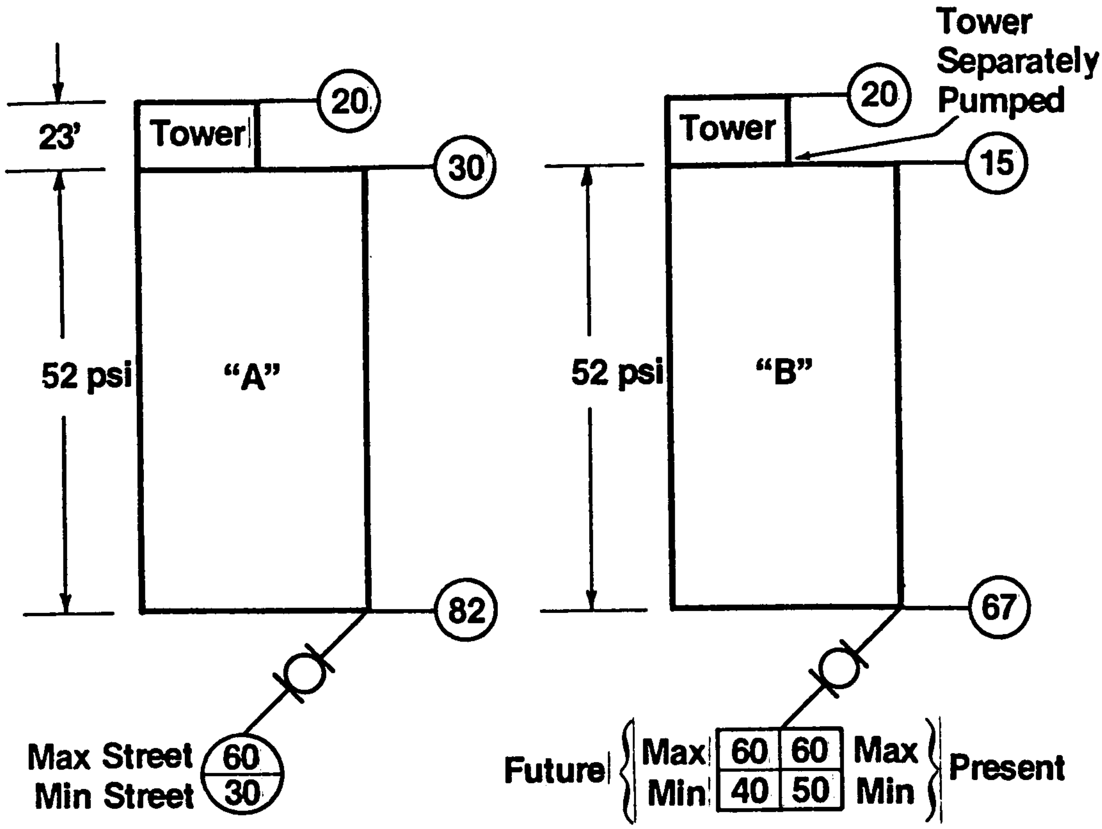
$$\text{Shut-off BHP} = 5.5/2 = 2.75 \text{ BHP}$$

When future needs require, the initial impeller can be replaced with a full size runner. Then, BHP will increase as follows:

$$\text{Full Flow BHP} = \frac{300 \times 74}{3960 \times .75} = 7.5 \text{ BHP}$$

$$\text{Shut-off BHP} = \frac{7.5}{2} = 3.75 \text{ BHP}$$

A comparison of multiple pumps selected for 150 gpm at 74'



INITIAL PUMP HEAD

Static	= (82 - 30)2.3	=	120'
PRV	=		15'
Pipe Loss	= 1/10 x 120	=	12'
	TOTAL	=	147'

Pump at 300 gpm and 147'

PRESENT PUMP HEAD

Static	= (67 - 50) 2.3 =	39'
Pipe Loss	=	12'
	TOTAL	51'

Pump at 300 gpm and 51'

FUTURE PUMP HEAD

Static	= (67 - 40) 2.3 =	62'
Pipe Loss	=	12'
	TOTAL	74'

Pump at 300 gpm and 74'

Figure 6-11. Energy Conservation Through Head Reduction.

(2 pumps) as against a single pump selected for initial conditions (300 gpm at 51') and for a single pump at final conditions (300 gpm at 74') is shown in Figure 6-12.

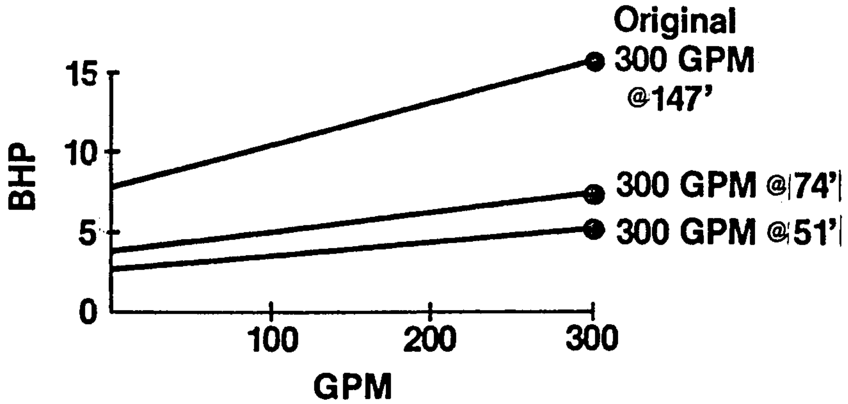


Figure 6-12. Reduced BHP for Reduced Head Specification.

A comparison of single operating costs is shown in Table 6-9.

Table 6-9

A Comparison of Single Operating Costs

APARTMENT BUILDING

System Pump Selection	BHP, Hr./Day	Cost/Yr. at 0.05773 KWH
Original - 300 gpm at 147'	261	$4827/Yr.
Initial (New) - 300 gpm at 51'	87.2	$1613/Yr.
Final (New) - 300 gpm at 74'	114.8	$2123/Yr.

OFFICE BUILDING

System Pump Selection	BHP, Hr./Day	Cost/Yr. at 0.05971 KWH
Original - 300 gpm at 147'	225	$4308/Yr.
Initial (New) - 300 gpm at 51'	47.1	$ 901/Yr.
Final (New) - 300 gpm at 74'	67.32	$1288/Yr.

The cost comparison includes about $200/Yr. needed to operate the cooling tower pump at 20 gpm and 34'. As will be noted, however, the trade-off of the cooling tower pump for reduced "top" pressure need in combination with a reevaluation of street pressure availability pays off.

As previously noted, the plumbing engineer can specify the pump to the initial condition illustrated and include in the pump specification need for and cost of the impeller needed for final operation. Over 5 years savings of $2214 to $2500 should be instituted. This more than pays for the minor order cost of the larger "ready to be installed" impeller.

6.11 Impeller Trimming for Already Installed and Operating Booster Pumps

Many installed Booster Pumps are overheaded, because street pressures are actually higher than anticipated by the design specification. If the building shown in Figure 11 "A" had a pump size as (300 gpm at 147') - and this pump could be reduced to 300 gpm at 74'-a yearly saving of $3214 would be established for either the apartment or office building.

Impeller trimming can match Booster Pump capacity to system needs. The minor cost of impeller trimming establishes that large savings can be established for minor investments.

Office building savings would become even greater if a small lead pump were to be added and impellers trimmed.

6.12 Roof Tank Systems

Roof tank systems offer the greatest potential savings of <u>any</u> pumping system. This is because the pump operates on an "on-off" time cycle; <u>when the pump is "off" no power is needed</u>.

Despite this apparent advantage, roof tank systems have become practically obsolete. Reasons for obsolescence seem to be both architectural and structural, and finally reference back to an old fire code need - requiring tanks have about a 3 hour supply of water for fire fighting. The tanks consequently become so large they required additional building structural strength.

Nowadays fire pumps provide fire fighting water, permitting large reductions in tank size. The use of roof mounted equipment rooms (boilers, chillers, towers, etc.) provide potential hiding for the relatively small tank needed for domestic water service.

Separate cooling tower and boiler feed pumps can be used to provide pressure for these services although they are located at the same level as the tank.

Considering the building illustrated in Figure 6-4, minimum pressure at the top fixture is estimated to be about 10 psi (water closet). The tank water level is about 23' above the top fixture. Figure 6-13 shows the changed building pressure conditions.

