SECTION 8.3
WATERHAMMER

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Waterhammer is a very destructive force that exists in any pumping installation where the rate of flow changes abruptly for various reasons. Most engineers recognize the existence of waterhammer, but few realize its destructive force. Much time and expense have been spent repairing pipelines and pumps damaged by waterhammer. It is thus essential for an engineer to be able to know when to expect waterhammer, how to estimate the possible maximum pressure rise, and, if possible, how to provide means to reduce the maximum pressure rise to a safe limit.

The computational procedures used for the analysis of waterhammer in pump discharge lines with electric-motor-driven pumps have been known for many years, beginning with the basic waterhammer contributions by Joukousky and Allievi. This work was followed in later years by many applications of numeric, graphic, and computer techniques. Although the theory and mechanics of computing waterhammer in pump discharge lines have advanced rapidly in recent years, there are many practical aspects of this subject that are still confusing to engineers. It is the purpose of this section to bring these to the reader's attention. The first and major portion of the section contains a discussion of some practical aspects of waterhammer control devices used in pumping plants; the second indicates the source of various charts that provide ready waterhammer solutions for a variety of these control devices.

NOMENCLATURE

The following is a list of variables commonly used in waterhammer computations. The SI conversion factors for these terms are to be found in Table 1.

a = velocity of pressure wave, ft/s

D = inside diameter of conduit, ft

TABLE 1 SI conversions

To convert	To	Multiply by
GD^2 (kg · m^2)	WR^2 (lb · ft^2)	23.73
kg/m^2	lb/ft^2	0.2048
kg/m^3	lb/ft^3	0.06243
m	ft	3.281
m/s	ft/s	3.281
m/s^2	ft/s^2	3.281
m^3/s	ft^3/s	35.32
mm	ft	3.281×10^{-3}

e = thickness of pipe wall, ft
E = Young's modulus for pipe material, lb/ft^2
g = acceleration of gravity, ft/s^2
H_o = pumping head for initial steady pumping conditions, ft
H_R = rated pumping head, ft
$K_1 = 91{,}600\, H_R Q_R / WR^2{}_{n R} N^2{}_{R,} s^{-1}$
K = volume modulus of liquid, lb/ft^2
L = total length of conduit, ft
$2L/a$ = round-trip wave travel time, s
N_R = rated pump speed, rpm
η_R = pump efficiency at rated speed and head, decimal form
ρ = pipe line constant $= a\overline{V}_o/gH_o$
Q_0 = initial flow through pump, ft^3/s
Q_R = rated flow through pump, ft^3/s
μ = Poisson's ratio of pipe material
V_0 = velocity in conduit for initial steady conditions, ft/s
w = specific weight of water, lb/ft^3
WR^2 = flywheel effect of rotating parts of motor, pump, and entrained water, lb-ft^2

BASIC ASSUMPTIONS

A considerable number of assumptions were made in the derivation of the fundamental water-hammer equations and in the solution of the various hydraulic transients in pumping systems. These assumptions are often overlooked and involve the physical properties of the fluid and pipeline, the kinematics of the flow, and the transient response of the pump as follows:

1. The fluid in the pipe system is elastic, of homogeneous density, and always in the liquid state.
2. The pipe wall material or conduit is homogeneous, isotropic, and elastic.
3. The velocities and pressures in the pipeline, which is always flowing full, are uniformly distributed over any transverse cross-section of the pipe.
4. The velocity head in the pipeline is negligible relative to the pressure changes.

5. At any time during the pump transient, when operation is in the zones of pump operation, energy dissipation, and turbine operation, there is an instantaneous agreement at the pump, as defined by the steady-state complete pump characteristics of the pump speed and torque corresponding to the transient head and flow that exist at that moment at the pump.
6. The length between the inlet and outlet of the pump is so short that waterhammer waves propagate between these two points instantly.
7. Windage effects of the rotating elements of the pump and motor during the transients are negligible.
8. Water levels at the intake and discharge reservoirs do not change during the transient period.

FACTORS AFFECTING WATERHAMMER

High- and Low-Head Pumping Systems Waterhammer is of greater significance in low-head pumping systems than in high-head systems. The normal steady water velocities in high-head and low-head pumping systems are usually of about the same order of magnitude. However, the pressure changes are proportional to the rate of change in the velocity of the water in the line. For a given rate of velocity change, however, the pressure changes in the high- and low-head pumping systems are of about the same order of magnitude. Therefore, a given head rise would be a larger proportion of the pumping head in a low-head pumping system than a high-head system.

Discharge Line Profile The pump discharge line profile is usually based on economic, topographic, and land right-of-way considerations. However, in selecting the alignment along which a pump discharge line is to be located, there are other considerations that often make one pipeline profile and alignment more favorable than another. For example, upon a power failure at the pump motors, the envelope of the maximum downsurge gradient along the length of the pipeline is a concave curve. Therefore it may be possible to avoid the use of expensive pressure control devices at a pumping plant if the pipeline profile is also concave and is not located above the downsurge gradient curve. In some cases, it may even be economical to lower the profile of the discharge line at the critical locations by deeper excavation. If a surge tank at the pumping plant is definitely required, the most favorable pipeline profile is one with high ground near the pumping plant where the surge tank structure can be placed so its height above the natural ground line will be much less than what would be necessary if there were no high ground near the plant.

Rigid Water Column Theory The question is often raised as to whether the rigid water column theory is sufficiently accurate for the computation of waterhammer in pump discharge lines. In the rigid water column theory, the water is assumed to be incompressible and the pipe walls rigid. In the author's experience, the accuracy and limitations of the rigid water column theory are often questionable for most waterhammer problems that occur in pump discharge lines.

