

RESEARCH ABOUT WATER HAMMER

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1.1 INTRODUCTION

Unsteady flow problems are of significant importance in engineering practice because they can result in excessive pressure, Cavitations, noise, and vibration. The problems created by transient flow may be severe to cause failure of a system.

The analysis of unsteady flow in pipeline systems is usually described as an elastic analysis and it is known as water hammer theory where in the elasticity of both the fluid and the pipe walls are taken into account in calculation.

The pressure wave created by velocity changes depend on a number of factors, including the rate of acceleration or deceleration of the fluid, The compressibility of the fluid, the elasticity of the pipe and the overall Geometry of the pipe system.

Water hammer is produced in a closed conduit flow due to a retardation or acceleration of the flow, such as with the change in opening of a valve In the pipeline, in other words a sudden change in flow velocity causes water hammer.

Pipelines must be designed to resist pressures caused by surge waves and must be avoiding column separation which occurs due to water hammer When the pressure drops to vapor level.

The basic equations which describe unsteady flow in pipes are developed by applying the principles of continuity and momentum to control volume.

1.2 WATER HAMMER DEFINITION

Water hammer or hydraulic shock is the momentary increase in pressure, which occurs in a water system there is a sudden change of direction or velocity of the water. When a rapidly closed valve suddenly stops water in a pipeline, pressure energy is transferred to the valve and pipe wall, shock wave are set up within the system. Pressure waves travel backward until encountering the next solid obstacle, then forward, then back again. The pressure wave's velocity is equal to the speed of the sound; therefore it "bangs" as it travels back and forth, until dissipated by friction losses. Anyone who has lived in an older house is familiar with the "bangs" that resounds through the pipes when a faucet is suddenly closed. This is an effect of water hammer.

A less severe form of hammer is called surge, a slow motion mass oscillation of water caused by internal pressure fluctuations in the system. This can be pictured as a slower "wave" of pressure building within the system. Both water hammer and surge are referred to as transient pressures. If not controlled, they both yield the same Results: damage to pipe, fittings, and valves, causing leaks and Shortening the life of the system. Neither the pipe nor the water will compress to absorb the shock.

Entrained air or temperature changes of the water also cause excess pressure in the water lines. Air trapped in the line will compress and will exert extra pressure on the water. Temperature changes will actually cause the water to expand or contract, also affecting pressure. If the pressure in pipeline drops below the vapor pressure of a liquid, then cavities are formed and water column is separated by an air space which grows in size as long as the pressure remains below vapor level. This phenomenon is called “water column separation”. This is the worst condition for water hammer, and may result in the pipe line collapse due to the combined effect of the external forces and the negative pressure effect.

An analysis of water hammer will include calculation the critical time, determining the maximum pressure increase, and selecting a method of control.

1.3 CAUSES OF WATER HAMMER

- 1) Starting and stopping of pumps, where pump start up can induce the rapid collapse of void space that exists downstream from a starting pump. This generates high pressures.
- 2) Valve opening and closing is fundamental to safe pipeline operation. Closing valve at the downstream end of a pipeline creates a pressure wave that move toward the reservoir.
Closing a valve in less time than takes for the pressure surge to travel to the end of pipeline and back is called “ sudden valve closure “. Sudden valve closure will change velocity quickly and can result in a pressure surge. The pressure surge resulting from a sudden valve opening is usually not as excessive.
- 3) Pump power failure can create a rapid change in flow, which causes a pressure upsurge on the suction side and a pressure down surge on the discharge side. The down surge is usually the major problem. The pressure on the discharge side reaches vapor pressure, resulting in vapor column separation.
- 4) Movement of air pockets along the pipeline.

1.4 CELERITY OF PRESSURE WAVE

The speed with which a pressure wave (or elastic wave) travel through still liquid in pipe is known as the celerity of pressure wave and it is represented by symbol (c). As shown in fig. (1.1) let there be a column of still liquid in the pipe. Let us consider a length l of this column as free body. Let,

r be the internal radius of pipe in meters and hence radius of liquid column.

p be the pressure-intensity, in Kg/m^2 , applied on the left vertical face of this column.

$p + \delta p$ be the pressure-intensity, in Kg/m^2 , applied on the right vertical face this column.

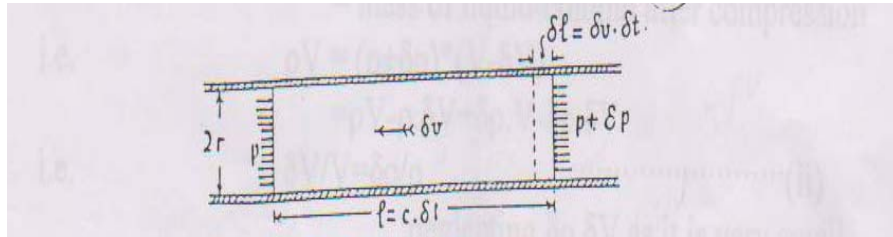


Fig (1.1)

Thus, on this column as a whole, there is more intensity of pressure on the right face, the excess of pressure-intensity from the right side being δp . As a result of this, the pressure wave will travel from the right to the left with celerity c (m/sec) and in time δt (seconds), this pressure wave will travel over a distance of $(c \cdot \delta t)$ meters. Let us consider length l , of liquid column, equal to $(c \cdot \delta t)$ meters.

The excess pressure intensity δp on the right face of liquid column of length $c \cdot \delta t$ will move the column to the left with velocity δv (m/sec), Considering the column will get compressed by δl (meters) in δt (seconds) such that $\delta l = \delta v \cdot \delta t$ (meters). Let,

- V be the volume of liquid-column before the excess pressure is applied.
- $V - \delta V$ be the volume of liquid-column after the excess pressure is applied.
- P be the mass density of liquid-column before the excess pressure is applied.
- $p + \delta p$ be the mass density of liquid-column after the excess pressure is applied.

Now, according to Newton's second law of motion, impulse of impressed force on liquid-column

= change in momentum of the liquid-column, caused by the impressed force.
 i.e. Impressed force * time for which it acts

= mass of liquid-column * change in velocity of liquid-column

i.e. $\delta p \cdot \delta t = (p \cdot v) \cdot \delta v$

Or, $(\delta p \cdot \pi r^2) \cdot \delta t = p (\pi r^2 \cdot c \cdot \delta t) \cdot \delta v$
 $p \cdot \delta v = \delta p / c \dots \dots \dots (I)$

Also, according to the law of conservation of mass, mass of liquid-column before compression

= mass of liquid-column after compression

i.e. $pV = (p + \delta p) \cdot (V - \delta V)$
 $= pV - P \cdot \delta V + \delta p \cdot V - \delta p \cdot \delta v$

i.e. $\delta V / V = \delta P / P \dots \dots \dots (II)$

Neglecting $\delta p \cdot \delta V$ as it is Very small quantity.

Or, $\pi r^2 (\delta v \cdot \delta t) / \pi r^2 (c \cdot \delta t) = \delta p / p$

i.e. $\delta v/c = \delta p/p$
 or, $p \cdot \delta v = c \cdot \delta p$ (iii)

From(i) and (iii) we have,

$\delta p / c = c \cdot \delta p$
 or, $c^2 = \delta p / \delta p$

$c = (\delta p / \delta p)^{1/2}$ m/sec(iv)

Formula (IV) gives the celerity c of pressure wave in terms of δp and the corresponding δp .

Now, let K be the bulk modulus of the liquid, in kg/m².

Then we have,

$K = \delta p / (\delta V/V)$

Or, $\delta V/V = \delta p/K$

i.e. $\delta p/p = \delta p/K$, because $\delta v/v = \delta p/p$ according to (ii).

$\Delta p / \delta p = k/p$.

Putting this value of $\delta p / \delta p$ in (iv) above we have

$C = (K/p)^{1/2}$ m/sec, where P is in metric slugs/m² = $(K/(w/g))^{1/2} = (g \cdot K/w)^{1/2}$,

Where w is the specific weight of liquid Kg/m².

For water at normal temperature and pressure,

$K = 2.1 \times 10^8$ Kg/cm² = $(2.1 \times 10^8 \times 10^4)$ Kg/m².

Putting values of g, K and w in above formula, we have, $c = 1430$ m/sec.

If we consider the elasticity of pipe walls along with the compressibility of liquid we shall have,

$C' = (K/P \{1 + K/E \cdot d/t\})^{1/2}$ m/sec

Where C'=celerity of pressure wave considering both K and E.

E= Elasticity of pipe walls, in kg/m².

D=Internal diameter of pipe, in meters.

T= Thickness of pipe wall, in meters.

ν/m = Poisson's ratio for the material of which the pipe walls are made.

Putting $\nu/m = 0.25$, we have

$C' = (K/p \{1 + K/E \cdot d/t\})^{1/2}$ m/sec
 $= (gK/w \{1 + K \cdot d/t\})^{1/2}$

For steel, $E = 2.1 \times 10^8$ Kg/m²

For C.I., $E = 1.0 \times 10^8$ Kg/m².

