Chapter 9

WATER DISTRIBUTION

9.1 Surface Water Supply

Surface water supplies are not as reliable as groundwater sources since quantities often fluctuate widely during the course of a year or even a week, and water quality is affected by pollution sources. If a river has an average flow of 10 cubic feet per second (cfs), this does not mean that a community using the water supply can depend on having 10 cfs available at all times.

The variation in flow may be so great that even a small demand cannot be met during dry periods, and storage facilities must be built to save water during wetter periods. Reservoirs should be large enough to provide dependable supplies. However, reservoirs are expensive and, if they are unnecessarily large, represent a waste of community resources.

One method of estimating the proper reservoir size is use of a mass curve to calculate historical storage requirements and then to calculate risk and cost using statistics. Historical storage requirements are determined by summing the total flow in a stream at the location of the proposed reservoir, and plotting the change of total flow with time. The change of water demand with time is then plotted on the same curve. The difference between the total water flowing in and the water demanded is the quantity that the reservoir must hold if the demand is to be met. The method is illustrated by Example 9.1.

Example 9.1: A reservoir is needed to provide a constant flow of 15 cfs. The monthly stream flow records, in total cubic feet, are

Month	J	F	M	A	M	J	J	A	S	O	N	D
Million ft^3 of water	50	60	70	40	32	20	50	80	10	50	60	80

The storage requirement is calculated by plotting the cumulative stream flow as in Figure 9.1. Note that the graph shows 50 million ft^3 for January, 60 + 50 = 110 million ft^3 for February, 70 + 110 million ft^3 for March, and so on.

Solution:

The demand for water is constant at 15 cfs, or

$$15\times10^6\,\frac{ft^3}{s}\times3600\,\frac{s}{h}\times24\,\frac{h}{day}\times30\,\frac{days}{month}=38.8\times10^6\,\frac{ft^3}{month}$$

This constant demand is represented in Figure. 9.1 as a straight line with a slope of 38.8×10^6 ft^3/month, and is plotted on the curved supply line. Note that the stream flow in May was lower than the demand, and this was the start of a drought lasting until June. In July the supply increased until the reservoir could be filled up again, late in August. During this period the reservoir had to make up the difference between demand and supply, and the capacity needed for this time was 60×10^6 ft^3. A second drought, from September to November required 35×10^6 ft^3 of capacity. The municipality therefore needs a reservoir with a capacity of 60×10^6 ft^3 to draw water from throughout the year.

A mass curve like Figure. 9.1 is not very useful if only limited stream flow data are available. Data for one year yield very little information about long-term variations. The data in Example 9.1 do not indicate whether the 60 million cfs deficit was the worst drought in 20 years, or an average annual drought, or occurred during an unusually wet year.

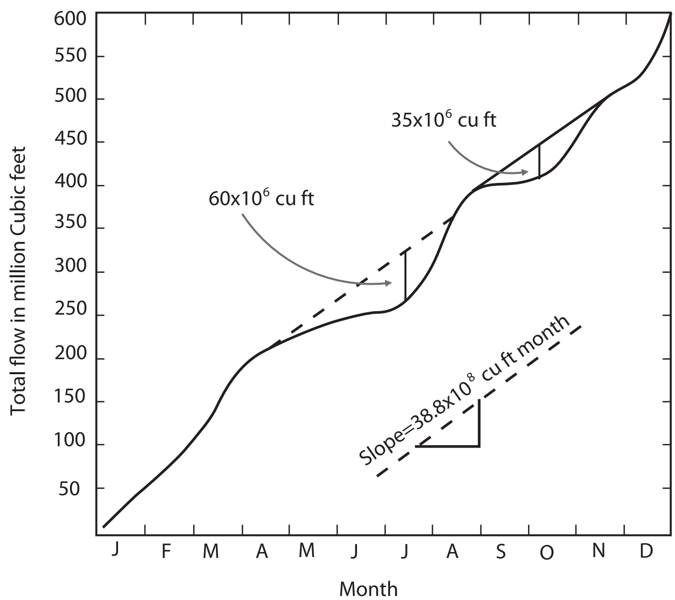


Figure 9.1: **Mass curve for determining required reservoir capacity**

Long-term variations may be estimated statistically when actual data are not available. Water supplies are often designed to meet demands of 20-year cycles, and about once in 20 years the reservoir capacity will not be adequate to offset the drought. The community may choose to build a larger reservoir that will prove inadequate only every 50 years, for example. A calculation comparing the additional capital investment to the added benefit of increased water supply will assist in making such a decision. One calculation method requires first assembling required reservoir capacity data for a number of years, ranking these data according to the drought severity, and calculating the drought probability for each year. If the data are assembled for n years and the rank is designated by m, with m = 1 for the largest reservoir requirement during the most severe drought, the probability that the supply will be adequate for any year is given by $m/(n+1)$. For example, if storage capacity will be inadequate, on the average, one year out of every 20 years,

$$m/(n+1) = 1/20 = .05$$

If storage capacity will be inadequate, on the average, one year out of every 100 years,

$$m/(n+1) = 1/100 = .01$$

The calculation of storage is illustrated in Example 9.2.

Example 9.2: A reservoir is needed to supply water demand for 9 out of 10 years. The required reservoir capacities, which were determined by the method of Example 9.1, are shown below:

Year	Required reservoir capacity ($m^3 \times 10^6$)	Year	Required reservoir capacity ($m^3 \times 10^6$)
1961	60	1971	53
1962	40	1972	62
1963	85	1973	73
1964	30	1974	80
1965	67	1975	50
1966	46	1976	38
1967	60	1977	34
1968	42	1978	28
1969	90	1979	40
1970	51	1980	45

These data must now be ranked, with the highest required capacity, or worst drought, getting rank 1, the next highest 2, and so on. Data were collected for 20 years, so that $n = 20$ and $n + 1 = 21$. Next, $m/(n + 1)$ is calculated for each drought

Rank	Capacity ($m^3 \times 10^3$)	m/(n+1)	Rank	Capacity ($m^3 \times 10^3$)	m/(n+1)
1	90	0.05	11	50	0.52
2	85	0.1	12	46	0.57
3	80	0.14	13	45	0.62
4	73	0.19	14	42	0.67
5	67	0.24	15	40	0.71
6	62	0.29	16	40	0.76
7	60	0.33	17	38	0.81
8	60	0.38	18	34	0.86
9	53	0.43	19	30	0.90
10	51	0.48	20	28	0.95