Waterhammer Wave Velocity From a practical viewpoint, a difference of 15 to 20% in the magnitude of the computed waterhammer wave velocity usually has very little effect on the waterhammer in pump discharge lines. The effect on the waterhammer due to a possible error in the wave velocity can be verified by first computing the wave velocity as accurately as possible and then recomputing the transients for the critical cases with a wave velocity about 20% higher or lower. At installations where alternative materials for the pipeline are being investigated, one waterhammer wave velocity and solution for waterhammer for either alternative will usually suffice regardless of the pipe material finally selected.

Pipeline Size The diameter of the pipeline is usually determined from economic consideration based on steady-state pumping conditions. However, the waterhammer effects in a pump discharge line can be reduced by increasing the size of the discharge line because the velocity changes in the larger pipeline will be less. This is usually an expensive method of reducing waterhammer in pump discharge lines, but there are sometimes occasions where an increase in pipe size may be justified to avoid the use of more expensive waterhammer control devices.

Number of Pumps The number of pumps connected to each pump discharge line is usually determined from the operational requirements of the installation, availability of pumps, and other economic considerations. However, the number and size of pumps connected to each discharge line have some effect on the waterhammer transients. For pump start-up with pumps equipped with check valves, the greater the number of pumps on each discharge line, the smaller the pressure rise. Moreover, if there is a malfunction at one of the pumps or check valves, a multiple pump installation on each discharge line would be preferable to a single pump installation because the flow changes in the discharge line due to such a malfunction would be less with multiple pumps. When a simultaneous power failure occurs at all of the pump motors, the fewer the number of pumps on a discharge line, the smaller the pressure changes and other hydraulic transients. For a given total flow in the discharge line, a large number of small pumps and motors will have considerably less total kinetic energy in the rotating parts to sustain the flow than a small number of pumps. Consequently, for the same total flow, the velocity changes and waterhammer effects due to a power failure are a minimum when there is only one pump connected to each discharge line.

Flywheel Effect (WR^2) Another method for reducing the waterhammer effects in pump discharge lines is to provide additional flywheel effect (WR^2) in the rotating element of the motor. As an average, the motor usually provides about 90% of the combined flywheel effect of the rotating elements of the pump and motor. Upon a power failure at the motor, an increase in the kinetic energy of the rotating parts will reduce the rate of change in the flow of water in the discharge line. In most cases, an increase of 100% in the WR^2 of large motors can usually be obtained at an increased cost of about 20% of the original cost of the motor. Ordinarily, an increase in WR^2 is not an economical method for reducing waterhammer, but it is possible in some marginal cases to eliminate other, more expensive pressure control devices.

Specific Speed of Pumps For a given pipeline and initial steady-flow conditions, the maximum head rise that can occur in a discharge line subsequent to a power failure where the reverse flow passes through the pump depends first on the magnitude of the maximum reverse flow that can pass through the pump during the energy-dissipation and turbine-operation zones. Secondarily, it depends on the flow that can pass through the pump at runaway speed in reverse. Upon a power failure, the radial-flow (low-specific-speed) pump will produce slightly more downsurge than the axial-flow (high-specific-speed) and mixed-flow pumps.[1] The radial-flow pump will also produce the highest head rise upon a power failure if the reverse flow is permitted to pass through the pump. There is usually very little head rise at mixed-flow and axial-flow pumps when a power failure occurs and if a water column separation does not occur at some other location in the line.

During a power failure with no valves, the highest reverse speed is reached by the axial-flow pump and the lowest by the radial-flow pump. Care must therefore be taken to prevent damage to the motors with the higher-specific-speed pumps because of these higher reverse speeds. Upon pump start-up against an initially closed check valve, the axial-flow pump will produce the highest head rise in the discharge line because it also has the highest shutoff head. On pump start-up, a radial-flow pump will produce a nominal head rise, but an axial-flow pump can produce a head rise of several times the static head.

Complete Pump Characteristics In order to determine the transient conditions due to a power failure at the pump motors, the waterhammer wave phenomena in the pipeline, the rotating inertia of the pump and motor, and the complete pump characteristics as well as other boundary conditions and head losses must be known. In the solution of waterhammer problems with computers, the complete pump characteristics are sometimes approximated by polynomial expressions in which the coefficients of the polynomial are obtained by fitting a representative curve through several points at specific locations on the pump characteristics diagram. Pump manufacturers sometimes provide limited information to determine such coefficients. However, a comparison between the polynomial values and the complete pump characteristics diagram indicates serious discrepancies in some cases, especially in the zone of energy dissipation. To ensure that a serious error does not result in the computation of the hydraulic transients, care must be exercised in the use of an approximate polynomial expression as a substitute for the correct complete pump characteristics.

Complex Piping Systems As noted previously in the basic assumptions, the waterhammer theory is strictly applicable for a pipeline of uniform characteristics. However, for waterhammer purposes, a complex piping system can be reduced to a satisfactory equivalent uniform pipe system. The approximations are made by neglecting the wave transmission effects at the junctions and points of discontinuity and by utilizing the rigid water column theory. The pertinent water-hammer equations are then found to be analogous to those used in electric circuits. In practice, the waterhammer analysis with these approximations will usually give more conservative results than those obtained experimentally from the pipe system.[2]

Available Waterhammer Solutions The waterhammer solutions for pumping systems with various surge control devices are given in convenient chart form in the references. These include the following:

1. Hydraulic transients at the pump and midlength of the pump discharge lines for radial-flow, mixed-flow, and axial-flow pumps with reverse flow passing through the pumps
2. Surge tanks
3. Air chambers
4. Surge suppressors
5. One-way surge tanks
6. Water column separation

The operation of these surge control devices is described next.