The roof tank system requires a pump head as follows:

Static	= (62 - 30) 2.3	= 74'
Pipe Loss	= (120 + 23) 1/10	= 14'
	Pump Head at 300 gpm	= 88'

When operating the pump would run at 300 gpm and 88, having an approximate BHP as follows:

$$BHP = \frac{300 \times 88}{3960 \times .7} = 9.6$$

Time period of pump operation can be estimated as shown in Table 6-10:

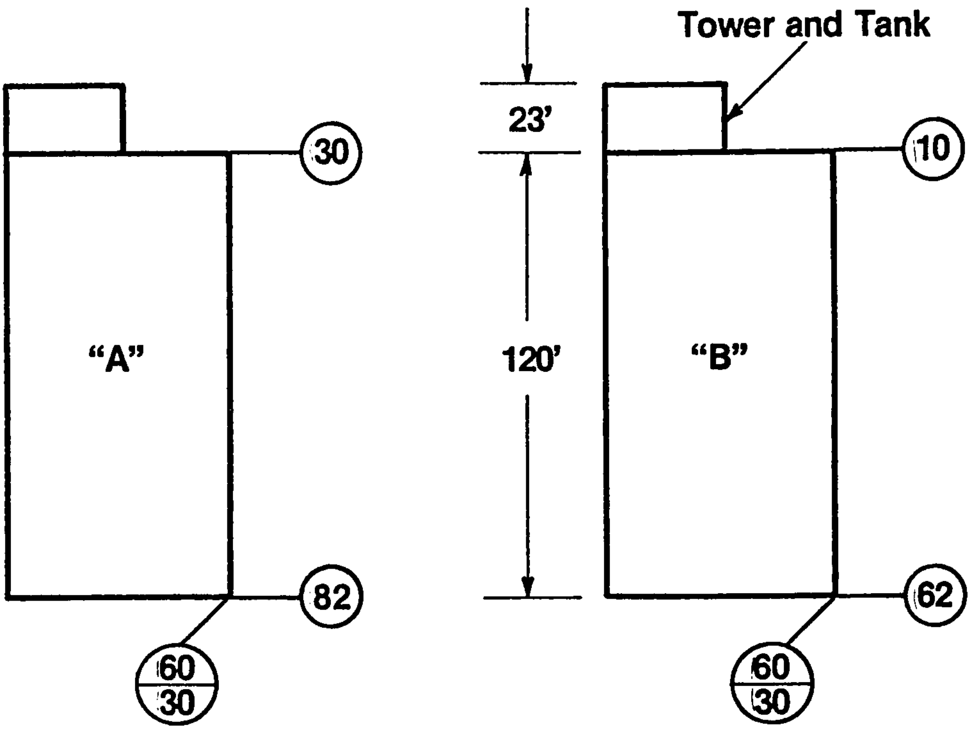


Figure 6-13. Roof Tank System.

Table 6-10

Time Period of Pump Operation

APARTMENT BUILDING			OFFICE BUILDING		
Hrs. @ % Design	% Design Flow	Actual Hours Operation	Hrs. @ % Design	% Design Flow	Actual Hrs Operation
4	.25	1	12	0	0
6	.30	1.8	1	20	.2
4	.35	1.4	4	30	1.2
4	.4	1.6	6	40	2.4
2	.45	.9	1	60	.6
3	.5	1.5			
1	.6	.6			
		8.8 Hr./Day			4/4 Hr./Day

Since pumps draw 16 BHP while operating the $\frac{\text{BHP Hr.}}{\text{Day}}$, yearly cost will become:

Apartment Building:

$$9.6 \text{ BHP} \times \frac{8.8 \text{ Hr.}}{\text{Day}} = 85 \ \frac{\text{BHP, Hr.}}{\text{Day}}, \text{ and } \$1572/\text{Yr.}$$

This should be compared with original 261 $\frac{\text{BHP, Hr.}}{\text{Day}}$ and $\frac{\$4827}{\text{Yr.}}$ - for the same conditions and for a single large pump.

The savings become even more substantial for the office building because of the long shut-down in terms of water draw:

Office Building:

$$4.4\ \frac{\text{Hr.}}{\text{Day}} \times 9.6\ \text{BHP} = 42.5\ \frac{\text{BHP, Hr.}}{\text{Day}} \text{ and } \$813/\text{year.}$$

The above calculations can be compared to the \$4014/Yr. need for a conventional single pump application.

Roof top tank systems should be re-evaluated in terms of energy conservation, as fire pumps can substantially reduce the required tank size.

6.13 Basement Tank Systems with Pressure Booster

Many city distribution systems have been inadequately sized in terms of actual draw imposed by unexpected high-rise construction. To solve this problem, some city codes require that "open" basement storage tanks are time sequence filled so that only half of the connected buildings are allowed to fill during a certain period of time. This, of course, doubles the flow capacity of the city main.

The booster pump, in this case, operates from 0 pressure at its pump suction to required pressure at the pump discharge. Pressure "boost" from the city main is lost and booster pump power consumption is increased.

It is interesting to speculate and note, however, that the combination of a basement storage tank and a relatively small roof tank provides for a reduction in city main flow draw with minor booster pump operational cost.

6.14 Variable Speed Pressure Booster Pumps

Variable Speed Pressure Booster Pumps are used to reduce power consumption and maintain a constant building supply pressure, despite variation in street pressure.

For any given <u>fixed</u> flow rate, BHP will vary approximately as the square of the flow change; a pump speed decrease to one-half design speed will decrease pump BHP to about 1/4 design.

The decrease in BHP with pump speed reduction is one of the salient features of variable speed drive pressure booster pumping. It is not often noted, however, that the decrease in pump BHP is partially countered by a decrease in "slip type" variable speed drive efficiency; the lower the pump speed the lower the drive efficiency. A plot of drive efficiency vs. % speed is shown in Figure 6-14.

Almost all commercially used variable speed drives operate in a "slip" manner. Design slip is at 10% for fluid drives and SCR interrupted voltage units. As noted in Figure 6-14, decreased slip increases drive efficiency.

Non-slip drives which maintain high order efficiency at low speed are available (SCR variable frequency, etc.) - but are so costly they are not often commercially applied.

The usual decrease in drive efficiency is of major importance because we do not pay for BHP - but rather for input power into the drive unit. This requires information on variable speed drives in terms of utility HP (UHP)(BHP divided by drive efficiency E_D).

It is important to understand how a picture of the operating characteristics of a variable speed drive can be obtained. The first step is to define the pump selection point. Figure 6-4 illustrates the base example and states the variable speed selection point; 300 gpm at 132' head.

Maximum variable speed in rpm is about 90% of synchronous speed. Since basic pump curves are plotted in terms of synchronous speeds, the true selection point (300 gpm at 132') must be converted to a synchronous speed selection point. This is accomplished as follows:

Synchronous speed selection:

$$\text{gpm} = \frac{300}{.9} = 334 \text{ gpm}$$

$$\text{ft. hd.} = \frac{132}{.9 \times .9} = \frac{132}{.81} = 163 \text{ ft. hd.}$$

Standard 3500 rpm curves are now checked for a selection at 334 gpm and 163 ft. hd. The pump shown in Figure 6-5 would be suitable; requiring about a 6-7/8" impeller. This replotted pump curve is shown in Figure 6-15. It will be noted that efficiency points have been plotted on the pump curve.

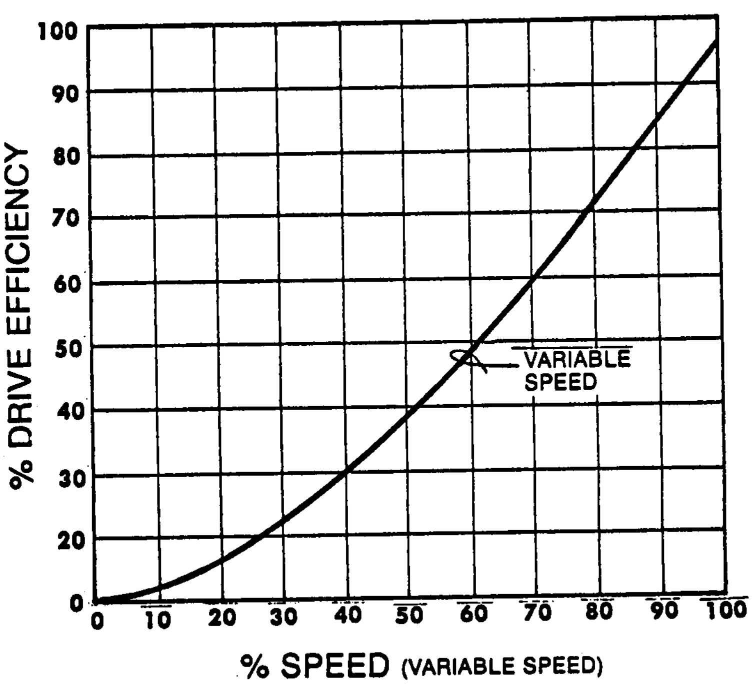


Figure 6-14. Percentage of Drive Efficiency Vs. Percentage of Speed.