Now, $c' = (K/p)^{1/2} \cdot 1/(1 + K/E \cdot d/t)^{1/2}$, taking $\nu/m = 1/4$

Hence we found that the pressure wave moves faster in rigid pipe than in pipe with elastic wall.

Value of c' will be more than 1430 m/sec and increase will depend on the values of K, E, d and t.

Actual velocity of pressure wave will be given by,

$uw = v \pm c$.

However, since v is very small as compared to c, we will have,

$$Y_w = \pm c.$$

As usual, ωw is positive when the pressure wave goes down the pipe in the direction of v ; ωw is negative when the pressure wave travels up the Pipe in a direction opposite to that of the velocity of flow v in the pipe.

1.0 **PRESSURE DUE TO SUDDEN VALVE CLOSURE**
(Neglecting the elasticity of pipe walls but considering the compressibility of the liquid)

Under this condition, it is assumed that the pipe walls are rigid and inelastic and hence, they do not get distended when the intensity of pressure increases or decreases periodically due to sudden closure. It is however assumed that when the intensity of pressure increases, The volume of liquid (in pipe) decreases due to the compressibility of liquid . let,

l be the length of pipe under consideration, shown in fig. (1.1).

d be the diameter of the pipe.

V be the volume of liquid in the pipe of length l .

v be the velocity of flow in the pipe, when the valve is fully open.

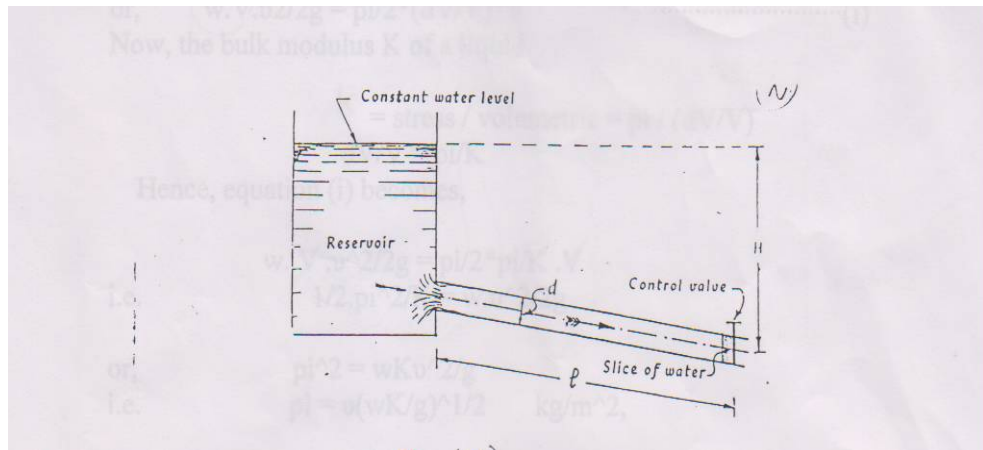


Fig (1.1)

Now, let the valve be suddenly closed. Due to this, the circular vertical slice of liquid-column (shown hatched), in the immediate backside of the valve will have its velocity reduced from v to 0 ; also due to the increase in pressure of the slice by ρv^2 Kg/m^2 , the slice of liquid gets compressed from its original small thickness b to the thickness $(b - \delta b)$. Other slices, in turn, will meet the same fate of the first slice and hence after some time t_w (where $t_w = l/v$) second, the entire column of liquid of length, l diameter d and velocity v will become of length $(l - \delta l)$, velocity zero, and same diameter d since the pipe walls are assumed to be rigid and unyielding.

Also this column, at zero velocity, will have the maximum increase in pressure of π Kg/m² and a decrease in volume of δV ; in other words, The strain energy (or resilience) has been stored in the column of liquid

The loss in K.E. Of liquid

= strain energy stored in the column of liquid

i.e. $Wv^{1/2}/g = \pi^{1/2} * (\text{volumetric strain due to } \pi) * \text{volume of liquid-column,}$

Where W is the weight of liquid-column

Or, $w.V.v^{1/2}/g = \pi^{1/2} * (dv/v)*V \dots\dots\dots (i)$

Now, the bulk modulus K of a liquid

= stress / volumetric = $\pi / (dV / V)$

$dV/V = \pi/K$

Hence, equation (i) becomes,

$w.V.v^{1/2}/g = \pi^{1/2} * \pi/K.V$

i.e. $v^{1/2} * \pi^{1/2}/K = w.v^{1/2}/g$

Or, $\pi^{1/2} = wKv^{1/2}/g$

i.e. $\pi = v (wK/g)^{2/3} \quad \text{Kg/m}^2$

Where w = specific weight of liquid, in Kg/m³.

And, K = bulk modulus of liquid, in Kg/m².

Note:

- a) $\pi = v (w/g * K)^{2/3} = v(\rho.K)^{2/3} = \rho.v(K/\rho)^{2/3} = \rho.v.c.$
- b) Water hammer in meters or, increased pressure head in meters caused = π/w meters of liquid.
- c) $1/vw$ is called the wave-reflection time after which a pressure wave, starting from one cycle of the pressure wave; it is also known as the periodicity of pressure wave.
- d) Since E is neglecting, π found out above is known as approximate π .

1.9.1 Pressure due to sudden valve closure

(Considering both the elasticity of pipe walls and the compressibility of the liquid)

When the intensity of pressure of the liquid increases, the pipe walls (which are actually elastic) get distended to diameter $d + \delta d$ and hence the diameter of liquid-column under pressure also increases from d to $(d + \delta d)$. Thus, at the end of t_w seconds we have:

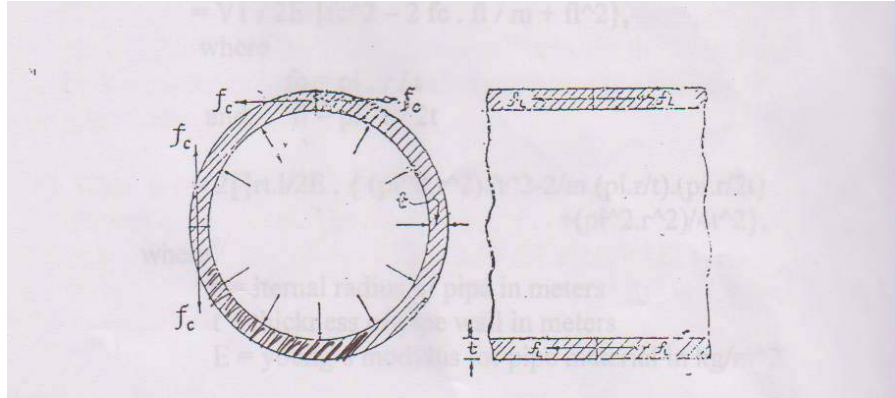


Fig (1.3)

A column of liquid of length $(l + \delta l)$, volume $V + \Delta V$, diameter $(d + \delta d)$, velocity zero and, with its pressure increased by $p_i \text{ kg/m}^2$.

In this case therefore we will get,

Loss in K.E. of liquid-column

= strain energy stored in liquid-column + strain

Energy stored in pipe walls(i)

Now, the increase p_i in radial pressure induces the circumferential stress f_c and the longitudinal stress f_l in pipe walls, as shown in fig. (1.3). Hence, strain energy stored in pipe walls

= strain energy due to the induced circumferential stress f_c + strain energy due to the induced longitudinal stress f_l (see fig. 1.3).

Thus, stored energy = $(f_c/\tau \times \text{strain in direction of } f_c \times \text{volume of pipe$

Walls) + $(f_l/\tau \times \text{strain in direction of } f_l \times \text{volume of pipe walls)$

$$= f_c/\tau \cdot \{f_c/E - f_l/mE\} \cdot V_1 + f_l/\tau \{f_l/E - f_c/mE\}/V_1$$

where

V_1 is the volume of pipe walls.