These data are plotted in Figure. 9.2 a semi log plot often yields an acceptable straight line. If the reservoir capacity is required to be adequate 9 years out of 10, it may be inadequate 1 year out of 10. Entering Figure 9.2 at $m/(n + 1) = 1/10 = 0.1$, we find that

$$m/(n+1) = 2/21 = 0.1 \quad .$$

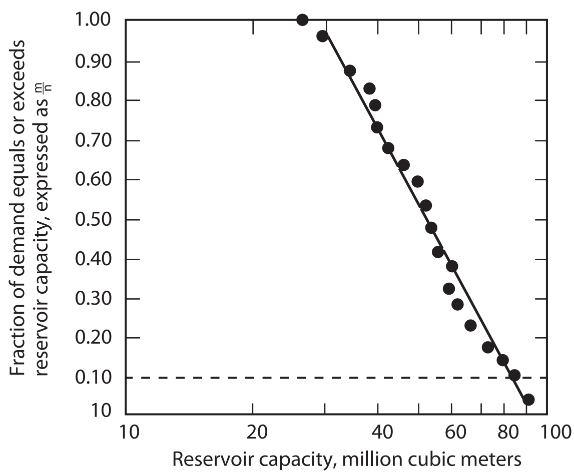


Figure 9.2: Frequency analysis of reservoir capacity

The 10% probability of adequate capacity requires a reservoir capacity of 82 million m^3. Had the community only required adequate capacity 1 year out of 5, $m/(n + 1) = 0.2$ and, from Figure 9.2, a reservoir capacity of 71 million m^3 would have sufficed.

This procedure is a frequency analysis of a recurring natural event. The frequencies chosen for investigation were once in 10 years and once in 5 years, or a "10-year drought" and a "5-year drought," but droughts occurring 3 years in a row and then not again for 30 years still constitute "10-year droughts." Planning for a 10-year recurrence interval, though usually reliable, is not absolute.

9.2 Intakes

Surface sources of water are subject to wide variations in flow, quality, and temperature, and intake Structures must be designed so that the required flow can be withdrawn despite these natural fluctuations. The intake itself normally consists of an opening (frequently screened in some manner) and a conduit which conveys the flow to a sump from which it may he pumped to the treatment plant. In locating intakes, one must consider anticipated variations in water level, navigation requirements, local currents and patterns of sediment deposition and scour, spatial and temporal variations in water quality, and the quantity of floating debris.

Impounding reservoirs are subject to rather wide variations in depth and thus require intake Structures which will permit withdrawal over a wide range of elevations. It is normally not satisfactory to locate a single inlet at the bottom, since the water quality in reservoirs varies with both time and depth. The quality is usually best close to the surface, although this may not be true for brief periods in spring and fall when overturns may occur.

The lower levels of deep impoundments are normally cool and change very little in temperature during the year. The surface, on the other hand, varies in temperature with the air and, during most of the year, is warmer than the lower levels. The water temperature decreases slowly with depth until, at some level where wind-driven mixing Currents become ineffective, it decreases rapidly to the uniform bottom temperature in a short vertical distance. This region of rapid temperature change is called the thermocline. In the fall, as the air temperature decreases, the surface layers will cool and sink, displacing the lower layers and driving them to the surface. A similar phenomenon may occur in spring as the water from thawing ice reaches its maximum density and sinks toward the bottom. The water at the bottom of an impoundment is normally low in dissolved oxygen and high in organic matter. It is desirable to avoid drawing this water into the intake; hence the optimum elevation for withdrawal is likely to change during the year.

A typical reservoir intake designed for an impoundment with an earthen dam is illustrated in Figure 9.3. In concrete or masonry dams the intake may be constructed in the dam itself.

Lake intakes should be located as far as possible from sources of pollution, and one should consider wind and current effects on the motion of contaminants. In particular, winds may stir up sediment from the bottom which may be carried to the intake if it is located in shallow water or too close to the bottom. Inlet velocities should be less than 0.15 m/s (0.5 ft/s) to avoid trapping excessive quantities of floating material, sediment, ice, or fish. A water depth of

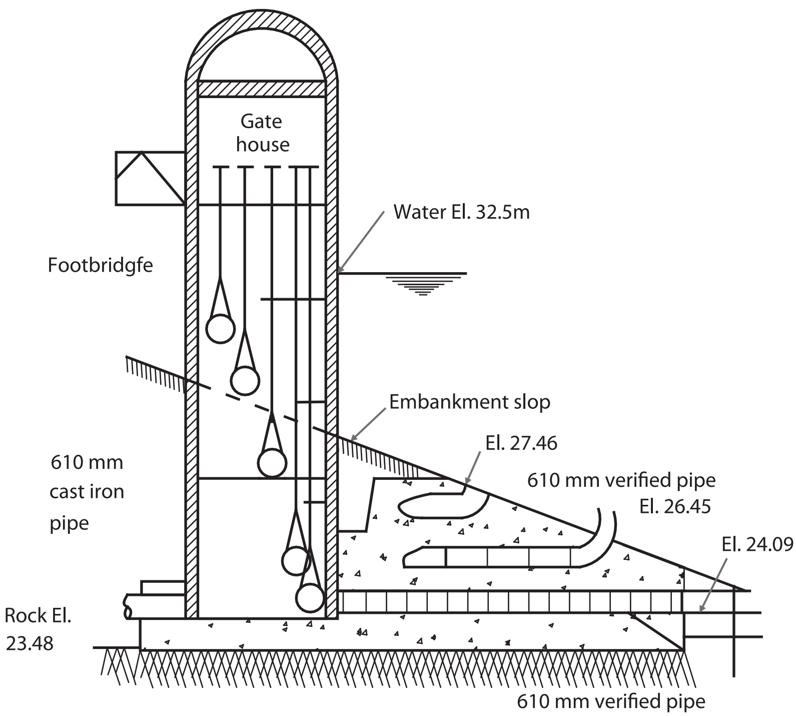


Figure 9.3: **Typical reservoir intake**

6 to 9 m (20 to 30 ft) is necessary to prevent blocking of the intake by ice jams which may fill the lake to the bottom in shallower depths. Anchor ice may form beneath the surface on screens, gates, and valves which are cooled below the water temperature by conduction through connecting appurtenances exposed to air inside the intake structure. Frazil ice crystallizes within the water in the form of needlelike masses which may adhere to anchor ice already formed in the intake or which may plug screens. Accumulations of ice have been removed by forcing compressed air through the blocked openings, but this technique is not always successful.