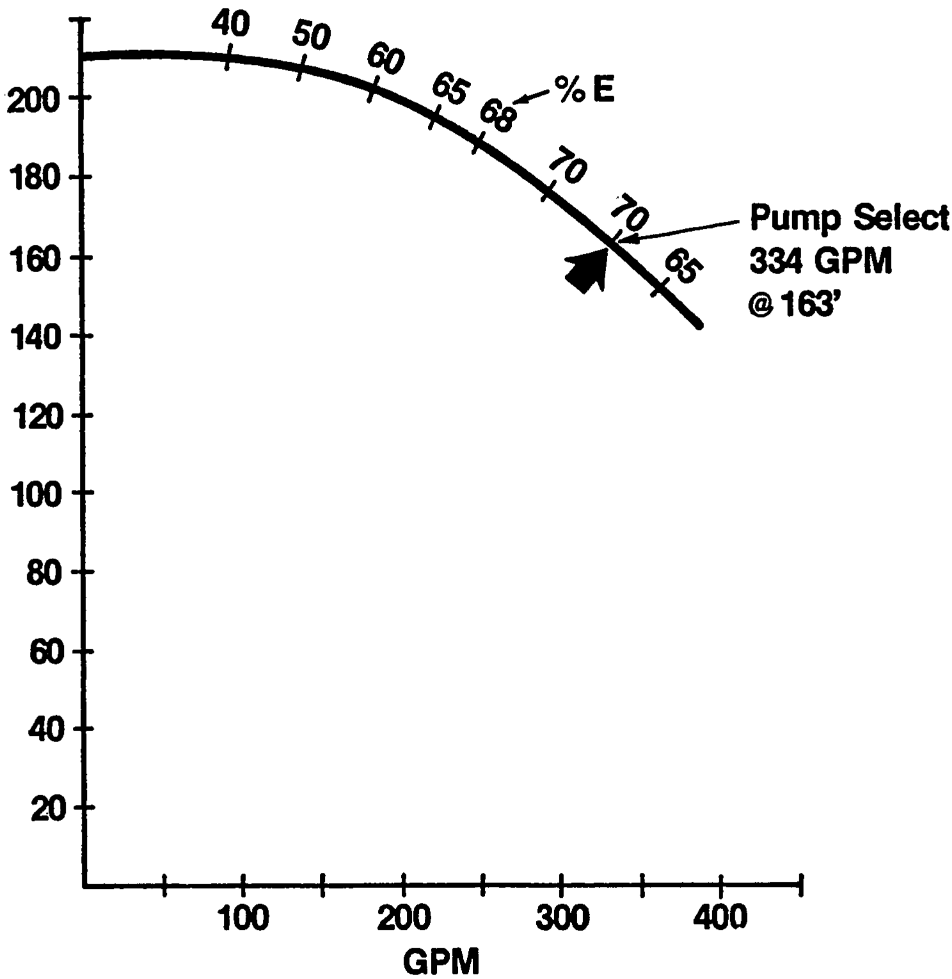


Figure 6-15. Efficiency Points on the Pump Curve.

For the variable speed pump, "constant efficiency" lines can be plotted. The constant efficiency line is similar to a system curve and follows the proposition that head varies as a function of the squared change in flow.

Complete gpm-ft. hd. conditions can be tabulated as in Table 6-11.

Table 6-11

% Speed

		100	90	80	70	60	50	40	30	20
70% E_p	gpm	334	300	257	227	193	161	128	97	64
	ft.hd.	163	132	97	76	54	38	24	13.8	6

Similar plot points for 65%E at 230 gpm and 192' and for 40%E at 92 gpm and 202' will result in three plot points for each speed - providing a plotting curve for each % speed, as shown in Figure 6-16. The constant efficiency lines also permit a cross plot of pump efficiency.

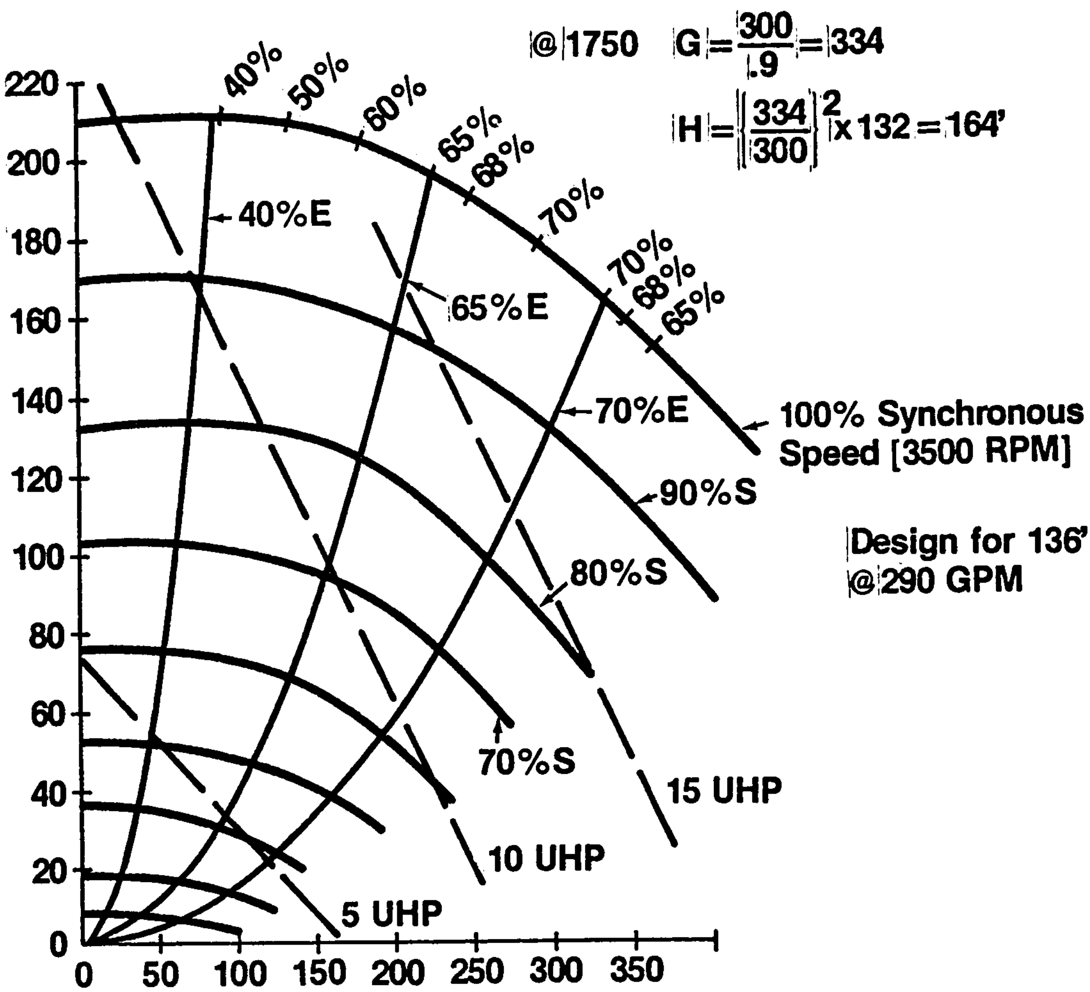


Figure 6-16. Constant Efficiency Lines Vs. Constant Speed Lines.

The previously prepared table includes ft. hd., gpm and efficiency - all that is needed to determine BHP. UHP can be determined by dividing BHP by drive efficiency (E_D), as available from Figure 6-14. The fully developed table showing UHP appears as Table 6-12.