$1/m$ is the poisson's ratio for the material of pipe walls.

$$= V_1/\tau E \{f_c^2 - \tau f_c \cdot f_l/m + f_l^2\},$$

where

$$f_c = p_i \cdot r/t$$

$$\text{and, } f_l = p_i \cdot r/\tau t$$

$$= \frac{2\pi r t \cdot \frac{1}{2} E \cdot \left\{ \frac{(\rho \cdot r^2) / t^2 - \rho / m \cdot (\rho \cdot r / t) \cdot (\rho \cdot r / 2t)}{+ (\rho \cdot r^2) / \xi t^2} \right\},$$

where

r = internal radius of pipe in meters

t = thickness of pipe wall in meters

E = young's modulus for pipe material in kg/m^2

$$\begin{aligned} \text{Stored energy} &= \frac{2\pi r t \cdot \frac{1}{2} E \cdot \rho \cdot r^2 / t^2 \left\{ \frac{\rho \cdot r^2 - \rho / m}{\rho \cdot r^2} \right\}}{=} \\ &= \frac{(\rho \cdot r^2 \cdot \frac{1}{2} \cdot \rho \cdot r^2) / \xi E t \cdot (\rho \cdot r^2 / m)}{=} \\ &= (V \cdot \rho \cdot r^2) / \xi E t \cdot (\rho \cdot r^2 / m) \end{aligned}$$

Hence, equation (i) becomes,

$$(w \cdot V) / g = \rho \cdot r^2 / K \cdot V + V \cdot (\rho \cdot r^2 / \xi E t) \cdot (\rho \cdot r^2 / m)$$

$$\text{i.e. } w / g = \rho \cdot r^2 / K + (r \cdot \rho \cdot r^2 / \xi E t) \cdot (\rho \cdot r^2 / m)$$

$$= \rho \cdot r^2 \left\{ \frac{1}{K} + r / \xi E t (\rho \cdot r^2 / m) \right\}$$

$$\rho = \sqrt{(w/g \left\{ \frac{1}{K} + r / \xi E t (\rho \cdot r^2 / m) \right\})} \cdot \text{kg/m}^2$$

where

ρ / m is usually taken at ρ / ξ if its value is not given and, water hammer of liquid = ρ / w meters of the liquid.

Note:

$$\begin{aligned} \text{a) } \rho &= \sqrt{(w/g \left\{ \frac{1}{K} + d / \xi E t (\rho \cdot r^2 / m) \right\})} \\ &= \sqrt{(\rho / \left\{ \frac{1}{K} + d / \xi E t (\rho \cdot r^2 / m) \right\})} \\ &= \rho \cdot \sqrt{(\frac{1}{\rho} \left\{ \frac{1}{K} + d / \xi E t (\rho \cdot r^2 / m) \right\})} \\ &= \rho \cdot \sqrt{(K / \rho \left\{ \frac{1}{K} + d / \xi E t (\rho \cdot r^2 / m) \right\})} \\ &= \rho \cdot v \cdot c', \text{ where } c' \text{ is the celerity of pressure wave, taking } K \\ &\text{and } E \text{ into consideration.} \end{aligned}$$

$$\text{b) } \rho \text{ can also be put as, } \rho = \sqrt{(w/g \cdot K)} = \sqrt{(w/g \cdot \frac{1}{(1/K)})}$$

Where $1/K' = 1/K + d/\rho E(\sigma - \epsilon/m)$; here K' is called the equivalent bulk modulus of liquid making allowance for the elasticity of pipe wall.

- c) When E of pipe wall is also considered, p_i so found out is known as correct p_i .
The ratio, correct p_i /approx. p_i

$$= \sqrt{\rho K'} / \sqrt{\rho K} = (K'/K)^{1/2}.$$

1.6 WATER HAMMER CONTROL

As mentioned before, the most frequently encountered water hammer problems in water engineering are related to either sudden pump stopping as due to power failure, or due to rapid valve closure. There is one basic principle which should be always considered in the design and operation of pipelines, that is avoiding sudden changes in velocity. As water hammer is related to changes in velocity, the change in pressure is directly related to the change in velocity.

Therefore, avoiding sudden changes in velocity will generally avoid serious water hammer pressures.

Practical control devices, which can be used to limit the water hammer effects due to pump cut-out, include:

- 1) Use of pump by-pass with non-return valve;
- 2) Installation of an air vessels (air chamber);
- 3) Use of surge tank;
- 4) Use of air valves;
 - i. Air release valve.
 - ii. Air / vacuum valve.
 - iii. Combination air valve.
- 5) Use of pressure relief valves;
- 6) Use of check valves;
- 7) Use a fluid kinetics hydro pneumatic surge arrestor.

1.6.1 PUMP BY-PASS PIPE WITH NON-RETURN VALVE

A classical method for the protection against water hammer is installing a non-return valve on a bypass pipe connected in parallel to the delivery pipeline and the other end connected to the suction pipe as shown fig (1.8). In normal pumping condition, the pressure in the delivery side is higher than the pressure in the suction side and therefore the non-return valve is closed. However, when the pumps are

suddenly stopped the pressure in the delivery pipe will be dropped. If the pressure in the delivery side drops below the pressure in the suction side, then the non-return valve will be opened allowing water to flow to the delivery side. In such cases, the pressure in the delivery side will be dropped to the suction pressure. This arrangement is only useful if the pressure drops below the suction pressure.

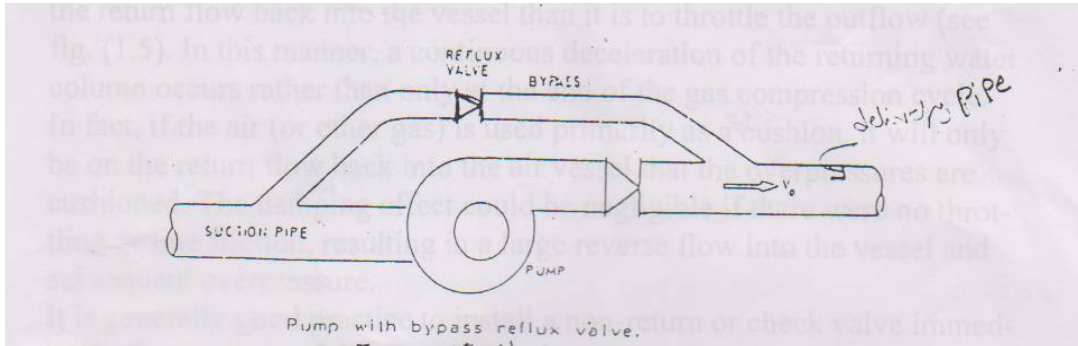


Fig (1.5): pump with bypass reflux valve

1.6.2 AIR VESSELS

The most effective way of preventing negative pressures and also for reducing overpressures is the use of compressed air vessels (also known as air chambers, pressurized surge tanks, pneumatic tanks).

Air vessels generally alleviate negative pressures more effectively than other forms of water hammer protection, and they can maintain a positive pressure in the line all stages following pump trip. This is accomplished by forcing water out of the vessel into the cavity, otherwise created following pump trip or flow stoppage at the upstream end. The compressed air forces water from the air vessel into the pipeline, allowing the water column traveling up the pipeline to maintain its momentum. Friction and other head losses tend to reduce the water velocity and therefore the subsequent oscillations. Thus, some degree of flow throttling is often used in conjunction with the cushioning effect of air vessels.

In particular, throttling of the outflow from the vessel may assist in reducing the water velocities more rapidly than a pure cushioning air vessel (see Graze and Holacher 1989). However, it is more common to throttle the return flow back into the vessel than it is to throttle the outflow (see fig. (1.6)). in this manner, a continuous deceleration of the returning water column occurs rather than only at the end of the gas compression cycle. In fact, if the air (or other gas) is used primarily

as a cushion, it will only be on the return flow back into the air vessel that the overpressures are cushioned. The damping effect could be negligible if there were no throttling or line friction, resulting in a large reverse flow into the vessel and subsequent overpressure.

It is generally good practice to install a non-return or check valve immediately downstream of the pumps (i.e., between the pumps and air vessel) to prevent flow backward through the pumps.

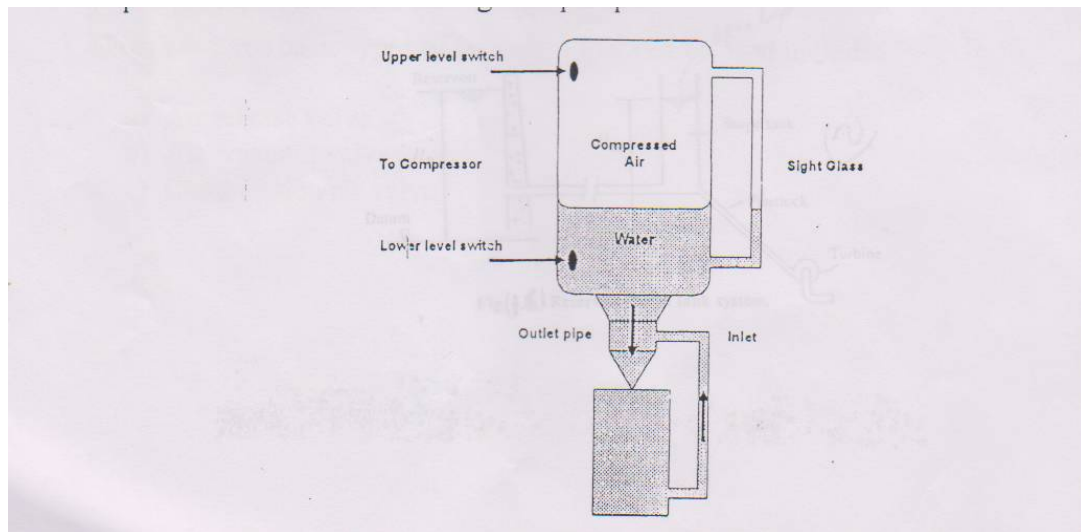


Fig (1.5) schematic of air vessel

1.1.3. SURGE TANK

Surge tanks are open-top vessels connected to the pipe system in which pressure transients are to be controlled. They are similar to air vessels, differing in the respect that the overlying air pressure remains constant. They give rise to a similar mass oscillation under transient flow conditions and are often used in hydropower plants, as illustrated in fig. (1.6) when flow to the turbine is throttled back, the penstock is subjected to water hammer transient pressure. The surge tank prevents these transients from reaching the supply main connecting the reservoir to the surge tank. This system may be analyzed using the water hammer equations already presented. Alternatively, the transient behavior of the reservoir-pipe-surge tank part of the system can be modeled as a simple mass oscillation:

$$\text{Momentum equation: } \rho g(H_R - H_{st}) A - \rho \cdot g \cdot h_f \cdot A = \rho \cdot A \cdot L \cdot \Delta v / \Delta t$$

$$\text{Continuity : } \Delta H_R = - Q \cdot \Delta t / A_R \quad \text{and} \quad \Delta H_{st} = Q \Delta t / A_{st}$$

Where h_f is the flow head loss between reservoir and tank, A_R and A_{st} are the reservoir and surge tank plan areas, respectively, and A is the pipe cross-sectional area.