Power Failure at Pump Motors

PUMPS WITH NO VALVES AT THE PUMP When the power supply to the pump motors is suddenly cut off, the only energy that is left to drive the pump in the forward direction is the kinetic energy of the rotating elements of the pump and motor. Because this energy is usually relatively small relative to that required to maintain the flow of water against the discharge head, the reduction in the pump speed is very rapid. As the pump speed is reduced, the flow of water in the discharge line is also reduced. As a result of these rapid flow changes, waterhammer waves of increasing subnormal pressure are formed in the discharge line at the pump. These subnormal pressure waves move rapidly up the discharge line to the discharge outlet, where complete wave reflections occur. Soon the speed of the pump is reduced to a point where no water can be delivered against the existing head. If there is no control valve at the pump, the flow through the pump reverses even though the pump may still be rotating in the forward direction. The speed of the pump now drops more rapidly and passes through zero speed. Soon the maximum reverse flow passes through the pump. A short time later the pump, acting as a turbine, reaches runaway speed in reverse. As the pump approaches runaway speed, the reverse flow through the pump is reduced. For radial-flow pumps, this rapid reduction in reverse flow produces a pressure rise at the pump and along the length of the discharge line. The results of a large number of waterhammer solutions for a given set of radial-flow (low-specific-speed)

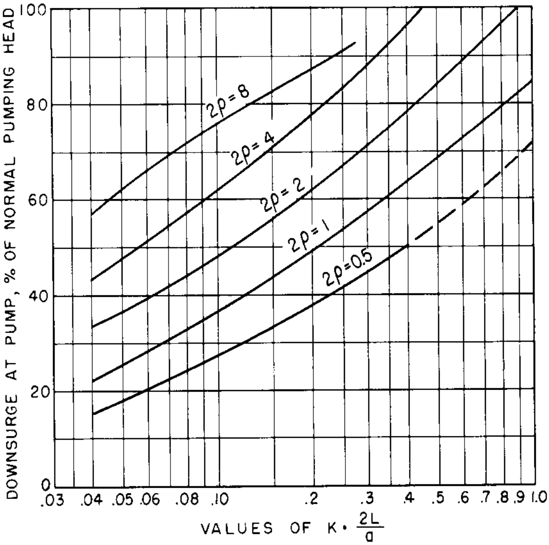


FIGURE 1 Downsurge at pump

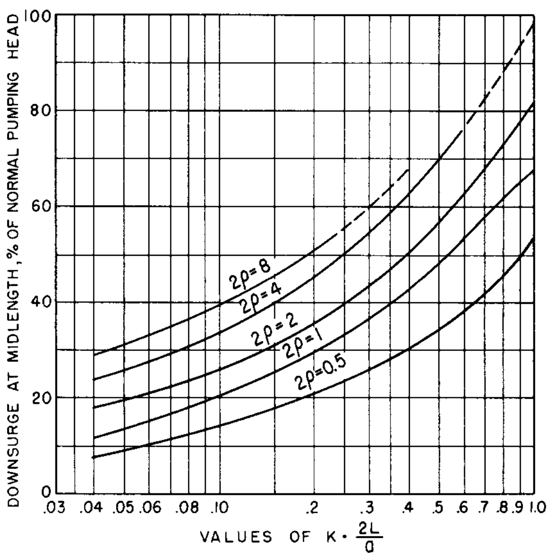


FIGURE 2 Downsurge at midlength

pump characteristics are given in chart form in Figures 1 to 8. These charts furnish a convenient method for obtaining the hydraulic transients at the pump and midlength of the discharge line when no control valves are present at the pump. Although the charts are theoretically applicable to one particular set of radial-flow pump characteristics, they are

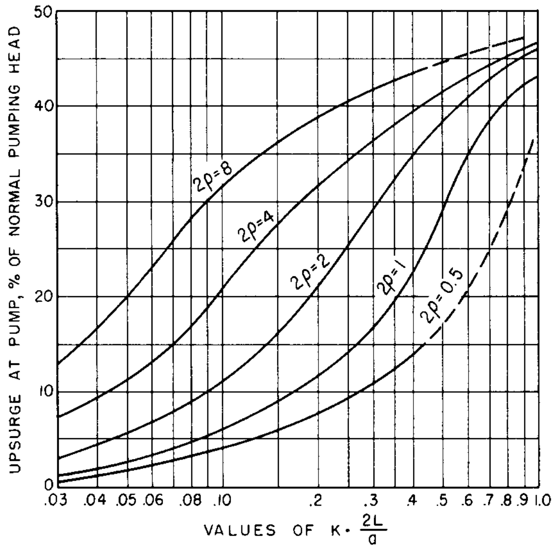


FIGURE 3 Upsurge at pump

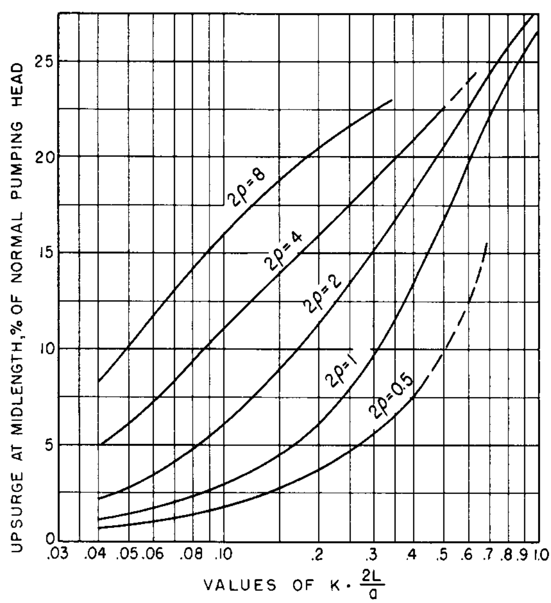


FIGURE 4 Upsurge at midlength

useful for estimating the waterhammer effects in any pump discharge line equipped with radial-flow pumps. The charts are based on two independent parameters: ρ, the pipeline constant, and $K(2L/a)$, a constant that includes the effect of pump and motor inertia and