Preventive measures generally involve heating the screens and the air within the structure to keep them at a temperature slightly above the freezing point of water. Figure 9.4 shows a typical lake intake.

Submerged cribs (Figure 9.5) are used by smaller communities. Their depth is dictated in part by navigation requirements and in part by the need to locate intake above high concentrations of sediment which may be suspended by wind action. If entrance velocities are low, no regular attendance is necessary.

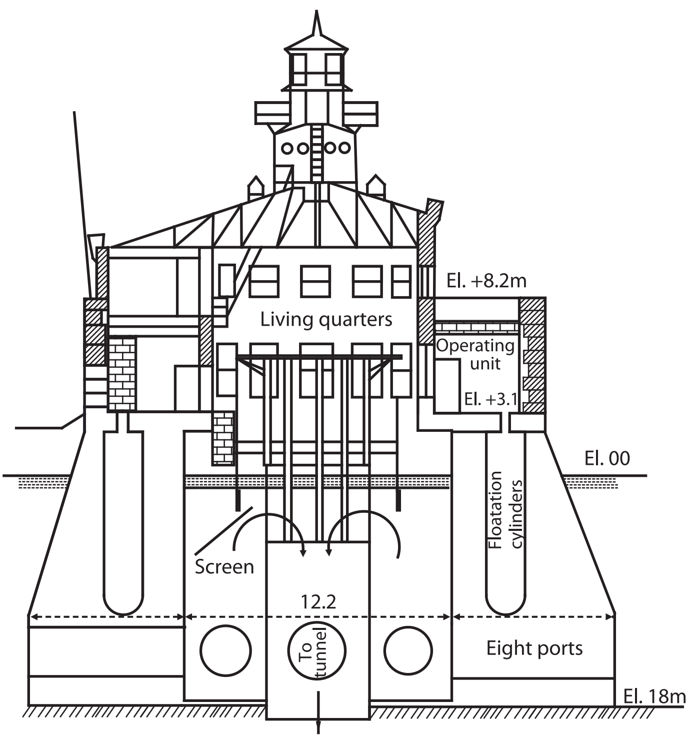


Figure 9.4: **Typical Lake Intake**

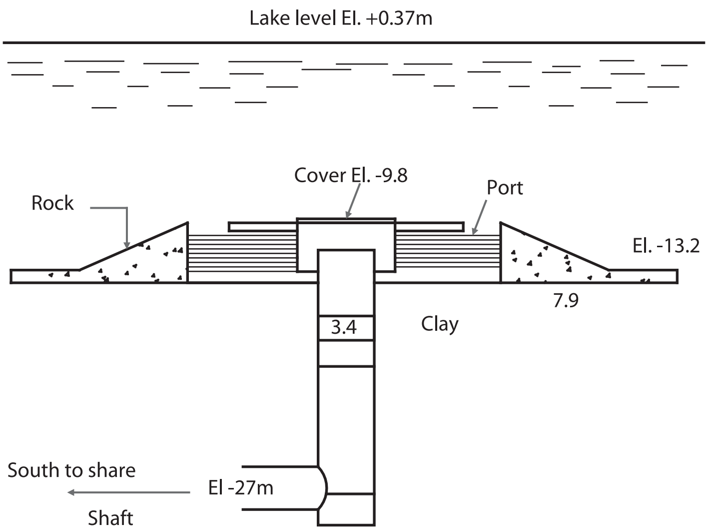


Figure 9.5: **Typical submerged crib intake**

The example shown consists of a wooden crib protected by riprap. Pipe intakes have also been used by small cities. This type of structure may consist only of a pipe projecting into the water with a screened opening on the end. More elaborate pipe intakes may be supported above the bottom on piers or pile bents. Multiple screened inlets may by connect to a single pipe to reduce inlet velocities. An intake of this sort is shown in Figure 9.6.

River intakes should be designed, when possible, to withdraw water from slightly below the surface in order to avoid both sediment in suspension at lower levels and floating debris. Some large cities, notably St. Louis and Cincinnati, have built elaborate river intakes resembling bridge piers with ports at various depths to accommodate variations in water level. Small cities may use simple pipe intakes located so that they are sufficiently below the low-water level that river traffic is not impeded. Such intakes must also be above the bottom so that materials being carried in traction will not cover them. These requirements often dictate that the intake opening be in the main channel, which may be quite far from the normal bank

Shore intakes may be constructed when the main channel of the river is at or near the bank. Low diversion dams may be built upstream to ensure that the total flow goes past the inlet during low river stages, thus minimizing silting of the channel.

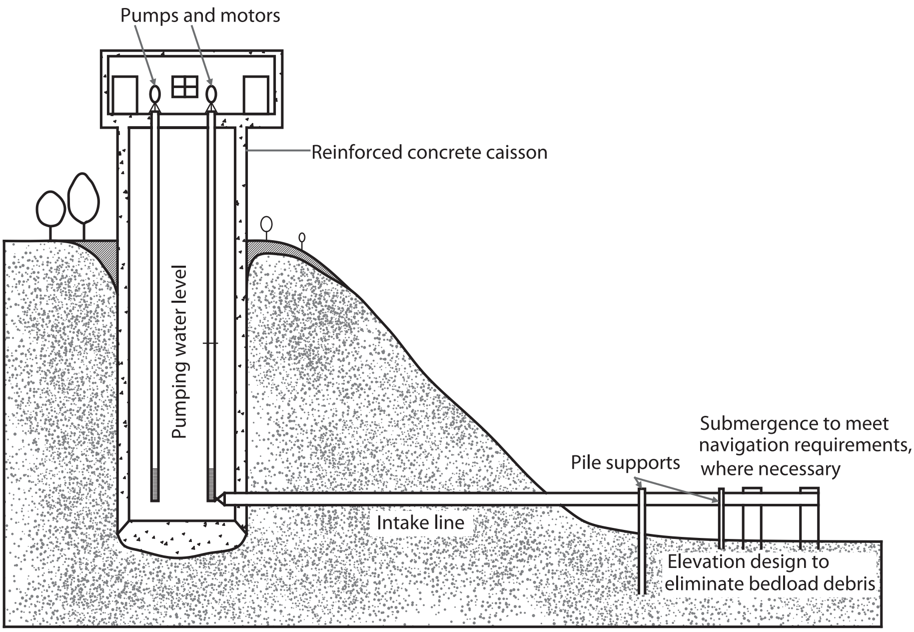


Figure 9.6: **Screened pipe intake.**

Screens are particularly desirable in river intakes, since large quantities of suspended material might otherwise enter the structure, in some areas, water clogged plant material may be carried below the surface and be drawn into the inlet—as may fish which venture too close. Automatically cleaned bar screens or movable fine screens are frequently necessary to prevent this material from clogging pumps.