This set of equations can be solved numerically to determine the variation of H_R and H_{st} with time, resulting from an abrupt change in flow to the turbine.

Surge tanks act as temporary storage for excess liquid that has been diverted from the main flow to prevent overpressure, or as supplies of liquid to be added in the case of negative pressure.

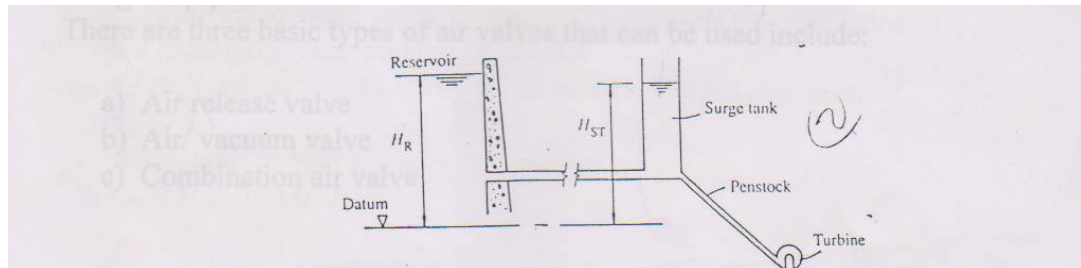


Fig (1.6) Reservoir- surge tank system

1.7.4 AIR VALVES

There are three primary sources of air in a pipeline. First, at start up, the pipeline contains air which must be exhausted during filling. As the pipeline is filled, much of the air will be pushed downstream and released through hydrants, facets, and other mechanical apparatus. A large amount of air, however, will become trapped at system high points. Second, water contains about 1% air by volume based on normal solubility of air in water. The dissolved air will come out of solution with a rise in temperature or a drop in pressure which will occur at high points due to the increase in elevation. Finally, air can enter through equipment such as pumps, fittings, and valves when vacuum conditions occur.

The effect of trapped air in a pipeline can have serious effects on system operation and efficiency. As air pockets collect at high points, a restriction of the flow occurs which produces unnecessary head loss. A pipeline with many air pockets can impose enough restriction to stop all flow. Also, sudden changes in velocity can occur from the movement of air pockets. When passing through a restriction in the line such as a control valve, a dislodged pocket of air can cause surges or water hammer. Water hammer can damage equipment or loosen fittings and cause leakage. Finally, corrosion in the pipe material is accelerated when exposed to the air pocket which can result in premature failure of the pipeline.

Air is sometimes removed from a line with a manual vent during initial start-up but this method does not provide continual air release during system operation nor does it provide vacuum protection. Today, municipalities use a variety of automatic air valves at the pump discharge and along the pipeline.

There are three basic types of air valves that can be used include:

- a) Air release valve.
- b) Air/ vacuum valve.
- c) Combination air valve.

a) **AIR RELEASE VALVE**

Air Release Valves function to release air pockets that collect at each high point of a full pressured pipeline. Air Release Valves can open against internal pressure, because the internal lever mechanism multiplies the float force to be greater than the internal pressure. This greater force opens the orifice whenever air pockets collect in the valve. Air Release Valves are essential for pipeline efficiency and water hammer protection.

Immediate availability

Sizes 1/2" thru 3". Sizes 4" thru 36".

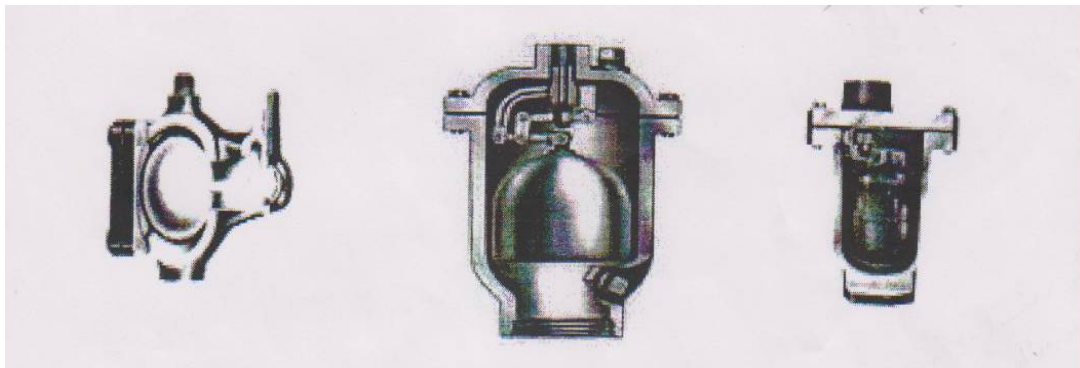


Fig (1.V)

b) AIR AND VACUUM VALVES

Are float operated, having a large discharge orifice, equal in size to the valves inlet. Air and Vacuum valves allow large volumes of air to be exhausted from – or – admitted into a water pipeline as it is being filled or drained. As the pipeline fills, water enters the Air Valve, raises the float and shuts- off. When draining the pipeline, the float drops, allowing air to enter, preventing vacuum, possible pipeline collapse and damaging water column separation.

Air & Vacuum Valves are an efficient means to fill and drain pipelines.

Immediate availability.

Sizes ½" thru 6".

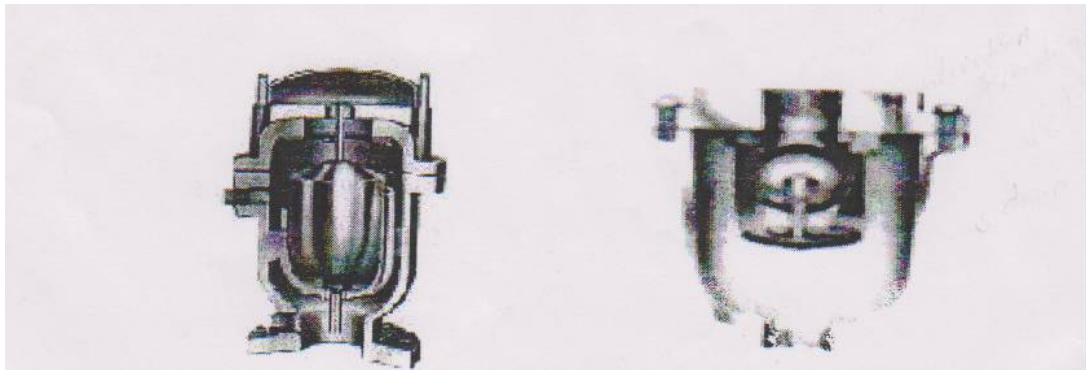


Fig. (1.8)

C) COMBINATION AIR VALVES

As the name implies, combines features of Air & Vacuum Valves and Air Release Valves. These valves are also called Double Orifice Air Valves. These Valves are installed on all high points of a system where it has been determined dual function Air & Vacuum and Air Release Valves are needed to vent and protect a pipeline. Generally, it is sound engineering practice to use (CAV) instead of simple purpose Air & Vacuum Valves. (CAV) are available in two body styles – (1) single body combination (2) custom built combination (CCAV) with two bodies.

The single body (CAV) is used where compactness is preferred and / or where risk of tampering exists due to accessibility of the installation.

Immediate availability

Sizes ½" thru 24".