Given this Table, we can plot a complete picture of variable speed performance, as in Figure 6-17. Figure 6-17 can be used to evaluate variable speed pump operation. Given that the apartment building shown in Figure 6-4 will operate with a street pressure variation as between 30 and 60 psig, the variable speed pump would operate at about 80% speed (132') with 30 psig street pressure. The pump would operate at about 55% speed when city pressure is at 60 psig. Assuming the pump operates at each condition for 50% of the year, Table 6-13 can be made for the variable speed pump in comparison with other pumping methods:

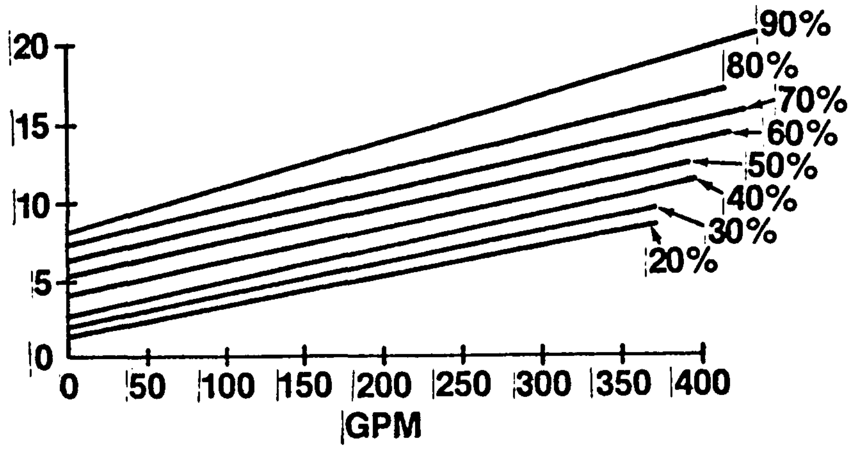


Figure 6-17. Variable Speed Performance.

Table 6-12

The UHP Table for Variable Speed Pumps

		100	90	80	70	60	50	40	30	20
	GPM	334	300	257	227	193	161	128	97	64
	Ft. Hd.	163	132	97	76	54	38	24	13.8	6
70% E_p	BHP		14.3	8.9	6.25	3.62	2.21	1.11	.484	.138
	E_p		.85	.70	.53	.40	.30	.20	.125	.05
	UHP		16.9	12.7	11.8	9.05	7.4	5.6	3.9	2.75
	GPM	230	207	184	161	138	115	92	69	46
	Ft. Hd.	192	156	122	94	69	48	30.5	17.4	7.7
65% E_p	BHP		12.5	8.7	5.9	3.69	2.15	1.085	.465	.137
	E_D		.85	.70	.53	.40	.30	.20	.125	.05
	UHP		14.8	12.4	11.1	9.2	7.2	5.45	3.72	2.75
	GPM	95	85.5	76	66.5	57	17.5	38	28.6	19
	Ft. Hd.	202	164	130	100	73	51	32.5	19.4	8.4
40% E_p	BHP		8.9	6.25	4.2	2.63	1.53	.78	.35	.098
	E_D		.85	.70	.53	.40	.30	.20	.125	.05
	UHP		10.4	8.9	7.95	6.6	5.1	3.9	2.81	1.95

Table 6-13

The UHP Draw For A Variable Speed Pump
(Apartment Building)

			80% Speed		55% Speed	
Hr.	% Flow	Actual Flow	UHP Draw	UHP, Hr. Draw	UHP Draw	UHP, Hr. Draw
4	25	75	9	36	5	20
6	30	90	9.5	57	5.25	31.4
4	35	105	9.75	39	5.75	23
4	40	120	10	40	6	24
2	45	135	10.25	20.5	6.25	13
3	50	150	10.5	31.5	6.5	19.5
1	60	180	11	11	7	7
				235 $\frac{\text{UHP}}{\text{Day}}$		137.9 $\frac{\text{UHP}}{\text{Day}}$

$\frac{235 + 137.9}{2}$ = 186.5 UHP/Day; $2931/Yr.

In comparison the constant speed pumps would draw:

Single Pump = $\frac{\text{261 BHP, Hr.}}{\text{Day x .85}}$ = 301 UHP/Day; $4827/Yr.

Multiple Pump = $\frac{\text{207.5 BHP, Hr.}}{\text{Day x .85}}$ = 244 UHP/Day; $3837/Yr.

Roof Tank = $\frac{\text{85 BHP, Hr.}}{\text{Day x .85}}$ = $1572/Yr.

Variable speed pump proposes savings in the apartment building as contrasted with the multiple pump system. The apartment building example has operating conditions most suited to variable speed pump application:

- High order street pressure variation (30 psi to 60 psi)
- Relatively continuous high order flow draw

Given that street pressure is relatively constant, at 30 psig, the variable speed pump would operate at a fairly constant 80% speed at all times and UHP draw/day would approximate 235 UHP/day and about $3694/Yr. Operating cost for the variable speed pump would closely approximate multiple constant speed operating cost when street pressures are constant, and when a relatively continuous high order flow draw condition obtains.

Office buildings are illustrative of systems that have long no draw time periods. Given a high order street pressure variation (30 to 60 psig), operating costs for the example office system would appear as follows:

Table 6-14

The UHP Draw For A Variable Speed Pump
(Office Building)

			80% Speed		55% Speed	
Hr.	% Flow	Actual Flow	UHP Draw	UHP, Hr. Draw	UHP Draw	UHP, Hr. Draw
12	0	0	7.5	90	3.75	45
1	20	60	8.5	8.5	4.75	4.75
4	30	90	9.5	38	5.25	21
6	40	120	10	60	6	36
1	60	180	11	11	7	7
				207.5 $\frac{\text{UHP}}{\text{Day}}$		113.75

$$\frac{113.75 + 207.5}{2} = \frac{160.6 \text{ UHP, Hr.}}{\text{Day}}$$ or About 2611/Yr.

The comparison is as follows:

Single Pump = $\frac{225 \text{ BHP, Hr.}}{\text{Day x .85}} = \frac{265 \text{ UHP, Hr.}}{\text{Day}}$; \$4308/Yr.

Multiple Pump (2 Pump) = $\frac{149.5 \text{ BHP, Hr.}}{\text{Day x .85}} = \frac{176 \text{ UHP, Hr.}}{\text{Day}}$; \$2861/Yr.

Multiple Pump w/Lead = $\frac{119.5 \text{ BHP, Hr.}}{\text{Day x .85}} = \frac{140 \text{ UHP, Hr.}}{\text{Day}}$; \$2276/Yr.

Roof Tank = $\frac{42.5 \text{ BHP, Hr.}}{\text{Day x .85}} = \frac{50 \text{ UHP, Hr.}}{\text{Day}}$; \$813/Yr.

The comparison illustrates that despite high street pressure variation, the effect of long time periods of low flow draw establishes that the multiple pump system with a lead pump operates at lower cost than the variable speed pump; $2276 per year as against $2611/Yr. for the variable speed application.

Given a constant street pressure at 30 psig, operating cost for the variable speed pump would increase to $3374/Yr.

The application area for variable speed pressure booster pumps would appear to be for those systems which have a high order street pressure variation in combination with a relatively constant high order flow draw.

6.15 Comments on Pump Energy Conservation

The application area for multiple stages constant speed booster pumps would appear to be for those systems which have <u>either</u> a fairly constant street pressure <u>or</u> substantial time periods of low flow draw. While the multiple stages pump system appears to exhibit energy saving advantages over the variable speed unit when either of the conditions stated above are present. Advantages are compounded when both conditions obtain.

The generalities stated above are based on approximate matching of booster pump head output to actual system booster head needs. For existing systems with overheaded pumps, pump matching can be accomplished by impeller trimming at minor order cost.

It is generally true that prior to impeller trimming, the overheaded variable speed pump will operate at less cost than the overheaded multiple constant speed pump system. The order of cost saving will, of course, be dictated by the order of overheading and the system type. There may, for example, be no appreciable savings if the overheaded system has long periods of no flow draw.

The overheaded variable speed pump will react by operating continuously at low speed. Operating at low speed implied low drive efficiency. A reduction in impeller diameter will increase pump speed and drive efficiency, decreasing operating cost.

The overheaded constant speed multiple pump system will react to overheading by establishing continuous very high order pressures drops (approximately 15 psi) across the PRV valve. Impeller trimming will reduce this pressure waste and save operating cost.

The discussion provides a method for evaluation of operating costs for pressure booster systems, which should be applicable to any building pumping system combination. The reader should be aware that the time - flow draw approximation is only an approximation. It is felt that the time-flow draw information shown is highly overstated in terms of actual flow draw. It is hoped that ASPE, together with ASME and the Bureau of Standards can provide more valid time-flow draw information for various building types.

6.16 References

1. "Designing Domestic Water Booster Systems to Save Energy," by David F. Hanson, Plumbing Engineer, July/Aug. 1981, page 28, edited by permission.