Intakes which are located far from the shore of either a river or take deliver the flow to a pipe conduit buried below the bottom. Velocities in the conduit must be sufficiently great to prevent deposition of sediment, normally in the range of 0.3 to 0.6 m/s (1 to 2 ft/s). The conduit terminates in a sump or well from which the flow is pumped to the treatment plant. Figure 9.7 shows a typical Lake Intake, conduit, and pumping station. Where rivers are contained by protective levees, the pumping station must be located within the flood plain. The pumped flow then passes over the top of the levee.

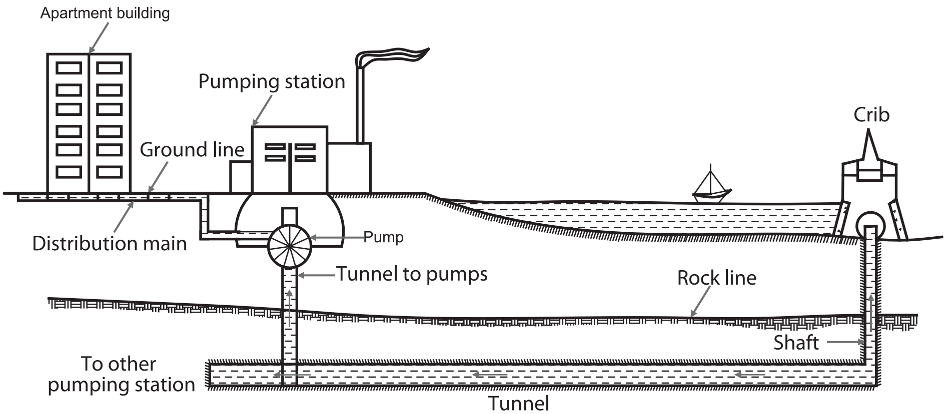


Figure 9.7: **Typical intake, conduit, and pumping station.**

9.2.1 Design of intakes and screens

The hydraulic considerations in intake structure design are energy losses due to acceleration and deceleration of water at bar racks, intake ports, and fine screens. The low velocities of a properly designed intake typically make these losses very small. The losses through the intake port can be calculated by using the orifice equ^n. The orifice area is submerged effective gate area.

$$H_L = \frac{1}{2g}\left(\frac{V}{C_d}\right)^2$$

Here,

H_L = Head loss, m (ft)
g = Acceleration due to gravity, m/s^2 (ft/s^2)
V = velocity, m/s
C_d = Coeff. of discharge for orifice (0.6 – 0.9)
A = Effective submerged orifice area, m^2 (ft^2)

Loss through screens for clean/dirty screens,

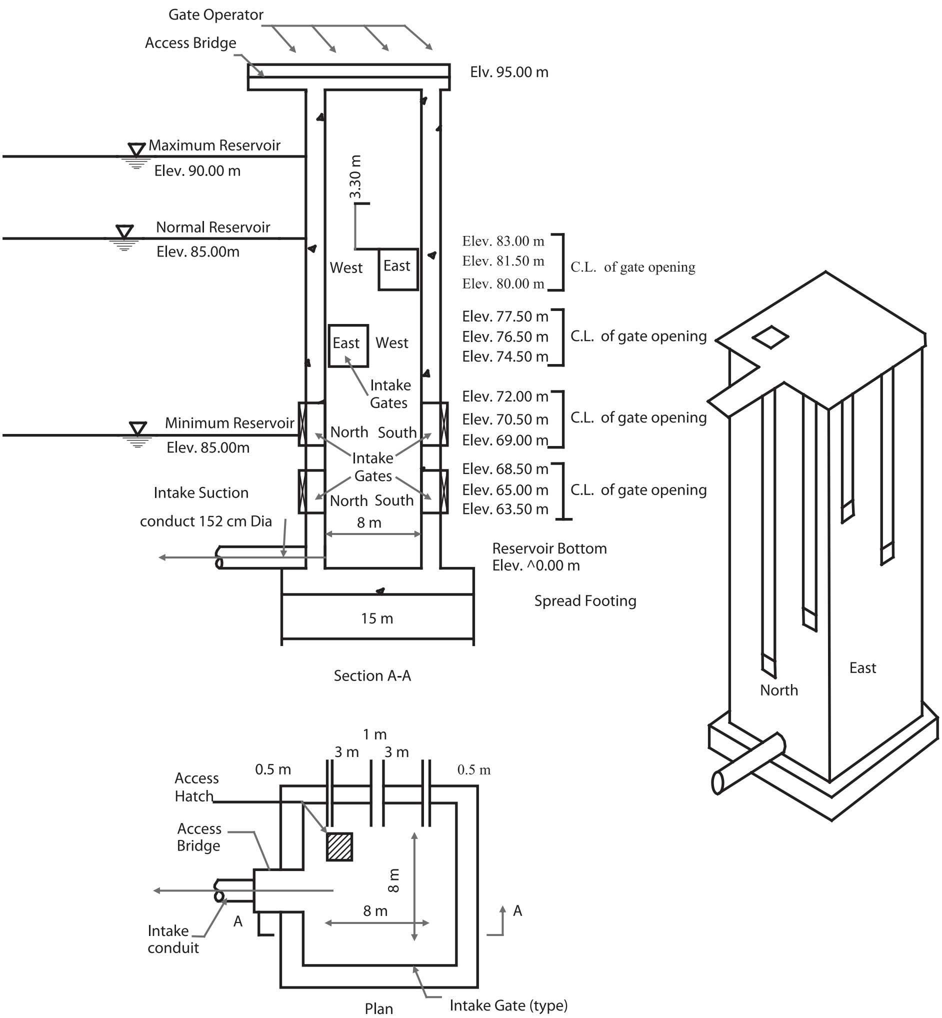
$$H_L = \frac{(v^2 - V_v^2)}{2g}\left(\frac{1}{0.7}\right)$$

Here, H_L = Head loss through screen, m (ft)
v = velocity through screen opening, m/s (f/s)
V_v = Velocity upstream of screen (zero in most case)
g = Acceleration due to gravity, m/s^2 (ft/s^2)