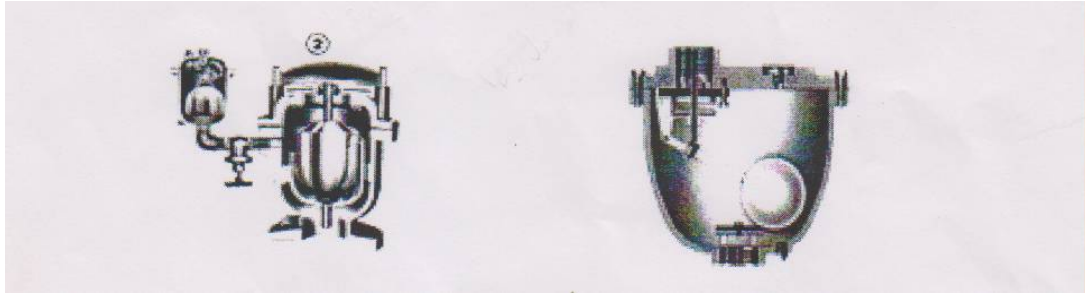


Fig. (1.9)

AIR VALVE LOCATIONS ALONG A PIPELINE

Air valves are installed on a pipeline to exhaust air and admit air to prevent vacuum conditions and air related surges.

The AWWA Steel Pipe Manual recommends Air Valves at the following points along a Pipeline (2).

- a. **High Points:** Combination Air Valve.
- b. **Long Horizontal Runs:** Air Release or Comb. Valve at 1200 to 2000 ft. (380 to 610M) intervals.
- c. **Long Descents:** Combination Air Valve at 1200 to 2000 ft. (380 to 610M) intervals.
- d. **Long Ascents:** Air/ Vacuum Valve at 1200 to 2000 ft. (380 to 610N) intervals.
- e. **Decrease in an Up Slope:** Air / Vacuum Valve.
- f. **Increase in a Down Slope:** Combination Air Valve.

Also, on very long horizontal runs, Air Release and Combination Air Valves will be used alternately along the pipeline. It should be noted that Combination Valves can be used at any location instead of Air Release or Air/Vacuum Valves to provide added air release capacity on the pipeline.

It is important to establish a smooth pipeline grade and not follow the terrain or an excessive number of Air Valves will be needed.

The designer must balance the cost of air valve locations with the cost of additional excavation. The high points and grade changes that are less than 1 pipe diameter are typically ignored because the flow will flush accumulated air downstream.

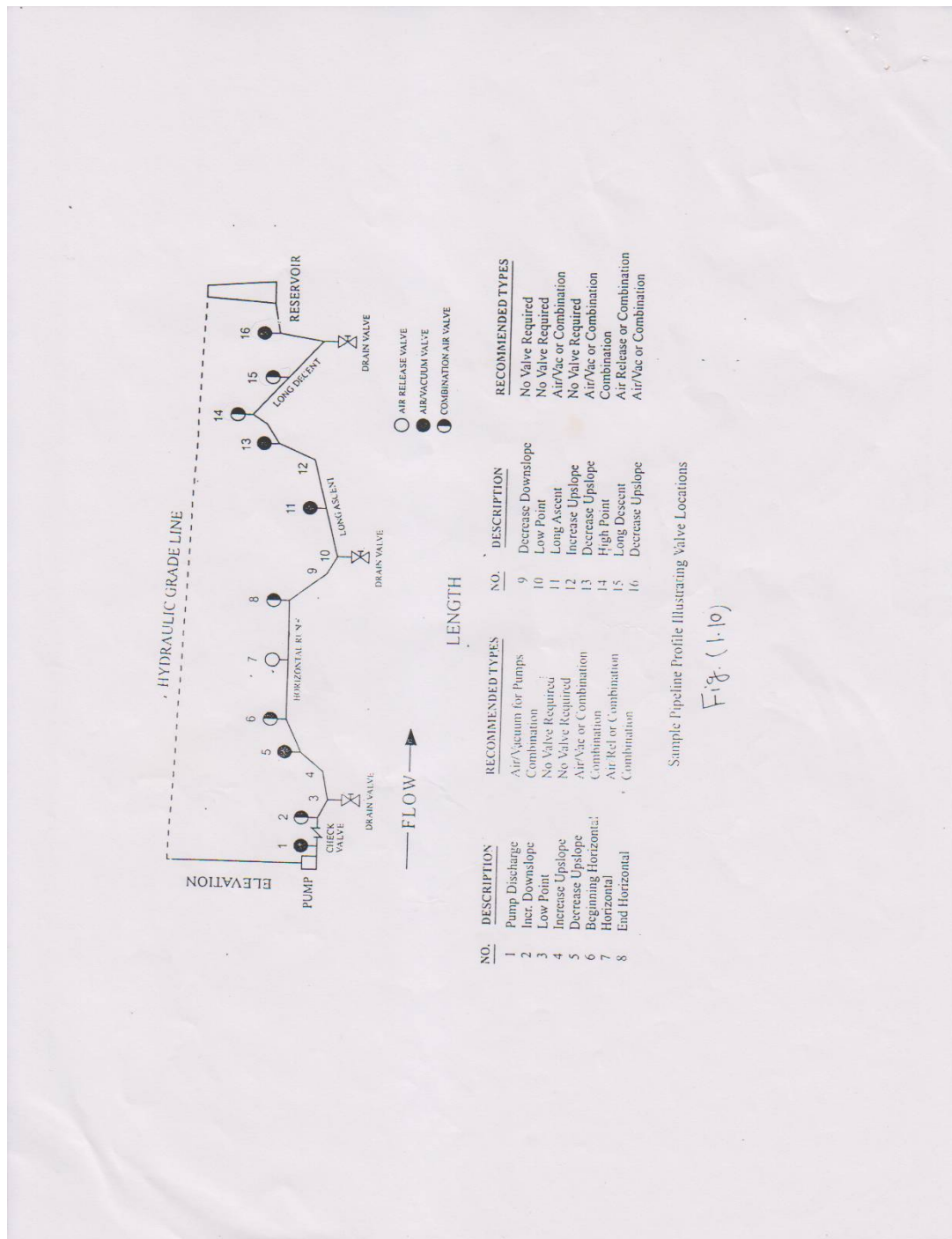


Fig. (1.10)

Fig (1.10): sample pipeline profile illustrating valve locations

AIR/ VACUUM VALVE SIZING

Some publications list a rule of thumb that suggests Air/ Vacuum Valves be 1 in. (25.4mm) per 1ft. (0.305M) of pipe diameter (D). So a 2 ft. (0.61M) diameter line would have a 2 in. (50.8mm) diameter valve. Based on over thirty years of successful air valve application, Val-Matic® has developed sizing criteria that form the basis for the following methodology. The methodology is based on sizing the air/vacuum valve for two conditions:

Admitting air to prevent a vacuum in the pipeline and exhausting air during filling of the pipeline.

The Air/Vacuum of Combination Air Valve should be capable of admitting air after power failure or line break at a rate equal to the potential gravity flow of water due to the slope of the pipe. The flow of water due to slope can be found by the equation:

$$Q = 0.000785 C^2 (S D^5)^{1/2} \quad (1)$$

Where:

Q = Flow of water, CFS

C = Chezy Coefficient = 110 for iron pipe.

S = Slope of pipe, vertical rise/horizontal run.

D = Pipe inside diameter, inches.

The gravity flow due to slope is calculated for every pipe segment. For stations where there is a change in up slope or down slope, the difference between the upstream and downstream flows is used for sizing because the upper segment feeds the lower segment and helps prevent a vacuum from forming.

When steel or any collapsible pipe is used, it is important to determine if there is a risk of pipeline collapse due to the formation of a negative pressure. The following equation finds the external collapse pressure of thin wall steel pipe using a safety factor of 2. A safety factor of 2 is recommended to take into account variances in pipe construction, variances in bury conditions, and possible dynamic loads.

$$P = 16,700 \cdot \dots \cdot (T/D)^2 \quad (\Sigma)$$

where:

P = Collapse Pressure, psi.

T = Pipe Thickness, in.

D = Pipe Diameter, in.

Collapse may also be a concern on large diameter plastic or ductile iron pipe. The pipe manufacturer should be asked to provide maximum external collapse pressures. The valve should be capable of admitting the flow due to slope without exceeding the lower of the calculated pipe collapse pressure or 0 PSI (0kPa). 0 PSI (0kPa) is used for sizing to remain safely below the limiting sonic pressure drop of VPSI (5kPa). Manufacturers provide capacity curves for their valves which can be used to select the proper size. The capacity of an Air/ Vacuum Valve can be estimated using:

$$Q = 178 \cdot Y \cdot d^2 \cdot C \cdot (DP \cdot P_1 \cdot T \cdot Sg)^{1/2} \quad (\circ)$$

where:

q= Air Flow, SCFM

Y= Expansion Factor

.79 (for vacuum sizing)

.80 (for exhaust sizing at 0 psi)

.92 (for exhaust sizing at 2 psi)

d= Valve Diameter, in

DP = Delta Pressure, psi

The lower of 0 psi or pipe collapse pressure (for vacuum sizing)

2 or 0 psi (for exhaust sizing)

P₁ = Inlet pressure, psi

$1.5 \cdot V$ (for vacuum sizing)

$16 \cdot V$ or $19 \cdot V$ psi (for exhaust sizing at γ or δ psi)

T_1 = Inlet Temperature = $\delta T \cdot R$

S_g = Specific Gravity = 1 for air

C = Discharge Coefficient = .6 for square edge orifice.