2. "Designing Domestic Water Booster Systems to Save Energy," by David F. Hanson, Plumbing Engineer, pp. 30-33, including Figs. 7-9, July/Aug. 1981. Edited by permission.

3. "Problems in Domestic Water Pressure Boosting," by Gilbert F. Carlson, Director Technical Servies, ITT Fluid Handling Division, Skokie, IL.
The ASPE Fourth Biennial Convention, pages 1-44, Figs. 1-19, Nov. 1974, edited by permission.

CHAPTER 7

PUMP OPERATION, MAINTENANCE AND TROUBLESHOOTING

7.1 Inspection After Shipment

After a pump is received from shipment, it should be inspected to determine if any damage occured during transportation. The receiving party should check for such items as broken or cracked castings; bent or broken seal by-pass lines; damaged motor and coupling; loose nuts and bolts.

Most pump manufacturers cover the pump inlet and outlet ports before shipment. A pump received with exposed inlet and outlet ports should be checked inside the pump casing to insure against its containing trash, dirt or other material picked up during shipment.

If the pump is not installed immediately, the ports should be covered before storage. Unpainted surfaces of iron pumps should be oil-coated to prevent rusting.

While checking for physical damage to the pump, instructions on proper pump installation, operation and maintenance should be located. Most pump manufacturers attach this information to their product.

This information often is lost in shipping, or discarded at the job-site. If the receiving inspector does not find the instructions, the pump manufacturer or representative should be contacted.

The instructions should be followed carefully to avoid pump damage, operating troubles or possible voiding of warranty.

7.2 Mounting the Pump

A pump should be located as near as possible to the source of liquid supply. This allows for the shortest run of inlet pipe to the pump and results in the best possible pump inlet conditions.

The pump's location should be dry and well-ventilated. A wet location is dangerous to service personnel and damaging to pumping equipment.

Without adequate ventilation, motor temperature may rise enough to cause thermal protection devices to shut off power to the motor, or possibly burn it up. The location should also provide adequate accessibility for servicing.

The pump should be mounted to a flat foundation; clear of irregularities; and substantial enough to absorb vibration and support the equipment. Providing a separate foundation or pump pad is the best way for mounting a pump unit.

The pad can be cast in concrete with embedded mounting bolts properly sized and located for holding down the pump base plate. In addition, the base plate should be grouted to the foundation pad to minimize pump vibration and noise.

7.3 Embedded Bolts

When mounting close-coupled pumps (Figure 7-1), mounting bolts should not be cast in the foundation. If they are, the unit cannot be serviced without breaking pipe joints because the motor cannot be moved from the pump casing.

Instead of mounting bolts, anchors should be installed in the concrete pad to receive capscrews. The capscrews can be removed to allow the motor to be moved from the pump casing for servicing.

Unless otherwise advised by the manufacturer, in-line circulator pumps should not be installed with supports to the motor. These pumps are designed to be supported by rigid pipe. Attempts to support these pumps can create coupling misalignment, resulting in coupling breakage, vibration, bearing failure, or other undesirable conditions.

If large pump equipment of appreciable value or high precision is required, it is recommended a pump manufacturer, manufacturer's representative, or pump distributor be employed to install or supervise installation. This will eventually insure the unit is installed properly and the pump user will receive proper operating and maintenance instructions.

Pump inlet piping should be as short as possible with a minimum of fittings. This reduces turbulence and provides the highest inlet pressure, both of which will affect pump performance.

A pump generates a pressure force that pushes liquid through piping systems. Therefore, inlet piping is much more critical than outlet piping. If any fitting is required in the inlet, it should be installed at a minimum length of 10 pipe diameters from the pump inlet. Any turbulence created by the fitting will then be dissipated before reaching the pump inlet.

Pump inlet piping should always include a strainer. This protects the pump from lost bolts, rags, etc. After the pump has been in operation long enough to thoroughly flush the system, the strainer may be removed.

Inlet piping to a horizontal split case pump should never be installed as in Figure 7-2. This pump design almost always employs a double inlet impeller. The inlet pipe configuration in Figure 7-2 would deliver more water to one side of the impeller, causing an uneven hydraulic load and excessive end thrust, resulting in poor pump performance.

PUMP SIDE VIEW

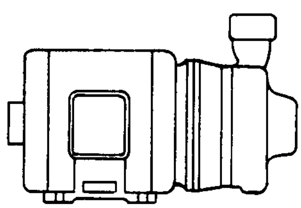


Figure 7-1.

PUMP TOP VIEW

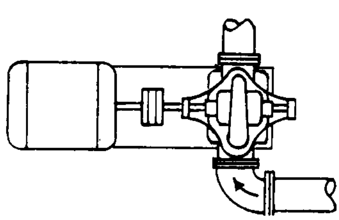


Figure 7-2.

If piped as shown in Figure 7-3, or with a straight section of pipe between the elbow and pump, as in Figure 7-2, the inlet pipe will deliver equal amounts of water to each inlet impeller eye, eliminating the undesirable conditions.

7.4 Turbulence

Concentric pipe reducers should not be used in a horizontal pipe, especially in the inlet pipe to a pump. As shown in Figure 7-4, a reducer of this type will trap air and create turbulence and noise in the piping system. An eccentric reducer installed as shown in Figure 7-5 will not trap air. Care must be taken to install the eccentric reducer with the straight side on top, or it too will trap air.

As shown by the standard piping arrangement in Figure 7-5, a gate valve should be installed on both the inlet and outlet. Then, the pump can be shut off, the gate valves closed and the pump serviced without having to drain the pumping system.

A check valve should also be installed in the pumping system's outlet pipe. This valve protects the pump against water hammer and backflow. Should reverse flow be allowed through a centrifugal pump after operation has stopped, the pump may turbine backward at over twice the normal pumping rmp. Such rotational speed could damage the pump.

Piping to and from a pump must be supported adequately independent from the pump. Pipe flanges should meet pump flanges so the flange bolts can be installed by hand. A pump is not designed to support pipe load.

If flexible pipe connectors are necessary, they should be installed to prevent excessive strain on pump casing. The reason is very important and can be understood by Figure 7-6.

This figure represents a flexible pipe connector between two pipes. The pipes are capped on the ends. If pressure is introduced into the pipe assembly, both caps are pushed and the flexible connector stretched.

The same thing happens if the pipe caps are elbows and the assembly is part of a pump. Consequently, a piping system using flexible connectors must have a rigid pipe support between the pumps and the connector. Flexible connectors always should be installed with rigid pipe support (Figure 7-5).

Solid electrical conduit should not be fastened to the motor or motor conduit box, unless required by electrical wiring codes. Motor vibrations may follow the conduit to where the conduit is fastened to the building, and transmit into the building structure.

To eliminate this possibility, the conduit should be fastened to the motor, or motor conduit box, with a section of flexible conduit, if local electrical codes permit.

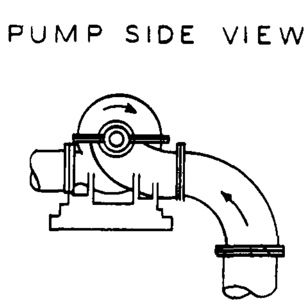


Figure 7-3.

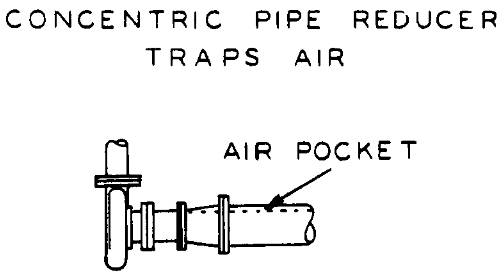


Figure 7-4.

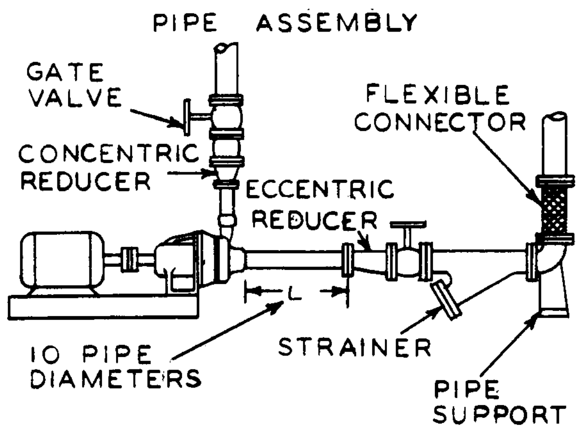


Figure 7-5.