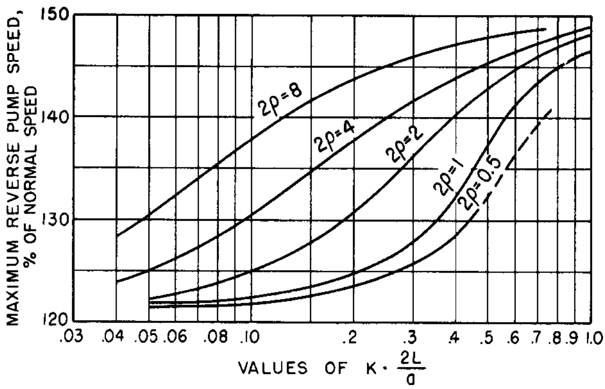


FIGURE 5 Maximum reverse speed

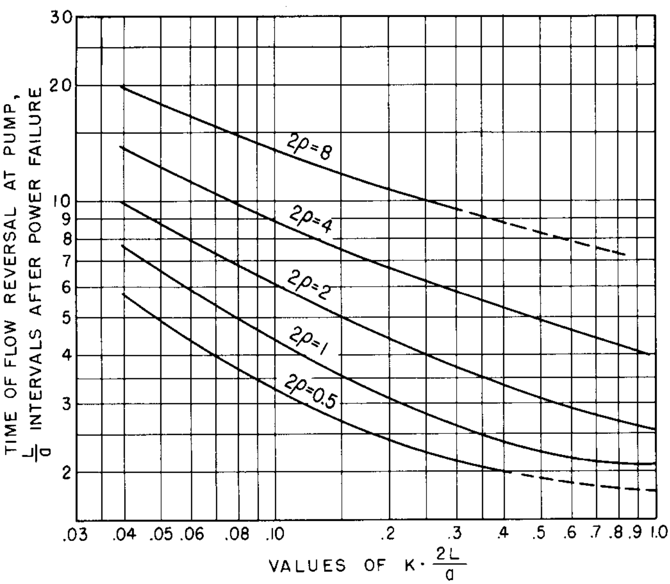


FIGURE 6 Time of flow reversal at pump

the waterhammer wave travel time of the discharge line. If the frictional head in the discharge line during normal operation is more than 25% of the total pumping head and if water column separation does not occur at any point in the line, the maximum head at the pump with reverse flow passing through the pumps will usually not exceed the initial pumping head.[3]

PUMPS EQUIPPED WITH CHECK VALVES There are a number of problems associated with the use of check valves in pump discharge lines. Under steady flow conditions, the pump discharge keeps the check valve open. However, when the flow through the pump reverses subsequent to a power failure, the check valve closes very rapidly under the action of the

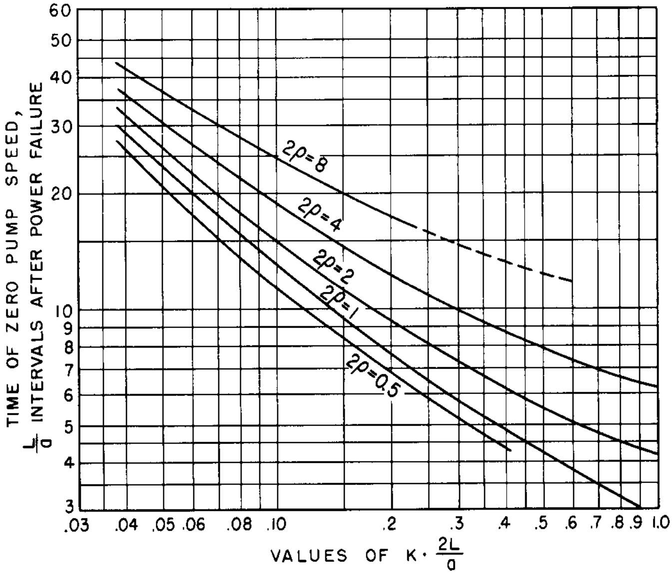


FIGURE 7 Time of zero pump speed

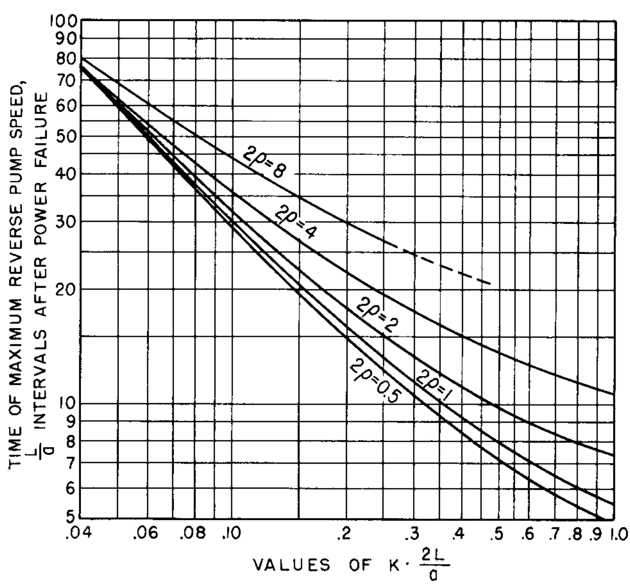


FIGURE 8 Time of maximum reverse pump speed

reverse flow and the resulting dynamic forces on the check valve disk. Under these conditions, neglecting pipeline friction, the head rise in the discharge line at the check valve is about equal to the head drop that existed at the moment of flow reversal. However, if the check valve closure upon flow reversal is momentarily delayed because of hinge friction, malfunction, or the inertial characteristics of the valve, the maximum head rise in the discharge line at the check valve can be considerably higher. On the other hand, if the check valve closure can be accomplished slightly in advance of flow reversal, the head rise in the pump discharge line at the valve is even lower than that obtained with a check valve that closes at the moment of flow reversal. This feature is utilized by a number of check valve manufacturers, who provide spring-loaded or lever-arm-weighted devices on the check valve hinge pins to assist in closing the valve disk before the flow reverses. With these devices, the hydraulic forces on the valve disk under normal flow conditions must be sufficient to overcome the spring or lever-arm-weight forces in order to keep the check valve disk wide open so the head losses at the valve under steady flow conditions will be a minimum.