The air valve should also be sized for exhausting air during filling of the system. The flow rate used for venting should be the fill rate of the system. The fill rate may be the flow rate from a single pump in a multiple pump system. If there is only one pump in the system, then special filling provisions should be taken such as the use of a smaller pump for filling or the ability to throttle the flow from the pump to achieve a fill rate in the range of 1 to 2 ft/sec (.3 to .6M/sec). Higher fill rates may cause surges in the line and Anti Slam Devices should be used to reduce the surges within Air/ Vacuum or Combination Valves.

If a fill rate is not given, the Air/Vacuum Valve will be sized for the design flow rate which may cause the valve to be oversized.

Every effort should be made to establish a reasonable system fill rate. The differential pressure used for sizing the Air/Vacuum Valve varies. 2PSI (13kPa) will be used in most cases. When the valve is equipped with an Anti Slam Device, the differential pressure may be as high as 8PSI (56kPa). Higher differentials are not used because the possibility of water reaching the Air/Vacuum Valve with excessive fluid velocities and to eliminate the noise associated with sonic velocities.

The final Air/Vacuum Valve size must have a capacity greater than both the required exhausting and admitting requirements.

AIR RELEASE VALVE SIZING

The capacity of releasing air under line pressure through an Air Release Valve can be estimated by using the Air/Vacuum Valve formula except P_1 will equal the operating pressure in the line. The differential pressure (DP) is limited by sonic velocity to about $0.5 \cdot V \cdot P_1$. The corresponding expansion factor (Y) is 0.5 .

$$q = 1.0 \cdot d^2 \cdot C \cdot (P + 1.0) \quad (6)$$

where:

q= Air Flow, SCFM

d= Orifice Diameter, in

P= Pipeline Pressure, psig

C= Discharge Coefficient = .60 for Air Release orifice

It is difficult to determine in advance the amount of entrapped air which must be released from a given system. Based on water containing 2% air, the maximum flow rate can be used to compute a nominal venting capacity.

$$Q = q \cdot (.12 \text{ cu ft/gal}) \cdot .2 \quad (7)$$

where:

q=Air Flow, SCFM

Q=System Flow Rate, GPM

In most cases, the size of the Air Release Valve is a judgment decision based on experience. The 2% air content can be varied depending on the potential for entrained air in the water source. The Air Release Valve inlet connection should be as large as possible to maximize the exchange of air and water in the valve.

1.6.9 PRESSURE RELIEF VALVE

Entrained air or temperature changes of the water can be controlled by pressure relief valves, which are set to open with excess pressure in the line and then closed when pressure drops. Relief valves are commonly used in pump stations to control pressure surges and to protect the pump station. These valves can be an effective method of controlling transients. However, they must be properly sized and

selected to perform the task for which they are intended without producing side effects.

If pressure may drop at high points, an air and vacuum relief valve should be used. All downhill runs where pressure may fall very low should be protected with vacuum relief valves. Vacuum breaker-air release valves, if properly sized and selected, can be the least expensive means of protecting a piping system. A vacuum breaker valve should be large enough to admit sufficient quantities of air during a down surge so that the pressure in the pipeline does not drop too low. However, it should not be so large that it contains an unnecessarily large volume of air, because this air will have to be vented slowly, increasing the downtime of the system. The sizing of air release valves is, as mentioned, critical.

1.1.1 CHECK VALVE

These valves are used on pump discharge to avoid reverse flow through, and rotation of pumps. They are also useful in some locations to reduce risks of large pressure rises following cavity collapse.

a) SILENT CHECK VALVES

Excellent to prevent Water Hammer, in multi-story building and vertical turbine pump installations, when pumping from a well to an elevated reservoir. The Valve closes, SILENTLY! Low in cost, reliable and requires no regular maintenance. Principle of operation: When pump stops, spring forces disc closed instantly against slight pump head at zero velocity, (theoretically a static condition) hence, SILENT CLOSURE! Body- Cast Iron, Ductile Iron, Cast Steel, Stainless Steel or Bronze.

Internals- Bronze, Stainless. Ratings 150# thru 600# class.

Available Immediate or maximum 4 weeks.

Sizes 1" thru 10".



Fig. (1.11)

1.6.7 USE OF A FLUID KINETICS HYDROPNEUMATIC SURGE ARRESTOR

A Fluid Kinetics Surge Arrestor is a hydro pneumatic tank with specially designed energy dissipating internals. Its function is to provide water to and accept water from the pipeline, thereby allowing gradual deceleration of the fluid column and reduction of return surge. Predetermined up surge and down surge limitations are met by correct sizing of the tank and its internals.

Open to line pressure. When the pump shuts down and the fluid column moves away, pressure at the surge arrestor connection to the line decreases, creating a pressure imbalance. The surge arrestor automatically reacts to this imbalance by "pumping" water rapidly into the line. This outflow action continues until the fluid column comes to rest and starts its return flow back towards the pump.

Now the surge arrestor acts as brake, absorbing this return energy in friction from the internal energy dissipater and compression of the air charge. This outflow-inflow pattern provides the correct amount of system dampening to prevent column separation and limit excessive return surge and pressure cycling.

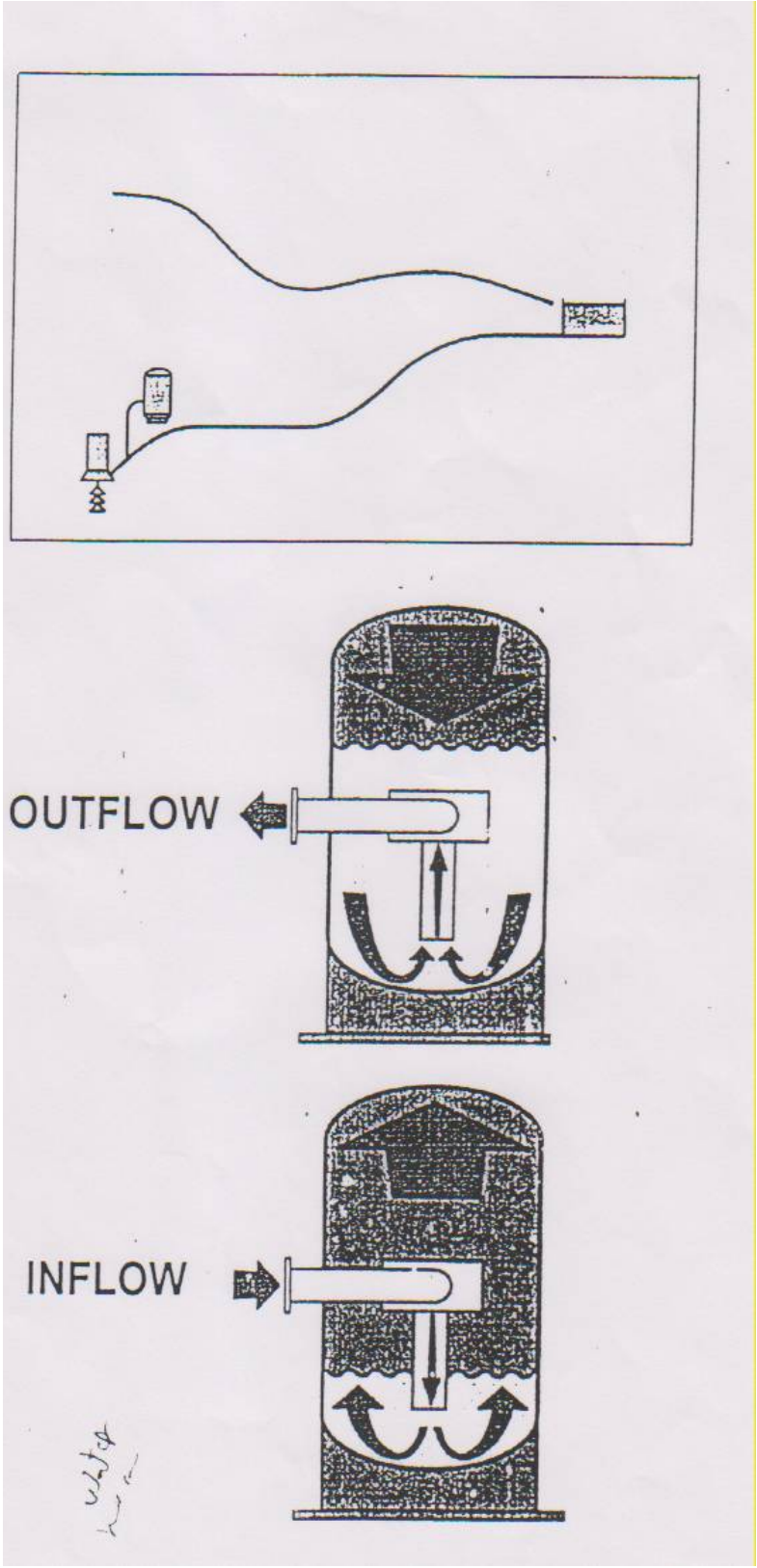


Fig (1.12)

1.7 DESIGN OF AIR VESSELS

Air vessels offer an effective means of reducing water hammer overpressures and negative pressures due to pump trip in pipelines. The process of sizing air vessels is simplified with the nomographs presented. Both air volume and total vessel size are calculated. The size can be minimized by correct selection of outlet and inlet connector diameter, and guides are provided to make these selections.