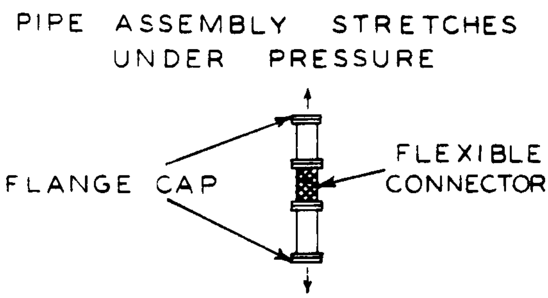


Figure 7-6.

7.5 Alignment

Before operation, every flexible coupled pump should have its coupling alignment checked. A pump manufacturer may align the coupling halves before shipment. But, even if this is done the alignment may be destroyed in transport. Also, when the pump base plate is anchored to the foundation pad, the base plate may distort enough to change alignment.

Jobsite coupling alignment should be done twice. First, it should be checked immediately after the pump base plate is anchored and before the pump motor is energized. After the pump is in service and the system has reached operating temperature, the pump's coupling alignment should be checked again. Pipe expansion and contraction due to pumping hot liquids often changes coupling alignment.

Before operating the pump, the power supply for the motor should be checked to insure the power supply matches the motor's requirements. Normally, power supply voltage will exactly match the motor's nameplate voltage. In practice, however, a 10 percent variation is permissable. Voltage should always be checked at the motor rather than at the power enclosure.

The pre-start-up electrical inspection will insure the electrical inspector that motor protection devices meet electrical codes. If required electrical equipment is properly installed, pumping conditions cannot result in motor burn-out.

If the pump delivers more capacity than anticipated and overload the motor, electrical overloads de-energize the motor before damage occurs. For three-phase motors, these overload devices should be in the form of three-leg magnetic starter protection. Recommended by manufacturers, these protective devices are required by most electric power wiring codes.

When a piping system is not electrically grounded, the pump must be grounded to protect against short circuits and slight voltage leaks. Such leaks can cause pump impellers to deteriorate due to electrolytic or galvanic actions.

7.6 Direction

After electric power is connected to a three-phase motor, the direction of motor rotation should be checked. If the motor is not rotating the impeller in the proper direction, the direction of rotation must be changed. This can be done by interchanging any two of the three wires.

When rotation is checked, the motor must be energized only momentarily. If the pump is not charged with liquid, the seal of a dry pump may burn up even when the pump is run only a few seconds. A mechanical seal operates in similar fashion to a slide bearing, and needs liquid for lubrication.

Before starting the pump, both inlet and outlet valves should be opened to prime the pump with liquid and flush air out of the pump. If the top of the pump casing is fitted with a pipe plug, the plug should be removed long enough to release any entrapped air.

After the pump is charged with liquid, the outlet valve should be closed to about three-fourths of its full turn. This insures start-up of the centrifugal pump at minimal horsepower requirements, while still permitting liquid to flow through the pump. In a new system, this procedure can also be desirable to slowly fill the system with liquid and avoid severe water hammer.

7.7 Open Outlet Valve

After the pumping system is completely charged with liquid, and the pump is operating smoothly, the outlet valve should be opened completely. The pump should be left in operation until the system reaches operating temperature. At this point, the discharge valve should be closed. The motor should be de-energized and the coupling alignment rechecked while the system is still at operating temperature.

The inlet valve should never be closed while the pump is operating. This will create cavitation, which may damage pump equipment. The pump should never be operated with both the inlet and outlet valves closed.

When a centrifugal pump is running without delivering liquid, turbulence in the casing may raise liquid temperature high enough to damage pump seals, gall wearing parts, or even explode the pump casing.

A centrifugal pump may be operated for a short time with the outlet valve closed and the inlet valve open. Because of the open inlet pipe, pressure would not build-up. Temperature, however, will increase gradually in the pump casing. Therefore, the pump should not be allowed to operate for any prolonged period with the outlet valve closed, even if the inlet valve is open.

7.8 Maintenance Functions

The main function of maintenance is to keep the pump properly lubricated to manufacturer specifications. Lubrication should be performed before start-up as well as according to the prescribed time interval during operation. Other specific maintenance instructions should also be followed.

An alert maintenance program will include a log of periodic pump pressure gauge readings. The log should include inlet and outlet gauge readings on initial system start-up and orderly periodic readings during the life span of a pump unit.

If properly used, the log can warn of approaching problems. Should a trend of increasing differential pump pressure be noted, the operator should suspect some restruction being built up in the pumping system. If a trend of decreasing differential pressure is noted, the operator may suspect unusual increased capacity, or pump deterioration. In either case, by watching the log data operator would have time to schedule a convenient shut down to inspect the pump and correct the problem. Without the log data, the operator may be in danger of unsuspected pump failure, pipe rupture, clogged pipe or strainer or other uncontrolled system shut-downs.

If a spare set of pump bearings and shaft seals are not supplied with the pump, the maintenance person should obtain them. Shaft seals and bearings are the parts of a centrifugal pump which may be most easily subjected to damage and wear.

After a seal starts leaking or a bearing fails is no time to initiate purchase of spare parts. Good shelf life of both items may be experienced if kept clean, dry and at a reasonable temperature.

Anytime disassembly of a pump is required, the mechanical seal should be replaced too. The price of the seal is small as compared with other repair costs and unscheduled downtime. Replacing the seal can eliminate the need of future pump repair.

Except for periodic lubrication, most pump maintenance is taken care of if the unit is correctly installed and placed into operation. Proper preparation before operation is the backbone of maintenance. In other words, maintenance attention will not overcome problems resulting from poor installation.

7.9 Major Trouble Categories

Proper pump selection will normally insure satisfactory pump operation. Occasionally, a pumping system does not perform as expected, and determining the specific problem may be difficult.

In general, common pump problems may be classified into the four major categories listed below. Problems not fitting into one of these categories usually need more detailed investigation than suggested here. In such cases, a pump representative or distributor should be consulted. The common pump problem categories are:

- Pump Capacity
- Motor Overload
- Seal Leaks
- Pump Vibration

7.10 Capacity Problems

Centrifugal pump capacity and head are closely related. A pump head too low for system design requirements will not deliver enough capacity. A pump head higher than system design requirements will deliver too much capacity. Consequently, head and capacity problems are both included in the category of capacity problems.

The outline in Figure 7-7 suggests a method of identifying pump capacity problems. Capacity problems can be broken down into the four areas listed below. The terms "head" and "capacity" relate to system design head and capacity.

- Too much capacity with too much head
- Too much capacity with not enough head
- Not enough capacity with too much head
- Not enough capacity with not enough head

Each step of investigation in Figure 7-7 presents two conditions. The condition proven correct will lead to the next step, and identification of the pumping problem.

For clarafication, assume the example of a system with insufficient capacity. First, determine if pump head is too high or too low. If pump head is higher than required, the problem must be in the system.

This situation leads to the next step of conditions. Either the original system head-loss calculations are in error, or there is an unexpected system restriction. If system head-loss calculations prove the original calculations are correct, then there must be some unexpected restriction in the piping system. This unexpected restriction may be a controlled restriction such as a partially closed gate valve, or it could be a clogged pipe or strainer, an air trap, etc.

Any of the four capacity problems may be solved by following this step-by-step procedure. Simply determining pump head and capacity conditions can immediately reveal if the problem is in the system or the pump. This determination may not solve the problem immediately, but confusion about where to search for identification of the problem may be removed.

7.11 Motor Overload

In cases of motor overload, a natural reaction is to accuse the pump of requiring excessive horsepower. In practice, however, the most common causes of motor overload are:

- Voltage not within 10 percent of nameplate rating
- Motor wired to wrong power supply
- Wrong motor overload fuses for protectors
- Unbalanced three phase power

Figure 7-8 is an outline to assist in identifying motor overload problems. It presents the same step-by-step procedure as Figure 7-7. Since power supply problems are the most probable causes of motor overload, that portion of the outline should be checked first. If all power supply conditions are satisfactory, the pump unit must then be investigated.

When the pump problem is hydraulic, the pump capacity is usually higher than system design requirements. Figure 7-7 should be used to identify the specific problem.