Check valves in pump discharge lines may be grouped into two general classes, rapid-closing and slow-closing. From the previous considerations noted, the primary requirement for a check valve upon a power failure is that it should close quickly, before a substantial reverse flow has been established. When this primary requirement for a fast-closing check valve cannot be met because of the flow characteristics of the system and the design of the check valve, an alternative is to provide a device such as a dashpot, which will slow down or cushion the last portion of the check valve closure. This feature has been utilized by a number of check valve manufacturers.

CONTROLLED VALVE CLOSURE At most large pumping plant installations, the use of a single-speed discharge valve closure subsequent to a power failure will usually limit the head rise in the discharge line to an acceptable value. However, it will be found that with the optimum single-speed closure, some reverse rotation below the maximum runaway speed of the unit in reverse will occur. If it is desired from other considerations to prevent or to limit the reverse speed of the unit, a two-speed valve closure can be used. In such cases, the discharge valve should close the major portion of its stroke very rapidly up to the moment that the flow reverses at the pump. It should then complete the remainder of its stroke at a lower rate in order to limit the pressure rise in the discharge line to an acceptable valve. At pumping plants where there is more than one pump on the same discharge line, a compromise must be obtained on the optimum single-speed and two-speed closure rates for the various combination of pumps that might be in operation at the time of a power failure.

SURGE SUPPRESSORS Surge suppressors are sometimes used in pumping plants to control the pressure rise that occurs in pump discharge lines subsequent to power interruptions. A typical surge suppressor consists of a pilot-operated valve that opens quickly after a power interruption either through the loss of power to a solenoid or by a sudden large pressure reduction or pressure increase at the surge suppressor. This valve provides an opening for releasing water from the pump discharge line. The valve is later closed at a lower rate by the action of a dashpot to control the pressure rise as the flow of water is shut off. A properly sized and field-adjusted surge suppressor can reduce the pressure rise in the discharge line to any desired value, provided that water column separation does not occur at other locations in the discharge line. The charts given in Reference 4 can be used to determine the required flow capacity of the surge suppressor.

The proper field adjustment of a surge suppressor is very important. If the suppressor opens too rapidly subsequent to a power failure, the downsurge at the pump and along the discharge line profile will be more than if no surge suppressor is present. As a result, a water column separation may be produced at some locations in the discharge line by the premature opening of the surge suppressor. If the suppressor closes too rapidly after the maximum reverse flow has been established, a large pressure rise will occur.

WATER COLUMN SEPARATION Water column separation in a pump discharge line subsequent to a power failure at the pump motors occurs whenever the momentary hydraulic gradient at any location reduces the pressure in the discharge line to the vapor pressure of water. Whenever this condition occurs, the normal waterhammer solution is no longer valid. If the subatmospheric pressure condition inside the pipe persists for a sufficient period, the water in the discharge line parts and is separated by a section of water and vapor. Whenever possible, water column separation should be avoided because of the potentially high pressure rise that often results when the two water columns rejoin. An approximate waterhammer solution for water column separation in pump discharge lines is given in Reference 5.

QUICK-OPENING, SLOW-CLOSING VALVES A quick-opening, dashpot-controlled, slow-closing valve can be used to limit the pressure rise at the high points in the discharge line, where water column separation frequently occurs. When the pressure in the pipeline at the point of water column separation drops below a predetermined value for which the valve is set, the valve opens quickly and a small amount of air is admitted in the pipeline. After the upper water column in the pipeline stops, reverses, and returns to the point of separation near the valve, the valve should be wide open. The air and water mixes and then the clear water discharges through the valve. The open valve then provides a point of relief to reduce the pressure rise caused by the rejoining of the water columns. The valve is later closed slowly under the action of a dashpot so the head rise that occurs in the discharge line at the valve location when the reverse flow is shut off is not objectionable. Whenever these valves are used, precautions should be taken to ensure that they are properly sized, field-adjusted to the proper opening and closing times, and adequately protected against freezing.

One-Way Surge Tanks The one-way surge tank, which was introduced by the writer[6] is an effective and economical pressure control device for use at locations where water column separation occurs. A one-way surge tank is a relatively small tank filled with water to a level far below the hydraulic gradient. It is connected to the main pipeline with check valves that are held closed by the discharge line pressure. Upon a power failure, when the pressure in the discharge line at the one-way surge tank drops below the head corresponding to the water level in the tank, the check valve opens quickly and the tank starts to drain, filling the void formed by the separation of the water columns. When the flow in the upper column starts to reverse, the check valves at the one-way tank close before any appreciable reverse flow is established in the discharge line. Thus there is no pressure rise when the water columns rejoin. The initial level of water in the one-way surge tank is usually maintained automatically with float control or altitude valves. It should be noted that the one-way surge tank does not act during the start-up cycle of the pump discharge line and that it must also be protected against freezing.

AIR CHAMBERS An effective device for controlling the pressure surges in a long pump discharge line is a hydropneumatic tank or air chamber. The air chamber is usually located at or near the pumping plant. It can be of any desired configuration and may be placed in a vertical, horizontal, or sloping position. The lower portion of the chamber contains water and the upper portion contains compressed air. The desired air and water levels are maintained with float level controls and an air compressor. When the power failure occurs at the pump motor, the head and flow developed by the pump decrease rapidly. The compressed air in the air chamber then expands and forces water out of the bottom of the chamber into the discharge line, thus minimizing the velocity changes and waterhammer effects in the line. When the pump speed is reduced to the point where the pump cannot deliver water against the existing head, which is usually a fraction of a second after a power failure, the check valve at the discharge side of the pump closes rapidly and the pump then slows down to a stop. A short time later, the water in the discharge line slows down to a stop, reverses, and flows back into the air chamber. As the reverse flow enters the chamber, usually through a throttling orifice, the air volume in the chamber decreases and a head rise above the pumping head occurs in the discharge line. The magnitude of this head rise depends on the throttling orifice and on the initial volume of air in the air chamber.