Example 9.3: The following design criteria are used to develop design of raw water intake and screening facility:

1) Hydraulic data.

a. Maximum day demand = 113,500 m^3/d
b. Minimum reservoir elevation = 70m (mean sea level)
c. Max^m reservoir elevation = 90m (msl)
d. Normal surface water elevation = 85.00m (msl)
e. Bottom elevation = 60m (msl)



2) Headwork location

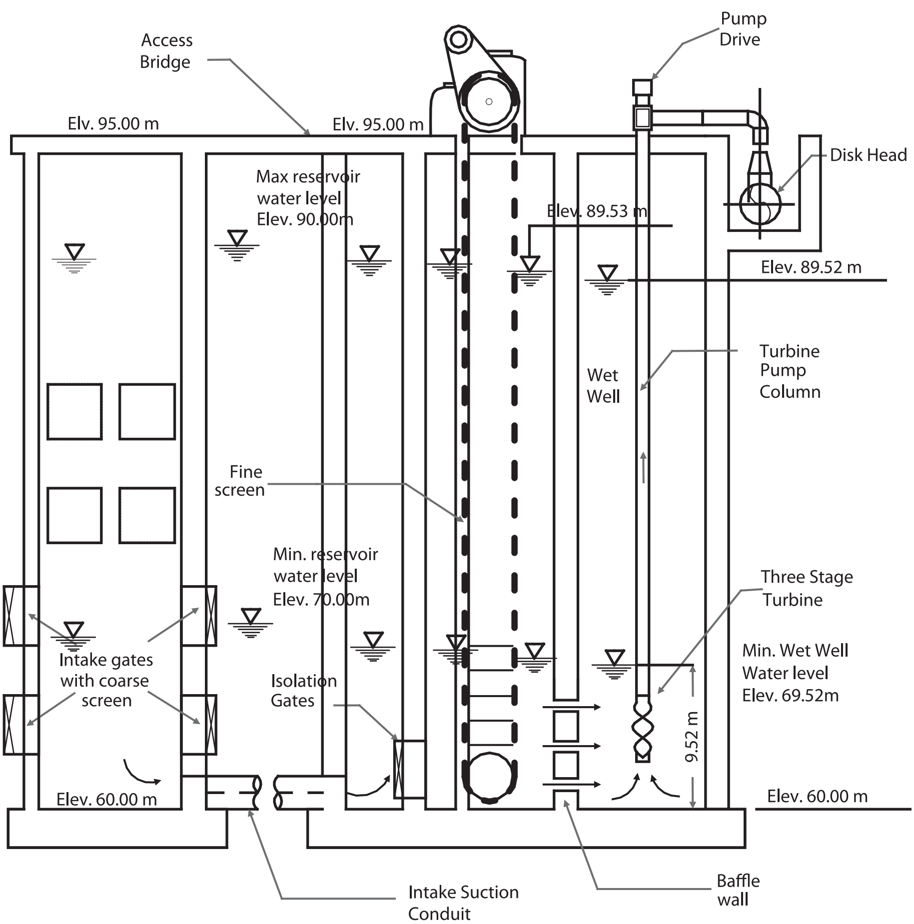
Raw water intake shall be located in the lake such that it is connected by a bridge to the on shore pump station.

3) General guidelines:

a. Raw water intake shall be located in the lake such that it will withdraw water from max^m and min^m elevations of 90.0m and 70.0m.

b. A coarse screen shall be provided at gate, velocity should be less than 8cm/s (0.3 ft/s).

c. Fine screen shall be provided at pump station, velocity shall be less than 0.2 m/s (0.6 ft/s). Intake will be dry tower.



Solution:

Step 1: Design of intake structure

a) Select size of intake gate

The gates are sized such that the entire maximum flow 113,500 m^3/d can be withdrawn from a single level at a max^m velocity 0.08m/s. withdrawn water should be best water quality.

$$Q = 113{,}500\ m^3/d = 1.31\ m^3/s$$

$$A = \frac{1.31 m^3/s}{0.08 m/s} = 16.38\ m^2$$

This is too large for a single gate, so select two equal size square gates.

$$\text{Width} = \left(\frac{16.38m^2}{2}\right)^{½} = 2.86\text{m}$$

Select standard size. So selected gate size is

Width and height = 3.00m

$$\text{Velocity through gate} = \frac{1.31 m^3/s}{3m \times 3m} = 0.07\ \text{m/s}$$

b) Determine layout of intake gates

The highest gate with its top 2m below normal water surface elevation (85m) is = (85-2) m = 83m

And centerline elevation = (83-1.50) m = 81.50

Set the lowest gate at a center line elevation of 65.0m. So bottom of the gate is (65 – 1.5 – 60.0) m = 3.5m

In order to provide flexibility to withdraw water from intermediate locations, provide additional gates at two levels equally spaced over range: (81.50-65) m = 16.5m

$$\text{Spacing} = \frac{16.5m}{3\ spaces} = 5.5\text{m / space}$$

So, gates will be provided at centerline elevations of 81.50, 76.0, 70.5, 65.0m.

Locate 2 gates on each side of the intake tower. Each gate is slightly more than 3m wide so, set width of the tower = 8m

This will provide approximately 0.5m between the gate and the wall and 1m between gates.

Select dimension of the tower = 10m x 10m

Intake – gate Layout

No.	Location	Centerline elevation, m
1	East	81.5
2	West	81.5
3	East	76.0
4	West	76.0
5	North	70.5
6	South	70.5
7	North	65.0
8	South	65.0

Step c) Design of Coarse Screen

- **Select layout of coarse screen**
  The coarse screen will be located at intake ports, slightly projected away from intake gate to prevent debris from interfering gate operation.

- **Select bar arrangement**
  Use 13mm (0.5") square bars, 4.8m long, spaced 8 cm on centers. The bar covers 3.6m over the gate.

  $$\text{Number of space} = \frac{360\ cm}{8cm\,/\,spaces}$$

  $$= 45 \text{ Spaces}$$

  $$\text{The number of bars} = 45 - 1$$
  $$= 44 \text{ Spaces}$$

- **Velocity through bar rack**

  $$\text{Rack area} = (3.6 \times 4.8)\ m^2 = 17.28\ m^2$$

  $$\text{Bar area} = 44 \text{ bars} \times 0.013m \times 4.8m = 2.75\ m^2$$

  $$\text{Open area} = (17.28 - 2.75)\ m^2 = 14.53\ m^2$$

Maximum, flow through the rack is half of the design flow as there are 2 gates at each level.

$$\text{Velocity} = \frac{1.31\,m^3/s}{2} \times \frac{1}{14.53\,m^2} = 0.0451\text{ m/s}$$

Step D) Design of fine screen

- **Select location of fine screen**
  Fine screens are located at pump station to provide better access for maintenance.