1.7.1 AIR VESSEL SIZING

The incompressible flow differential equations of motion were analyzed for a number of cases in order to obtain a generalized air vessel volume as a function of the minimum relative head at the pumping station. Fig. (1.12). Shows the nomenclature and minimum and maximum head envelope for a generalized pipeline. Using the results of the analyses, summarized in Fig. (1.13), the minimum head can be calculated as a function of the initial pumping head.

The symbols are as follows:

S' = dimensionless gas volume.

$S \cdot gH_0 / ALV_0$;

S = air vessel volume = $S_0 + S_w$;

S_0 = gas volume at steady state operating pressure;

S_w = volume of water in vessel at steady state operating pressure;

g = gravitational acceleration;

H_0 = initial steady state pumping head above pump level;

L = length of pipeline;

V_0 = initial steady state water velocity in pipeline;

X = distance from pump end of pipeline;

and A = cross-sectional area of pipe.

Fig. (1.10) compares the results for air vessel volume S , after performing a number of analyses using the full water hammer equations, and with the simplistic incompressible flow equation. The theoretical incompressible flow equation gives a cavity volume of $(ALV_0^2 / 2gh)$, where h is the head difference across the water column.

The accompanying chart Fig. (1.10) presents the dimensionless air volume and expanded air volume (i.e., air vessel volume for different desired relative minimum head requirements at the pump station).

The values calculated for plotting the line from the incompressible flow theory are indicated as squares, whereas the crosses indicate values obtained from a full elastic analysis. The elastic analysis was performed by computer simulation using finite differences (Stephenson 1966), which accounts for elastic wave celerity. However, the more sophisticated analysis results agreed remarkably with the simple analysis hereunder. Incompressible flow theory suggests the following relationship between decelerating head on a water column and the rate of deceleration:

$$h = -L/g \cdot dV/dt \quad (1)$$

Which may be integrated to obtain the maximum cavity volume remaining upstream before the water column reverses, i.e.,

$$S_w = ALV \cdot \sqrt{\gamma/g} h \quad (2)$$

where

S_w = volume of water the air vessel would force into the pipeline behind the water column,

A = cross-sectional area of the pipe,

L = its length,

V_0 = initial water velocity,

g = gravitational acceleration,

and h = average decelerating head (h_{min}/γ)

For rapid expansion (small air vessels), the gas expansion law may be adiabatic (i.e., no heat loss), but for slower expansion there may be heat loss and the temperature is nearer constant (i.e., isothermal)

(Graze and Horlacher 1986).

From Boyle's law for isothermal gas expansion in the air vessel

$$S \cdot H_0 = (S + S_w) (H_0 - h_{min}) \quad (3)$$

Where

H_0 = initial head (absolute) on the gas with volume S_0

and h_{min} = most severe drop in pressure head of the gas in the vessel.

Solving for S_0 and substituting for S_w

$$S_0 = (H_0/h_{min})^{-1} (ALV_0^{\gamma}/\gamma H_0 g(h_{min}/\gamma H_0)) \quad (\xi)$$

Or

$$S_0/(ALV_0/H_0 g) = H_0/h_{min}(H_0/h_{min})^{-1} \quad (\phi)$$

Which is plotted as squares in Fig. (1.16) (initial air volume, incompressible liquid). That is, the line gives dimensionless gas volume as a function of relative minimum head at the pump station. Also plotted are points from various full elastic water hammer analyses. The upper line is a plot of dimensionless total vessel volume, i.e.

$$(S_0 + S_w)/(ALV_0^{\gamma}/H_0 g) \quad (\tau)$$

obtained from Boyle's law, Eq. (3).

No outlet throttling or line friction was considered in the calculation, and the air expansion was assumed to be isothermal,

Whereas for small vessels it is nearer adiabatic. Therefore, the more general expansion equation is

$$S \cdot P^k = SP^k \quad (v)$$

Where

P = pressure, and k is between 1.0 (isothermal) and 1.4 (adiabatic).

As a first approximation, isothermal conditions were assumed, although the results are not highly sensitive to the assumed value of k .

1.4.2 OUTLET AND INLET PIPE SIZES

a) OUTLET

The outlet pipe from an air vessel can be throttled to reduce the outflow volume and to more rapidly decelerate the water column in the pipeline. The outflow from the vessel is out of phase with the minimum pressure Fig. (1.16), so the

maxima of the two pressure drops (air expansion and outflow head loss) cannot be summated. The maximum rate of outflow occurs immediately after pump trip, when the air in the vessel is still at operating pressure. Then, before any deceleration of the water column has occurred, the head loss into the main is

$$h_1 = K V_e^2 / \gamma g \quad (8)$$

and since from continuity

$$V_e D_e^2 = V_p D_p^2 \quad (9)$$

$$h_1 = \gamma V_p^2 (D_p / D_e)^4 / \gamma g \quad (10)$$

where

K=loss coefficient accounting for entrance contraction and expansion losses plus pipe work losses, and is typically 2.0.

V= velocity, with subscript e referring to the exit pipe,

p= to the main pipe,

and D= internal diameter of the respective pipe.

The permissible head drop in the main pipe should be less than the static head H_0 , and often only a small fraction of H_0 to avoid negative heads further along the line. The actual permissible head drop should be determined from the pipeline profile. Typically, the head sag curve is convex down and the pipeline profile convex up, so a head loss of H_0 / γ is more typical. Then solving for D_e / D_p

$$D_e / D_p = (\gamma V_p^2 / g H_0)^{1/4} \quad (11)$$

Generally, it will be found that the exit diameter $D_e = 0.70 - 0.8$ times the pipe diameter D_p , e.g.,

$$D_e / D_p = (\gamma V_p^2 / g H_0)^{1/4} = 0.70 \quad (12)$$

This value is probably a lower limit for outlet pipe size as generally minimum heads along the line are critical so the outlet should be bigger. The value is not as critical as the inlet diameter. To avoid abrasive damage to the air vessel connector, V_e should be less than approximately 10 m/s . An example is appended illustrating the calculations.

b) INLET

The return surge (i.e., the reversal of the water column in the main) compresses the air in the air vessel, gradually decelerating the water column. The maximum compression of the air, which coincides with the maximum pressure in the pipeline, occurs when the water column has stopped. On the other hand, the maximum head loss into the air vessel occurs when the water column is at maximum velocity (i.e., it is out of phase with gas compression), so it can be designed to be a maximum, subject to not exceeding the gas compression head.

This head loss can be further utilized to decelerate the water column, requiring less air to cushion the surge at its furthest extent.

It has been found that the inlet to the air vessel can be relatively small, as most of the water column's energy is expended in outlet and line losses, and overpressures are typically not as problematic as negative pressures. Stephenson (1989) suggested an inlet size $1/10$ of the main diameter, and Parmakian (1962) suggested an inlet head loss 2.0 times the outlet head loss. Thorley (1991) used Graze and Horlacher's results to plot results of various inlet sizes.

As a general rule, additional air vessel volume is not needed if the vessel is greater than two times the air volume, as the pressure is then halved and reverse flow is likely.

For example, a simplistic calculation of inlet diameter can be made assuming the air volume at full expansion is twice the operating volume and the velocity attenuation in the outflow is due to the head loss.

From

$$H = -L/g \cdot dV/dt \quad (1)$$

$$dV = gh_{min}T/\gamma L \quad (12)$$

Since the average decelerating head would be h_{min}/γ . The time T over which the deceleration occurs is related to the air expansion volume.

If

$$S = \gamma S_0 = \gamma/Q_0 \cdot T \quad (14)$$

Then

$$T = \gamma S/Q_0 = \epsilon S_0/Q_0 \quad (15)$$

Where

S=air volume, S_0 =initial volume, and Q_0 =initial flow rate.