The results of a large number of graphic waterhammer-air chamber solutions are given in Reference 7. Another presentation of air chamber charts using the rigid water column theory is given in Reference 8.

SURGE TANKS Because it has no moving parts that can malfunction, a surge tank is one of the most dependable devices that can be used at a pumping plant to reduce waterhammer resulting from rapid changes of flow in the discharge line subsequent to a power failure at the pump motor. Following a power failure, the water in the surge tank provides a nearby source of potential energy that will effectively reduce the rate of change of flow and the waterhammer in the discharge line. The charts given in Reference 7 provide a ready means for calculating the surges in the pipeline due to the sudden starting or stopping of a pump.

One of the disadvantages of a conventional surge tank is that because the top of the tank must extend above the normal hydraulic gradient to avoid spilling, the tank must be quite tall and expensive at high-head pumping installations. In order to obtain the most economical surge tank design, care should be given to the proper sizing of the throttling device at the base of the tank.

NONREVERSE RATCHETS Another device occasionally used for reducing waterhammer in a pump discharge line upon a power failure is a nonreverse ratchet on the pump and motor shaft, which prevents the reverse rotation of the pump. This device is effective for controlling waterhammer when there is a power failure because of the large reverse flow that can pass through the stationary impeller. Except on small pumps, experience to date with nonreverse ratchet mechanisms has been very disappointing. At a number of moderate-size pump installations where these devices were used, the shock to the pump and motor shaft system caused by the sudden shaft stoppage created other serious mechanical difficulties.

Automatic Restart of Motors At small unattended pumping plants, it is often desirable after a power failure to automatically return the pumps to service as soon as the power is restored. However, it was found that occasionally, after a very short power outage, an induction motor could restart and come quickly up to forward speed while a reverse flow was still passing through the pump. Under these conditions, waterhammer in the discharge line is very objectionable. If the pump motor has the capability of restarting under such transient conditions, a time delay or similar device should be installed at the motor controls so the pump can be started only when it is safe to do so.

Normal Pump Start-Up

WITH CONTROLLED VALVE OPENING At some pumping plants, the pump is brought up to speed against a closed valve on the discharge side of the pump. The valve is then opened slowly, and there is very little waterhammer in the discharge line. However, it will be found that nearly all of the pump flow in the discharge line is established with only a relatively small valve opening because the head losses across the valve decreases very rapidly during the opening stroke. For long discharge lines, the head loss and flow characteristics of the valve during the opening stroke must be considered in determining the optimum rate of opening.

WITH CHECK VALVES At pumping plants where the pipeline is held full with pump check valves, waterhammer in the discharge line due to a pump start-up can be objectionable in some cases. If the motor comes up to speed very rapidly, the pump will develop a pressure rise in the discharge line as the sudden increase in flow moves into the line. As noted previously and in Reference 1, this pressure rise is lower for radial-flow (low-specific-speed) pumps than for axial-flow (high-specific-speed) pumps (see Section 8.1).

WITH CASING UNWATERED At pumping plants equipped with large pumps, normal starting of a pump is often performed with the pump casing unwatered. This is accomplished by depressing the water level below the pump impeller by means of compressed air, which is admitted into the pump casing with the pump discharge valve closed and the discharge line full. After the motor has been synchronized on the line, the compressed air in the pump casing is released, allowing water to re-enter the pump from the suction elbow, after which the discharge valve is slowly opened. This type of operation has been satisfactory with most large pumping units, and there are normally no significant waterhammer effects on the discharge line. However, there have been some difficulties with this type of operation at a few large pump units. In the latter case, when the rising water level in the suction elbow first reaches the pump impeller, a very fast pumping action occurs within a few seconds and a severe uplift of the pump and motor from the thrust bearing could occur at this moment. If the discharge valve is still closed when this fast pumping action occurs, there is no waterhammer effect in the discharge line.

WITH SURGE OR AIR CHAMBERS With a surge tank or air chamber at the pumping plant, it makes very little difference whether the increased pump flow is sudden or gradual, inasmuch as the major portion of the sudden increased flow will enter the surge tank or air chamber. With these devices, the steep front of the pressure rise at the pump is transformed into a smaller pressure rise in the discharge line and a subsequent slow oscillating movement in the surge tank or air chamber.

NORMAL PUMP SHUTDOWN The pumping installation that produces the least waterhammer effect in a pump discharge line during a normal pump shutdown is one in which the control valve on the discharge side of the pump is first closed and the power to the pump motor then is shut off. If only check valves are in operation on the discharge side of the pumps and the power to one of several pumps motors connected to the same discharge line is cut off, the flow at the pump that has been shut down will reverse rapidly and the check valve will close rapidly. The use of antislam or slow-closing features at the check valves will reduce the waterhammer effect in the discharge line.

CONCLUSIONS

A variety of waterhammer control devices for pumping plants are available to the designer. In most cases, the experienced designer can narrow the choice of the most suitable device to a few practical alternatives. A prior knowledge of the available waterhammer solution for these devices will reduce the amount of detailed computational work that must be done to determine the critical hydraulic transient effects.