With high-head, low-capacity pumps, the problem may be caused by too large an impeller diameter. This situation produces

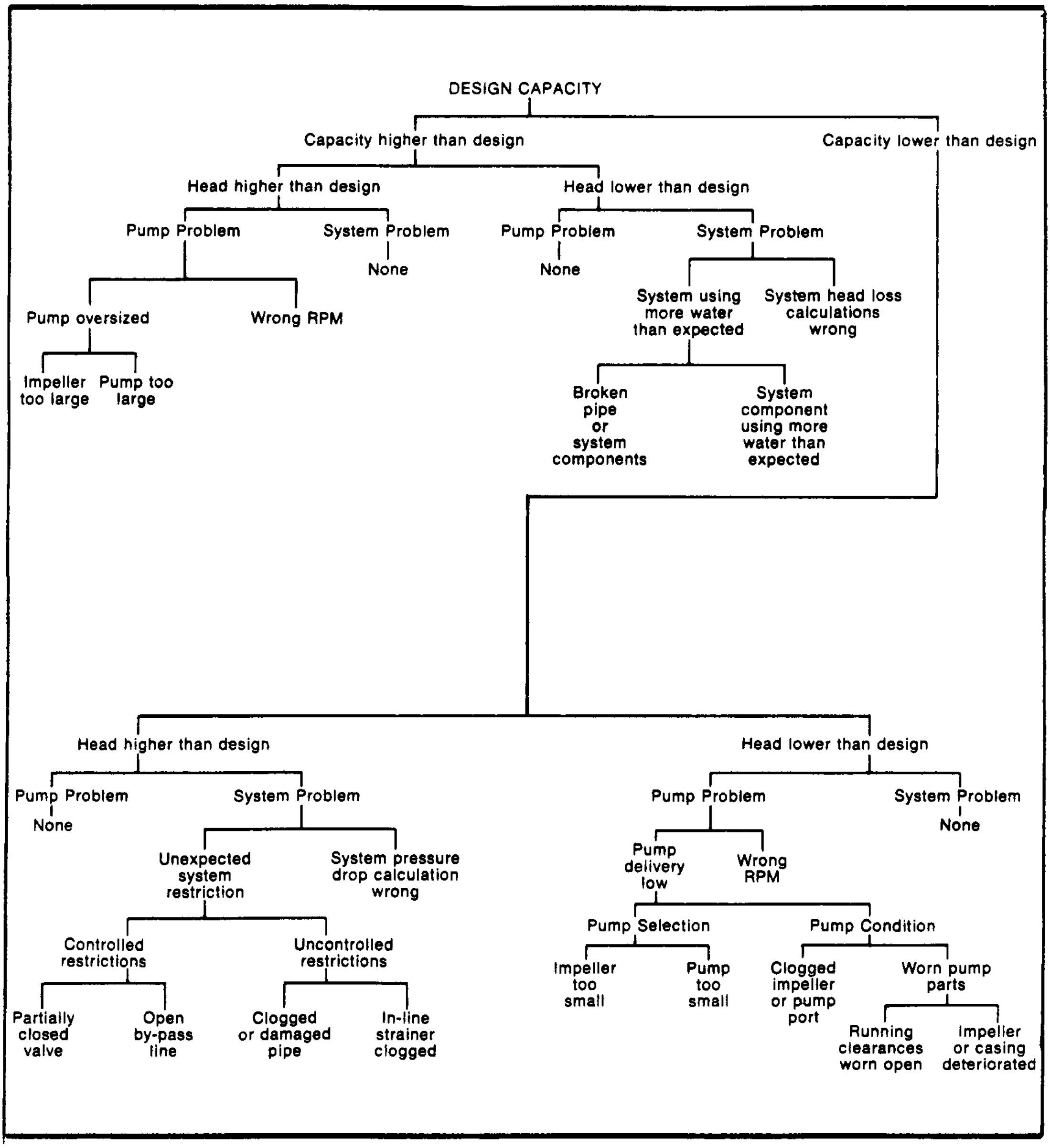


Figure 7-6. A Method of Identifying Pump Capacity Problems.

a higher head than required. But, the slight change in capacity may not be detected readily.

Motor mechanical problems are usually easy to check. First, the motor size should be checked against the pump performance curve to determine if the motor horsepower is correct. If satisfactory, the temperature of the pump room should be checked under operating conditions. If ambient temperature is excessive, the motor may not be able to dissipate heat properly, and can "trip out" electrically because of overheating.

Binding of rotating parts can be discovered by rotating the pump shaft manually when the unit is shut off. Of course, shaft deflection may occur during operation and cause rubbing between the impeller and casing. This problem may not be apparent when making the manual check.

In cases of pump binding, it may be necessary to seek help from an authorized pump representative or distributor. If desired, other items may be added to the outline in Figure 7-8.

Installing proper three-leg temperature compensated protection to three-phase motor installation cannot be overstressed. If motor protection is installed properly, as required by most wiring codes, motor burnout cannot be caused by pumping conditions.

7.12 Seal Leaks

The most common causes of seal leaks are:

- Abrasive material in pumped liquid
- Liquid absence between seal surfaces
- Cracked carbon or seal seat
- Seal installed improperly

If seal leakage is caused by abrasive materials in the liquid being pumped, grooves will be cut into the seal seat or carbon washer, causing them to resemble a phonograph record. If abrasive particles become imbedded in the soft carbon washer, the rotating action of the washer will cut into the hard seat, similar to a grinding wheel action. In this situation, the seat will be grooved, but carbon grooving is questionable.

If abrasive particles are not imbedded in the carbon washer, the washer will be grooved and wear away rapidly. In either situation, steps must be taken to remove abrasive material from the seal cavity by straining, filtering, etc.

7.13 Liquid Absence Between Seal Faces

A mechanical seal serves two functions. First, it acts as a check valve to prevent liquid leaking from, or air leaking into, the pump. Secondly, it acts as a bearing slider. This second function is necessary since relative motion exists between the rotating carbon washer and the stationary seat. Any slider bearing must be lubricated. The lubricant for the mechanical seal of a centrifugal pump is the liquid being pumped.

The liquid in contact with the seal infiltrates between the rotating carbon and stationary seat. This infiltration forms a thin liquid film between the carbon and the seal. Should the seal area be deprived of liquid, even for a few seconds of operation, the liquid film disappears and the carbon comes into rubbing contact with the seat. Friction will bring rapid temperature increase, causing seal surface cracking, carbon deterioration, or both. Such failure may result from operating without liquid, or by "boiling" liquid in the seal cavity.

Mechanical seal installation should be done with great care. The rotation carbon is easily scratched, cracked or chipped by a slight "bump." If the stationary seal is ceramic, the same damage can occur. Unfortunately, the pump may be placed in ser-

Figure 7-8. Motor or Electrical Overload

- Power Supply
  - Wrong Voltage
    - Voltage not within 10% of nameplate rating
    - Unbalanced 3 phase power
  - Improper Wiring
    - Motor wired to wrong power supply
    - Wrong Fuses or overload protection
- Pump Unit
  - Hydraulic
    - Pump Delivering Too Much Capacity (System Loss Too Low)
    - Impeller Diameter Too Large
  - Mechanical
    - Motor
      - Too Small
      - Pump Room Ambient Temperature Too High
    - Binding Rotating Parts

vice before such damage is detected. The stationary seat must be installed carefully. If the seat is not properly aligned, leakage will occur since the seat face and the rotating washer face are not pressed together evenly.

Persistent mechanical seal leaks also may be due to other factors. Should the above observations fail to reveal the seal leakage problem, assistance should be obtained from the pump representative or distributor.

7.14 Vibration

The most common cause of pump vibration is misaligned couplings or pipe strain. To insure against pump vibration, couplings must be aligned during pump installation prior to starting the pump.

On high temperature pumping applications, temperature increases of pump parts may cause coupling misalignment due to expansion of metal parts. An installer should shut off the pump and align the coupling a second time after the pump has reached operating temperature.

Should pipe be installed to transmit strain to the pump casing, vibration may also result. Such strain might cause binding of rotating parts or undue bearing leading.

7.15 Fire Pump Maintenance and Operation

A fire pump must be depended upon in an emergency. Therefore, it must be properly operated and maintained at all times. It is desirable to have someone operating the pump and its driver at the building site. A short test by the regularly scheduled operators should be made each week by discharging water from a convenient outlet.

When a fire alarm is actuated, unless the alarm indicates an automatic fire pump is operating, the person responsible for the fire pump should proceed to its location immediately. The pump should preferably be put in manual operation and allowed to run until the emergency is over, when it can be shut down manually.

During this and every other operating period, all related equipment should be carefully checked to insure that it is performing properly.

To prevent an excess number of starts and stops, an electric motor controller is equipped with a timer which keeps the motor running for at least one minute for each 10 hp motor rating (not more than 7 minutes required). It is usually preferable with most types of pump drivers to permit the unit to run until it is manually shut down.