- **Select screen equipment**
  Provide two fine screens, each with 9.5 mm opening. From figure, the depth of flow at the screen chamber is 9.53 m at minimum reservoir level of 70m.

  Assume maximum velocity = 0.2 m/s and screen efficiency 0.56 for stainless steel screens.

- **Calculate Screen width**

$$\text{Width} = \frac{0.66\,m^3/s}{9.53m \times 0.2m/s \times 0.56} = 0.62\text{m}$$

$$\text{Select smallest the screen} = \frac{0.66\,m^3/s}{9.53m \times 0.9m/s \times 0.56}$$
$$= 0.14\text{ m/s}$$

So, velocity through a screen will be 0.14 m/s.

9.3 Water Distribution: Terminology

The rest of the chapter describes criteria for design and construction of potable water distribution systems. Within the context, a water distribution system is considered to consist of all mains, service lines, valves, pumps, hydrants, and ancillary equipment needed to carry water from the source of potable water to the various points of use.

- **Backflow:** The flow of any foreign liquids, gases, or other substances into the distributing pipelines of a potable supply of water from any source or sources not intended.

- **Back-siphonage:** The backing up, or siphoning, of a foreign liquid into a potable water system; this occurs when the potable water system, at any point or place, is at a pressure less than atmospheric, with an opening or break in the system, thereby drawing the foreign liquid toward the potable water.
- **Cross connection:** Any physical connection which provides an opportunity for nonpotable water to contaminate potable water.
- **Distribution mains:** All pipelines of the distribution system, except the small service pipes connecting building systems to the supply.
- **Transmission mains:** Those pipelines or conduits which carry water from one point to another without intermediate service connections; e.g., pipelines from a pumping station to a reservoir.

9.4 Methods of Distribution

Water may be distributed by gravity, by pumps alone, or by pumps in Conjunct ion with on-line storage. Gravity distribution is possible only when the source of supply is located substantially above the level of the city. This is the most dependable technique, provided there are multiple well-protected conduits carrying the flow to the community. High pressure for fire fighting may require use of motor pumping trucks, and low-lying areas may need to be isolated to prevent excessive pressure.

Pumping without storage is the least desirable method of distribution, since it provides no reserve flow in the event of power failure and pressures will fluctuate substantially with variations in flow. Since the flow must be constantly varied to match an unpredictable demand, sophisticated control systems are required. Peak water use and thus peak power consumption are likely to coincide with periods of already high power use, increasing power costs. Systems of this kind have the advantage of permitting increased pressure for fire fighting, although individual users must then be protected by pressure reducing valves.

Pumping with storage is the most common method of distribution. Water is pumped at a more or less uniform rate, with flow in excess of consumption being stored in elevated storage tanks distributed throughout the system. During periods of high demand, the stored water augments the pumped flow, thus helping to equalize the pumping rate and to maintain more uniform pressure in the system. It may be economical, in some cases, to pump only during off-peak hours to minimize power costs.

The stored water provides a reserve for fire flow and ensures a reliable general-purpose flow when power fails. Motor pumpers may be used to provide the pressure necessary for fire fighting or a fire pump may be operated at the pumping plant. The latter technique has the same drawbacks as in pumping without storage; in addition, it requires that the storage be isolated from the system when the fire pump is run.

Water is stored to equalize pumping rates in the short term, to equalize supply and demand in the long term, and to furnish water during emergencies such as fires and loss of pumping capacity.

Elevated storage may be provided by earthen, steel, or concrete reservoirs located on high ground or by standpipes or tanks raised above the ground surface. Standpipes are cylindrical structures, usually of relatively small diameter. The amount of water available for fire protection from a standpipe is that volume above the level which provides a residual pressure of 140 kPa (20 lb/in^2) at the fire pumps. Elevated tanks (Figure.9.8) are designed and constructed of steel in capacities up to 15,000 m^3 (4×10^6 gal) by firms which specialize in this work. In large systems, a number of elevated tanks may be located a points selected to minimize pressure variations during periods of high consumption.

Normally, elevated storage is located so that zones of high consumption lie between the pumping station and the tanks as shown in Figure. 9.9. During periods of high use, the district will be fed from both sides, which reduces the pressure drop to about one-quarter that which would exist if flow were only from one direction.

The capacity of the elevated storage tanks depends upon the flow variations expected in the system. Equalization of the pumping rate that is, provision of sufficient capacity to permit pumping at a constant rate, normally requires storage equal to 15 to 30 percent of the maximum daily use. Figure 9.10 shows the typical pattern of water use on the maximum day for a community which has an average rate of use on the peak day is 41.5×10^3 m^3/day, if it is possible to pump at the average rather than the peak rate, the pumping station will be smaller and energy costs will be lower. In order to equalize the rate, the water pumped when use is less than average must be stored for use when the rate is higher than average. The cross-hatched area above the average rate is equal to that below and

Figure 9.8: Elevated steel tank

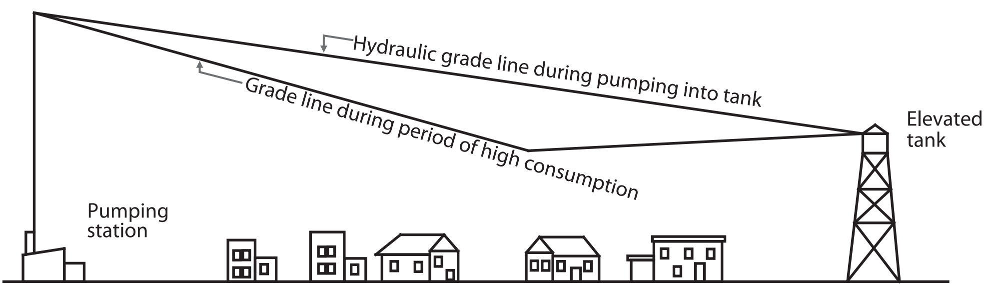


Figure 9.9: Effect of elevated storage on pressure

either represents the storage volume required. The calculated amount must be increased to provide for future growth in population.