EXAMPLE Consider a power failure at the pumping plant installation shown in Figure 9. This installation consists of three pumps that discharge into a steel pipeline. Aside from isolation valves, there are no check valves in the system. The basic data for this installation are as follows:

$$D = 32 \text{ in}; e = 3/16 \text{ in}; Q_0 = Q_R = 33.7 \text{ ft}^3/\text{s (for three pumps)}$$
$$V_0 = 6.03 \text{ ft/s (for three pumps)}; H_0 = H_R = 220 \text{ ft}$$

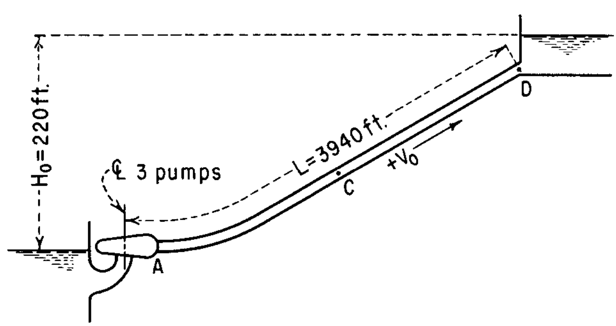
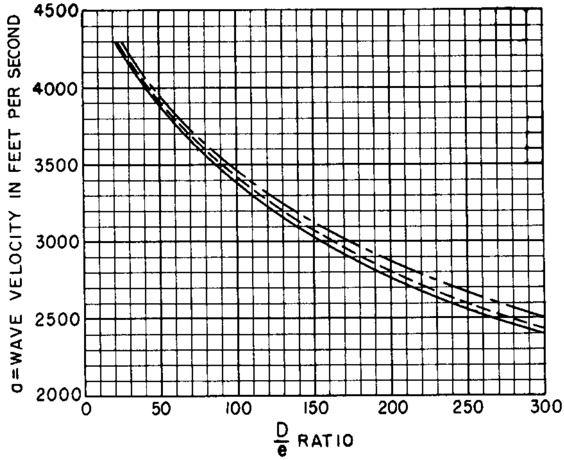


FIGURE 9 Pipeline profile



$$a = \sqrt{\frac{1}{\frac{w}{g}\left(\frac{1}{K} + \frac{Dc_1}{Ee}\right)}}$$

where:

a = wave velocity (ft. per sec.)

g = acceleration of gravity (ft. per sec²)

$\frac{D}{e} = \frac{\text{diameter of pipe}}{\text{thickness of pipe}}$

E = Young's modulus for steel pipes $= 4.32 \times 10^9$ (lb. per ft.²)

K = volume modulus of water $= 43.2 \times 10^6$ (lb. per ft.²)

$w = 62.4$ = specific weight of water (lb. per ft.³)

$\mu = 0.3$

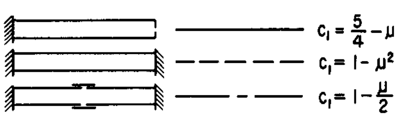


FIGURE 10 Pressure wave velocity in steel pipes

400-hp motors at each pump

WR^2 of each pump and motor $= 385$ lb-ft^2

$N_R = 1760$ rpm; $\eta_R = 0.847$

$$\frac{D}{e} = \frac{32}{0.1875} = 171$$

$a = 3000$ ft/s from Figure 10

$$\frac{2L}{a} = \frac{2(3940)}{3000} = 2.63 \text{ s}$$

$$2\rho = \frac{aV_0}{gH_0} = \frac{(3000)(6.03)}{(32.2)(220)} = 2.55$$

$$K = \frac{(91{,}600)(H_R Q_R)}{WR^2 \eta_r N_R^2} = \frac{(91{,}600)(220)(33.7)}{3(385)(0.847)(1760)^2} = 0.224$$

$$K \frac{2L}{a} = 0.59$$

From Figures 1 to 8 the following results are obtained.

1. Downsurge at pump = (0.92)(220) = 202 ft
2. Downsurge at midlength = (0.64)(220) = 141 ft
3. Upsurge at pump = (0.42)(220) = 92 ft
4. Upsurge at midlength = (0.23)(220) = 51 ft
5. Maximum reverse speed = (1.45)(1760) = 2550 rpm
6. Time of flow reversal at pump = 3.5 L/a = 4.6 s
7. Time of zero pump speed = 5.8 L/a = 7.6 s
8. Time of maximum reverse speed = 10.0 L/a = 13.1 s

As noted in the discussion on pumps equipped with check valves, the upsurge, or head rise, at the pump above the normal head would have been about 202 ft if there were check valves at the pumps that closed at the time of flow reversal.

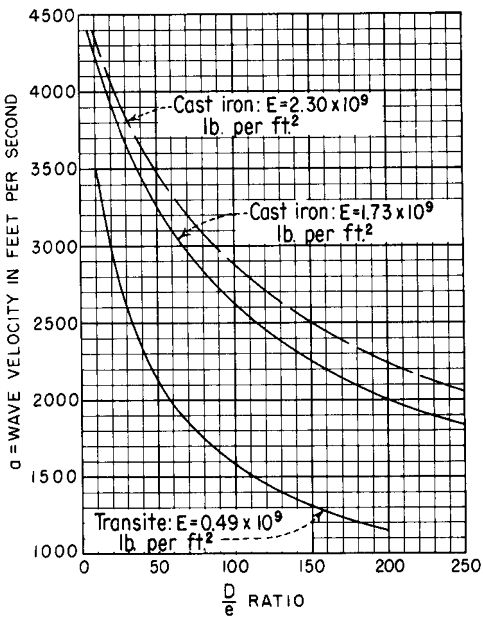
PRESSURE PULSATIONS IN PUMPING SYSTEMS

Motor-driven centrifugal pumps often produce objectionable pressure pulsations in pump discharge lines. The frequency of these pulsations results from the rotating and stationary components of the pump. The following pressure pulsation frequencies have been observed at a number of major pump installations:

1. Fundamental shaft frequency, once-per-revolution pressure pulsation
2. Impeller vane frequency, product of the shaft frequency and the number of impeller vanes
3. Scroll case frequency, product of shaft frequency and the number of guide vanes at either the inlet or discharge side of the pump

Pressure Pulsations at Fundamental Shaft Frequency In pumping systems, objectionable pressure pulsations with a frequency corresponding to the shaft frequency result primarily from a hydraulic unbalance in the pump impeller. The source of such unbalance is usually an eccentricity of the flow passage in the impeller, but in some cases it may be the eccentricity of the outer periphery of the impeller and the pumping action exerted by this eccentricity.