When there is more than one automatic fire pump, a predetermined sequence for control is arranged. Pump control from one or more remote push buttons - which will start, but not stop the pump - should be provided.

Also, if there is deluge valve control of an open discharge device system, the pump may be started by a dropout relay provided with a closed circuit.

The cooling and lubrication of a centrifugal fire pump is important. The pump must never be allowed to run without the pump casing full of water. Close attention should be given to the bearings and stuffing boxes, particularly during the first few minutes of operation to determine that there is no heating and/or need of further adjustment.

When water reaches the water seal, a small leak at the stuffing box glands is often desirable. Suction inlet and discharge outlet pressure gages should be read occasionally to see the inlet in not obstructed by a choked screen or foot valve.

With a vertical shaft turbine-type fire pump, the water level can be observed when a visible supply is provided. If the pump takes suction from a well, water level testing equipment must be used. The ground water level at the pump should be checked regularly and the draw down should be determined during the annual 150 percent capacity test. These tests normally should indicate any important change in the ground water supply.

The direction of the pump's rotation and speed of operation should always be checked.

7.16 Power Supply Maintenance

The source of pump power should also be checked. With an electric motor drive this means a reliable current supply for the motor and its auxiliary equipment. For steam turbine drive it means providing a steam supply to the control valve and removal of condensate from supply piping and/or turbine exhaust. If the pump is driven by a diesel engine, there must be an adequate fuel for each estimated operational hour. Also, batteries must be fully charged.

Starting equipment must be tested periodically and carefully checked. Any evidence of a drop in electric motor voltage or a drop in steam pressure to a turbine should be investigated immediately.

When employing a diesel engine driven crankcase, oil must be replenished or renewed as needed and the oil filter, air draw and automatic battery should be checked. The specified battery gravity should be determined at least once a month.

7.17 INSTALLATION AND OPERATING PROBLEMS LIST

The following outlines suggest procedures of identifying pump installation and operating problems:

1. Suction and discharge piping should be as short and straight as possible to avoid excessive friction losses. Reducers in the piping should be eccentric.

2. Pumps should not support piping.

3. Line sizes should not be smaller than pump nozzles.

4. Suction line size should be larger in diameter than the pump's suction nozzle.

5. High points in suction piping, which can promote formation of air pockets, should be avoided.

6. Pumps should be located below the suction side liquid level, where possible.

7. A non-slam check valve should be installed in the discharge piping to protect against sudden surges and reverse rotation of the impeller.

8. Where the diameter of a discharge line is greater than the diameter of a discharge nozzle, an eccentric increaser, check valve and gate valve (in that order) should be used.

9. A line strainer in the pump suction piping should be provided unless the pump is equipped with a non-clog impeller.

10. Pumps should be checked for correct alignment during instalation.

11. Pump suction lines should be checked for tightness to avoid drawing air into the pump. Entrained air tends to accumulate in the center of the impeller and causes a reduction in developed head and air lock.

12. When pump suction is directly connected to an open shallow tank, baffles should be placed at the suction piping entrance to break up any vortexes. This also causes the incidence of the entrained air entering the pump to air lock.

13. When cavitation is diagnosed as the source of pump noise (characteristically a crackling sound), installation of a throttling valve in the discharge piping (to reduce pump capacity) should materially reduce the problem.

14. Noise may arise from any of the following conditions:

    a. Excessive velocities in the interconnecting piping or from improperly supported piping.

    b. Motor and/or bearing noise in high speed pumps

    c. Poor selection with operating point substantially higher or lower than manufacturer's recommended best efficiency point

    d. Excessive vibration of pump or driver caused by misalignment, bent shaft, loose mounts or unbalanced hydraulic forces acting on impeller

    e. Improper installation and sizing of the piping, which may cause noise transmission to the building

    f. Improper vibration mounting of pumps

15. Reverse rotation of impeller, or impeller installed in reverse direction (though direction of rotation is correct), resulting in substantially reduced developed head and capacity with high power demand than the manufacturer indicates for measure flow rate.

16. Lack of liquid delivery due to lack of prime, insufficient available NPSH, clogged strainer, system total head exceeding pump total head at zero capacity.

17. Loss of pump prime while operating due to loss of suction line liquid seal, air leak or liquid vaporizing in suction line.

18. Excessive pump power due to excessive impeller speed, tight shaft packing, insufficient clearance between impeller and casing, higher than specified liquid density or viscosity, and other contributing causes mentioned above.

7.18 References

1. "Centrifugal Pump Trouble Shooting," (Part II), by John Aymer. Plumbing Engineer, Part IV, May/June 1974, Figs. 1, 2. Edited with permission.

2. "Pump Installation Operation," by John R. Aymer, (Part III), Plumbing Engineer, Mar/Apr. 1974, (including Figs. 1-6.) Edited with permission.

3. "Pump Installation, Operation and Maintenance," by John R. Aymer, Plumbing Engineer, page 11, including Figs. 5 and 6, March/Apr. 1974. Edited by permission.

Appendix A

General Characteristics of Different Pumps

Pump Type	Construction Style	Construction Characteristics	Normal Number of Stages	Maintenance[a]	Solids Tolerance	Notes
			Dynamic Type Pumps			
Centrifugal (horizontal)	Single-stage over-hung, process type	Impeller canti-levered beyond bearings.	1	L	M-H	Capacity varies w/head
	Two stage overhung	Two impellers cantilevered be-yond bearings.	2	L	M-H	Used for heads above single-stage capability
	Single-stage im-peller between bearings	Impeller between bearings, casing radially or axi-ally split.	1	L	M-H	Used for high flows to 1083 ft (330 m) head.
	Chemical	Casting patterns designed with thin sections for high-cost alloys	1	M	M-H	Have low pressure and temperature ratings.
	Slurry	Designed with large flow pas-sages.	1	H	H	Low speed and adjust-able axial clearance. Has custom control features.
	Canned	No stuffing box and motor en-closed in a pres-sure shell.	1	L-M	L	Low head capacity lim-its when used in chem-ical services.
	Multistage, hor-izontally split casing	Nozzles located in bottom half of casing.	Multi	L	M	Have moderate tempera-ture-pressure ranges.

Appendix A (Cont'd.)

General Characteristics of Different Pumps

Pump Type	Construction Style	Construction Characteristics	Normal Number of Stages	Maintenance	Solids Tolerance	Notes
		Dynamic Type Pumps				
	Multistate, barrel type	Outer casing contains inner stack of diaphragms.	Multi	L	M	Used for high temperature-pressure ratings
Centrigugal (vertical)	Single-stage, process type	Vertical orientation.	1	L	M	Used to exploit low net positive section head (NPSH) requirements.
	Multistage	Many stages with low head per stage.	1	L	M	Low-cost installation
	Inline	Inline installation, similar to a valve.	1	L	M	Low-cost installation
	High speed	Speeds to 380 rps heads to 5800 ft (1770 m).	1	M	L	High head/low flow. Moderate costs.
	Sump	Casing immersed in sump for easy priming & instalation.	1	L	M-H	Low cost.
	Multistage, deep well	Long shafts	Multi	M-H	M	Used for water well service.
Axial	Propeller	Propeller-shaped impeller.	1	L	H	Vertical orientation

Appendix A (Cont'd.)

General Characteristics of Different Pumps

Pump Type	Construction Style	Construction Characteristics	Normal Number of Stages	Maintenance	Solids Tolerance	Notes
		Dynamic Type Pumps				
Turbine	Regenerative	Fluted impeller. Flow path resembles screw around periphery.	1,2	H	M	Capacity independent of head. Low flow/ high head performance
		Positive Displacement Pumps				
Reciprocating	Piston, plunger	Slow speeds	1	H	M	Driven by steam engine cylinders or motors through crankcases.
	Metering	Consists of small units with precision flow control system.	1	M-H	L	Diaphragm and packed plunger types.
	Diaphragm	No stuffing box.	1	H	L	Used for chemical slurries. Can be pneumatically or hydraulically actuated
Rotary	Screw	1,2 or 3 screw rotors.	1	M	M	For high-viscosity high-flow-high-pressure services.
	Gear	Intermeshing gear wheels.	1	M	M	For high-viscosity, moderate-pressure/ moderate-flow services.

[a]L = low; M = medium; H = high.

Source: "Fluid Flow: pumps, pipes and channels," by Nicholas P. Cheremisinoff, Table 18. Reprinted by permission.

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