Additional storage beyond that necessary to equalize pumping may be required to provide for fire protection. The insurance services office, which bases

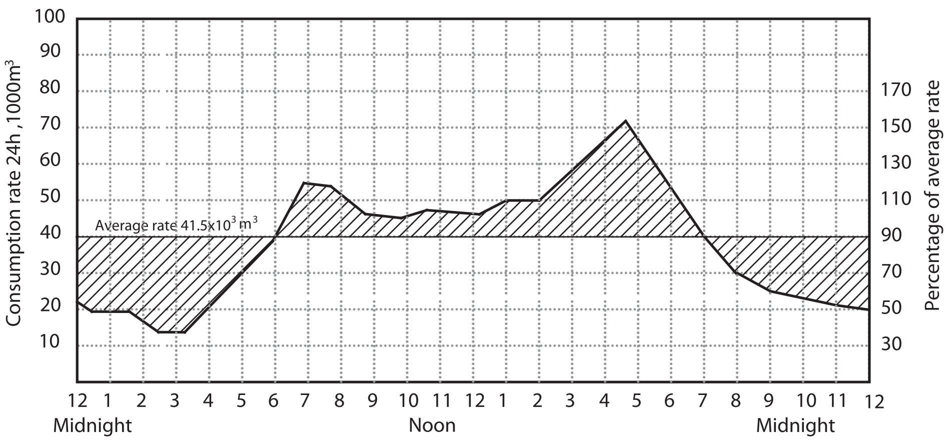


Figure 9.10: Diurnal variation in water consumption

insurance rates partly on the level of fire protection, considers a water supply adequate if it can furnish the required fire flow in addition to the average consumption on the maximum day. The required fire flow may be entirely pumped or may be provided by a combination of pumping and storage in excess of that required equalizing normal demand.

Storage may also be necessary to equalize demand over a lengthy period of high use, such as a cold period in winter or a dry period in summer. Storage of this type is particularly desirable when the source or treatment plant is limited in capacity. The required storage can be determined only from the study of extensive records of water consumption. Data from periods of high use are used to construct mass diagrams from which the required storage is obtained. The calculated amount must be increased to provide for future growth. When flow data are not available, the Goodrich formula may be used.

Elevated storage tanks in areas of high Consumption and low pressure will increase pressure during periods of peak use without increasing the size of the water mains. The tanks fill during the night when consumption is low and pressure is high. When use is high, the tanks provide water to the system and maintain the pressure in their vicinity. Elevated tanks are commonly provided with automatic valves which close when the tank is full and open when the pressure in the mains falls below that at the bottom of the tank.

9.5 System Planning

The distribution system must reliably and economically supply water, in adequate quantities and at adequate pressures, to all water users. In order to plan or design a water distribution system, the location or point of demand must be known or assumed, and the magnitude of each demand known or estimated; water demands may then be categorized by purpose as domestic, industrial, special, or fire protection. Sizing of the water treatment plant, water storage facilities, distribution pumps, or distribution mains, is dependent on the size of the other parts of the system. It is not practical to size individual distribution mains without considering the other elements of the system. The effectiveness of any proposed combination of storage, pumping, and distribution works in meeting projected peak demands is best determined by hydraulic analyses of the system. Such hydraulic analyses, usually performed on digital computers, are very helpful to system planning.

9.6 Cross Conection

9.6.1 Avoidance of cross connections

If fires are to be fought with both potable and nonpotable supplies, separate distribution systems must be used to deliver the two types of water to the required area. Hydrants or other connections for each system should be suitably identified to discourage improper use. Standby water reservoirs serving fire protect ion systems are sometimes filled from both potable and nonpotable supplies. If this is the case, the potable water shall be discharged to the reservoir through an air break not less than 12 inches above the maximum water level of the reservoir. In a similar manner, where potable water is to be used as gland seal on a pump handling nonpotable water, the potable water must be stored in a tank with an air gap between the end of the water supply line and the top of the tank. Special care must also be taken of such items as valve pits and water storage facilities to ensure that surface water runoff cannot enter potable water systems. Other situations that can result in back-siphonage are flexible hose having one end immersed in nonpotable water and the other end connected to a potable water hose bib, potable waterlines entering swimming pools without air gaps, lawn irrigation systems with sprinkler heads flush with the ground, and improper connections at vehicle wash racks.

9.6.2 Prevention of backflow

Devices for the prevention of backflow include air gaps and reduced-pressure-principle backflow preventers. Air gap distances should be at least twice the diameter of the water supply line, and reduced-pressure-principle backflow prevention devices should meet the criteria of American Water Works Association (AWWA) C506. Double check valves for backflow prevention are not considered suitable and should not be used. Back-siphonage can be prevented with air gaps, atmospheric-type vacuum breakers, or pressure-type vacuum breakers.

9.7 Pressure Regulation Alternatives

The pressure levels for the distribution system are discussed in the article 9.4. Alternative means to maintain these pressures consist of gravity systems, direct pressure systems, pneumatic system or a combination of the above. Pressure regulating valves are available to reduce system pressures if required.

9.7.1 Gravity pressure systems

This is the preferred method of maintaining adequate pressure in the system. Gravity pressure systems are inherently associated with elevated storage. A storage facility provides a reservoir in which the inflow and outflow of water can better match the hourly consumer demand and can be a supply source during emergency situations such as interruptions in the normal supply service or heavy demands for fire fighting. Reservoirs should be located within or adjacent to load centers (i.e., areas of high demand) of the distribution grid to meet water demands in those areas without causing high velocities and head losses in the distribution mains. The pressure in the system supplying water to the storage facility needs to be sufficient to fill the reservoir. If it is not, booster pumps may be required. Two types of tanks may be used in a gravity pressure system: elevated and ground storage tanks.

Elevated tank: Where ground elevations are relatively uniform, an elevated tank will be considered to maintain pressure in lieu of ground storage facilities where practical. The tank will be adequately sized. The height of the tank will be determined from the topography of the area served, the height of the buildings and the pressure losses in the distribution system. Standard and special designs are available in sizes up to 3,000,000 gallons. Standard design will be utilized except where special conditions warrant other designs. In addition, altitude valves, check valves and shut off valves are necessary to control the level of water in the tank and to provision or isolate portions of the distribution system during emergencies. These are to be contained in a valve pit near the base of the tank, protected from freezing, and will provide for appropriate connections to the distribution system.