Considerable care must be taken when dynamically balancing a pump impeller to avoid pressure pulsations. In accomplishing this, balance weights are often welded to the top surface or an eccentric machine cut is taken on the outer periphery. Although such surfaces are out of sight after the unit is assembled, objectionable pressure pulsations with a frequency corresponding to the rotational speed of the unit can occur. Such sources of pressure pulsations are very difficult to correct after the unit has been installed. In order to avoid such difficulties, any welding at the top surface or metal removal at the outer periphery of the impeller should be done in such a manner that these surfaces remain smooth and concentric with the axis of rotation. Balance weights on surfaces normal to the axis of rotation should be applied or covered in such a manner that the surface remains flat, smooth, and normal to the axis of rotation.



$$a = \sqrt{\frac{1}{\frac{w}{g}\left(\frac{1}{K} + \frac{Dc_1}{Ee}\right)}}$$

where:

a = wave velocity (ft. per sec.)

g = acceleration of gravity (ft. per sec.^{2})

$\frac{D}{e} = \frac{\text{diameter of pipe}}{\text{thickness of pipe}}$

E = Young's modulus for pipe material (lb. per ft.^{2})

K = volume modulus of water $= 43.2 \times 10^6$ (lb. per ft.^{2})

w = 62.4 = specific weight of water (lb. per ft.^{3})

$c_1 = 1 - \mu^2$

$\mu = 0.3$.

FIGURE 11 Pressure wave velocity in cast iron and transite pipes

Pressure Pulsations at Impeller Vane Frequency The tongue of the pump is the source of the pressure pulsations that are transmitted to the discharge line with a frequency equal to the product of the shaft frequency and the number of impeller vanes. An explanation of the phenomenon is as follows. The velocity distribution at the exit of the pump impeller is not uniform. As this nonuniform flow passes the tongue of the pump casing, an abrupt change in the direction of the impeller exit velocity vector occurs at the proximity of the tongue. This produces a positive pressure wave at the pressure face and a negative pressure wave at the back face of the tongue. From this location, the positive pressure wave travels directly up the discharge line and the negative pressure wave travels completely around the pump casing and is attenuated before reaching the discharge line. These positive pressure pulsations have a frequency equal to the product of the pump speed and the number of impeller vanes. The most effective method for reducing the magnitude of these pressure pulsations is to provide large radial clearance between the outer diameter of the impeller and all guide vanes, consistent with the head discharge requirements. Several manufacturers adopt a minimum radial clearance of 5% of the impeller diameter.

Pressure Pulsations due to Blade-Vane Combinations Objectionable pressure pulsations in pumping systems have been observed at certain multiples of impeller blade passing frequency and the frequency of guide vanes times rotation speed. These pulsations arise from interaction of the flow fields of critical combinations of the numbers of impeller blades and adjacent guide vanes which are located upstream or downstream of the impeller.

The strongest excitation from such pulsations occurs when the number of guide vanes is an exact multiple of the number of impeller blades. This is explained as follows: As the pump impeller rotates, the flow fields of all of the impeller blades simultaneously cross the flow fields between a corresponding number of guide vanes. This disturbance in the flow pattern by all of the runner blades simultaneously produces pressure changes inside the unit at a frequency equal to the number of impeller blades or guide vanes times the rotating speed.

More complex interactions occur for other blade-vane combinations. The remedy for eliminating such excitations is to avoid adverse combinations of impeller blades and adjacent guide vanes. Guidelines for making the proper choice of these combinations are presented in Section 2.1 under the subheading, Blade-Vane Combinations, and the theory described in more detail in References 33 and 65 of that section.

REFERENCES

1. Donsky, B. "Complete Pump Characteristics and the Effects of Specific Speeds on Hydraulic Transients." *ASME J. Basic Eng.,* December 1961.
2. Jones, S. S. "Water-Hammer in a Complex Piping System: Comparison of Theory and Experiment." *ASME* Paper 64-WA/FE-23, New York, 1964.
3. Kinno, H., and Kennedy, J. F. "Waterhammer Charts for Centrifugal Pump Systems." *Proc. ASCE, J. Hydraulics Div.,* May 1965.
4. Lundgren, C. W. "Charts for Determining Size of Surge Suppressors for Pump Discharge Lines." *ASME J. Eng. Power,* Jan. 1967.
5. Kephart, J. T., and Davis, K. "Pressure Surges Following Water-Column Separation." *Trans, ASME J. Basic Eng.,* September 1961.
6. Parmakian, J. "One-Way Surge Tanks for Pumping Plants." *Trans. ASME* 80, October 1958.
7. Parmakian, J. *Waterhammer Analysis.* Dover, New York, 1963.
8. Combes, G., and Borot, B. "New Graph for the Calculation of Air Reservoirs, Account Being Taken of the Losses of Head." *La Houille Blanche,* October–November. 1952.

FURTHER READING

Martin, C. S. "Representation of Pump Characteristics for Transient Analysis: Performance Characteristics of Hydraulic Turbines and Pumps." FED vol. 6 *ASME,* 1983.

Parmakian, J. "Pressure Surges at Large Pump Installations." *Trans, ASME,* August 1953.

Parmakian, J. "Pressure Surges in Pump Installations." *Trans. ASCE* 120, 1955.

Parmakian, J. "Unusual Aspects of Hydraulic Transients in Pumping Plants." *J. Boston Soc. Civil Eng.,* January 1968.

Proceedings of the Second International Conference on Pressure Surges. London, 1976. BHRA Fluid Engineering, Cranford, Bedford MK430AJ, England.