During this time, $dV = gh_{min} \gamma S_0/Q_0 \cdot L$. So maximum return velocity in the main is

$$V_r = V_0 - \gamma gh_{min} S_0/Q_0 \cdot L \quad (16)$$

And limiting the maximum head rise to $0.1H_0$ for this example

$$H_{max} = 0.1H_0 = \gamma V_0^2 / \gamma g (D_p/D_i)^{\epsilon} \quad (17)$$

$$D_i/D_p = [1 \cdot V_0^2 / H_0 \cdot g]^{\epsilon} = (1 \cdot / H_0 \cdot g)^{\epsilon} (V_0 - \gamma gh_{min} S_0/Q_0 \cdot L)^{2\epsilon} \quad (18)$$

or if h_{max} is a variable,

$$T = \gamma S/Q_0 \quad (19)$$

$$dV = gh_{max} S/LQ_0 \quad (20)$$

$$S = ALV_0^2 / \gamma g (h_{max}/\gamma) \quad (21)$$

Then

$$D_i/D_p = (V_0^2 / gh_{max})^{\epsilon} \cdot (1 / (\gamma^{\epsilon} / \gamma)) \quad (22)$$

Which is plotted in fig. (1.1V)

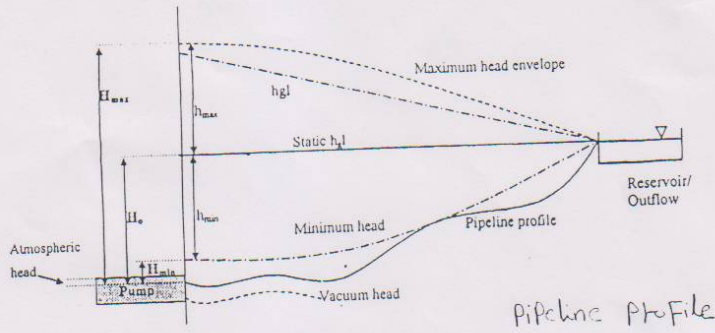


Fig.(1.13)

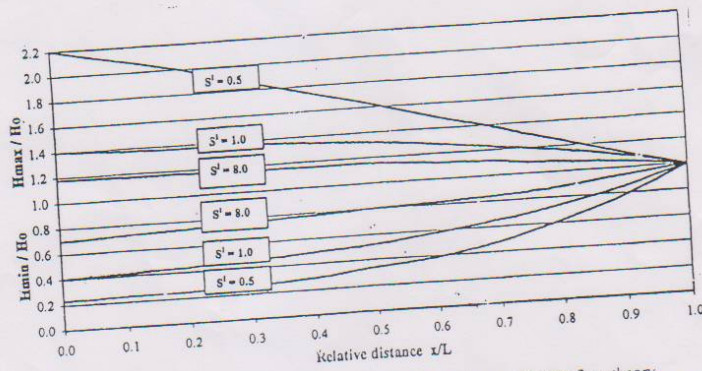


Fig. 1.14 Maximum and minimum head envelopes using incompressible flow theory

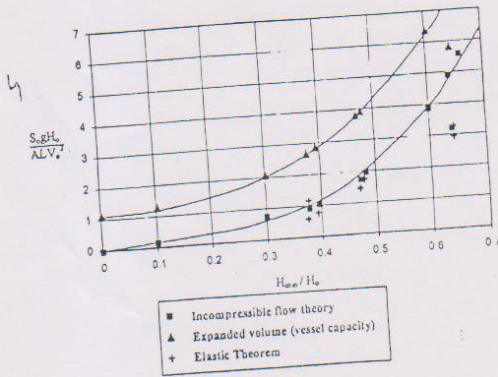


Fig (1.15)

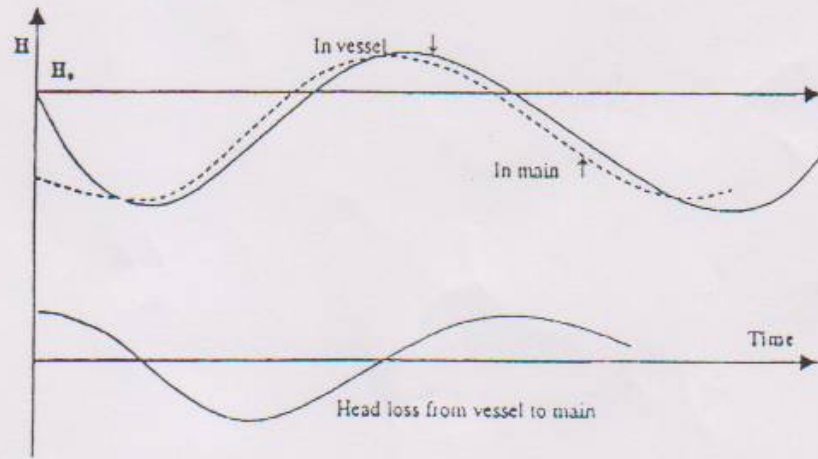


Fig.16. Heads in vessel and main after pump trip

Fig. (1-16)

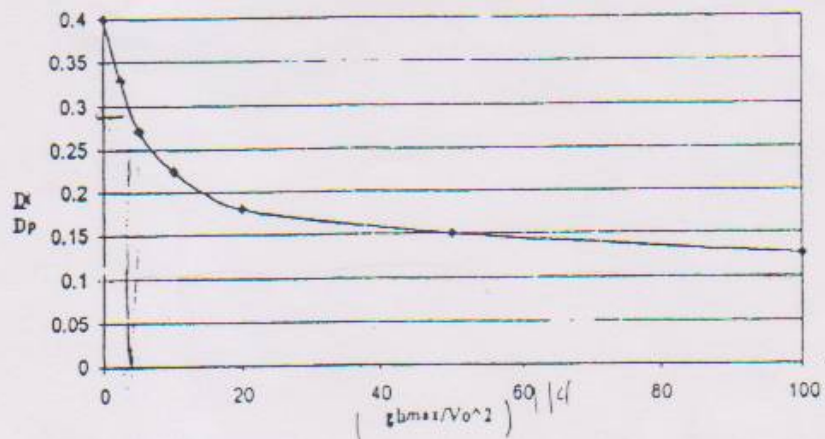


Fig (1.1V)

Example

A pumping pipeline, 900 mm in diameter, 18,000 m long, with a static head of 210 m, conveys water at an initial velocity of 1.5 m/s.

The air vessel characteristics are calculated below to limit minimum head to 20% of the static head, and maximum to 20% above static, neglecting friction.

Air vessel volume

From fig. (1.12) $S' = 1.0$

$$S' = S \cdot H \cdot g / ALV \cdot v^2$$

$$S = 1 \cdot \pi / (2 \cdot 9.81^2 \cdot 18000 \cdot 1.5^2 / (9.81 \cdot 210)) = 0.6 \text{ m}^3$$

From

Fig. (1.10), at $h_{\min}/H = 0.2 \cdot 210 / 210 = 0.2$

$$S \cdot H \cdot g / ALV \cdot v^2 = 3$$

$$S = 3 \cdot \pi / (2 \cdot 9.81^2 \cdot 18000 \cdot 1.5^2 / (210 \cdot 9.81)) = 16.8 \text{ m}^3$$

Outlet pipe diameter

From equation (9)

$$D_e/D_p = (v \cdot v^2 / 2gh_{\min})^{.25}$$

$$D_e = .9(3 \cdot 1.5^2 / 2 \cdot 9.81 \cdot .6 \cdot 210)^{.25} = .10 \text{ m}$$

This is rather small and would result in a theoretical water velocity of nearly (20 m/sec), so a compromise large diameter, e.g., 700 mm may be used $D_e = .70 \text{ m}$

Inlet pipe diameter

From fig. (1.14),

$$\text{at } g \cdot h_{\max} / V \cdot \Delta t = 9.81 \cdot \Sigma \cdot \Sigma 10 / 1.5 \Delta t = 816$$

$$D_i / D_p = 1$$

$$D_i = 1 \cdot 9 = 0.9 \text{ m}$$

take $D_i = 1 \text{ m} = 1000 \text{ mm}$

1.8 CONCLUSIONS

Water hammer refers to fluctuations caused by sudden increase or decrease in flow velocity. These pressure fluctuations can be severe enough to a rupture water main. Potential water hammer problems should be considered when pipeline design is evaluated, and a surge analysis should be undertaken, in many instances, to avoid costly malfunctions in distribution system. Every major system design change or operation change—such as the demand for higher flow rates—should include consideration of potential water hammer problems. This phenomenon and its significance to both the design and operation of water systems is not widely understood, as evidenced by the number and frequency of failures caused by water hammer.

Water hammer cannot be completely eliminated in an economical design, but, by taking precautions during management and operation, the effects can be minimized. Start-up is critical, especially when pipe lines are empty. Empty lines should be filled as slowly as possible to allow entrapped air to escape. In addition the following cautions should be observed.

1. To help minimize surge pressures the maximum velocity of water in the pipeline should be limited. General recommendations are to limit maximum operating velocities to 1.0 m/sec. In no case should the velocity exceed 3.0 m/sec.
2. Make sure that all air has been discharged from the system before operating the system at full throttle.
3. Close all manual valves slowly. No valve should ever be closed in less than 10 seconds; 30 seconds or more is preferable.
4. Install a non-return valve on a bypass pipe connected in parallel to the delivery pipeline and the other end connected to the suction pipe to the suction well.

- ο. Study the pipeline profile along with the piezometric line, and if the distance between them is practically small at some locations then consider the installation of surge tanks to reduce water hammer effect.
- ϒ. Use the same precautions in stopping the irrigation system as used in start-up and general operation.

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