Ground level storage: Ground level storage can consist of steel standpipes and steel or concrete ground storage reservoirs. These are to be designed where there is sufficient difference in ground elevation to maintain adequate pressure in the distribution system. Concrete reservoirs can be designed for any size system, but are more often used for larger sizes, i.e., those exceeding 1,000,000 gallons. Standpipes of 6 to 20 feet in diameter may be installed for small systems. If the difference in natural ground elevations is insufficient to maintain pressures, booster pumps may be required in conjunction with ground storage to increase system pressure.

9.7.2 Direct-pressure systems

A direct pressure distribution system is one in which no elevated storage is provided, and the required distribution pressures are maintained only by pumping facilities. A ground level storage tank may be provided to serve as an intake supply for the pumping facilities. Direct-pressure distribution systems will be considered only where the military use or special requirements will not permit the utilization of elevated storage tanks. Caution must be used in design to reduce surge pressure and compensate for variable volume demands. Provisions must be made to ensure the availability of sufficient supply to meet fire and emergency demands. The pumping facilities in a direct-pressure system must have firm capacities equal to or greater than the peak demand rates exerted on the system. The firm capacity of a pumping facility is the total pumping capacity with the largest pump out of service. Automatic controls are available which react to pressure sensors and cycle the pumps according to a sequence which may be predetermined by the operator.

- **Pumping stations:** For variable flow requirements, consideration will be given to variable speed pumps, multiple pumps with stage control, flow regulating valves, or flow recirculation. The usual location is at the supply and treatment facility. Additional units may be located within the distribution system. Consideration should be given to providing a by-pass around pumps in the distribution system so that some flow may be maintained even when the pump is out of service. The pumps and associated equipment shall be contained in a vault or pump house to protect the equipment from the environment.

- **Line boosters:** Line booster stations may be designed where system head loss dictates their use. This may include distribution system areas that are remote from pumping stations, high rise building areas where normal pressure is inadequate, localized areas of higher elevation or extensions to existing distribution system where the cost of additional elevated storage is prohibitive. These pumps may be submersible turbine pumps, mounted in housings which can be installed in a water main much the same as a regular section of pipe. The pumps may be buried underground. As with all electrical mechanical devices, they are subject to maintenance needs. Therefore, provision must be made for future maintenance which may include excavation of the installation. Other types of pumps, most commonly centrifugal, may be installed in a vault or pump house. This installation is designed as any other pump station.

- **Multiple pressure levels:** In multiple pressure level distribution systems, where pumps are installed in the system, the designer should check for circulation around the pumps. If recirculation of water from the high pressure system to the low pressure system is possible, which would cause the water to be pumped twice, distribution line valves must be closed or check valves should be installed.

9.7.3 Pneumatic System

A hydropneumatic tank "riding" on the system serves two functions. First, it can act as a reservoir of water for emergency supply for a short period of time such as a supply for a sprinkle- head; second, it can act as an air spring or piston and is a reservoir of stored energy to maintain pressure in the system and help avoid short- cycling of the pumps.

- **Applicability:** Hydropneumatic distribution systems are applicable where demands are less than 500 gallons per minute.

- **Pressure settings:** The low pressure setting on the hydropneumatic tank is determined by distribution system requirements. The recommended minimum operating pressure is 30 pounds per square inch (psi), at the highest ground elevation in the distribution system. The high pressure setting on the hydropneumatic tank is dependent on the maximum allowable pressure in the distribution system. The recommended maximum operating pressure is 100 psi. For a specific application, the pressure variation in the tank is normally about 20 psi. The low water level (water level at the low pressure setting) must be high enough to provide a water seal. At the low water level, the water remaining in the tank should be at least 10 percent of the capacity of the tank. The high water level should be calculated to provide maximum efficiency. The pumps will be sized to deliver 125 percent of the calculated peak demand of the distribution system. The tank size will be at least 10 times the rated capacity of the pump. The tank will be sized so that the pump cycles not less than 4 times per hour, nor more than 10 times per hour, unless the pump motor horsepower rating exceeds 50, in which case the maximum number of cycles will be 6 per hour. Completely automatic hydropneumatic tank controls are available to maintain proper operating conditions (correct air-water volume ratios) during each pump cycle. An auxiliary air compressor-type, air charging system will be used for tanks larger than 750 gallons and pressures higher than 75 psi. An air volume control valve operation will be used to maintain correct air-water volume ratios for all other applications.

9.7.4 Pressure regulating valves

Pressure regulating valves function to reduce an existing high pressure to a uniform downstream pressure. Although this function can be accomplished by partially closed line valves, this method requires manual operation or motorized operators with remote control and continuous monitoring. Automatic pressure reducing and sustaining valves are available which react to distribution system pressures. These valves operate on two principles.

- **Direct action:** A direct-acting regulator cannot regulate pressure closely if considerable range of variation between the wide open and nearly closed positions is required. The regulated pressure is influenced considerably by variations in the high pressure side, and a great differential must always exist between the high side and the regulated side. Such regulators give excellent service in small sizes where accurate regulation is not important or where the rate of flow is fairly steady.

- **Pilot operated:** In water distribution regulation, it is important to sustain the pressure as load increases. With pilot-operated reducing valves, it is possible to get extremely close regulations at any flow up to the full capacity of the valve wide open. Pilot-operated valves may chatter and perform improperly when flow is very small and the disc or piston is close to the seat. Each valve must be provided with two gate valves, permitting it to be shut off for repairs without interfering with other valves. Pressure regulators, like other automatic equipment, should be inspected weekly to insure good operation and discover the need for preventive maintenance before a serious breakdown occurs.

9.8 Distribution Mains

9.8.1 Main sizes

Water distribution mains of various materials are readily available in sizes ranging from 6 to 48 inches inside diameter; large pipes up to 144 inches and greater can be specially made. Minimum diameter for distribution mains and fire branches is 6 inches.

- **Domestic requirements:** The system should be capable of delivering the peak domestic demand as described, plus any special requirements, at pressures not lower than 30 pounds per square inch at ground elevation. The required daily demands should be determined by calculating the effective populations of various areas to be served and applying the appropriate per capita water allowances. For small installations not having elevated storage, the peak domestic demand will be determined.
- **Fire flows:** The distribution system will be designed to deliver the necessary fire flow requirements. And any industrial or special demands which cannot be reduced during a fire. When only hose streams are supplying the required fire flow streams, residual ground level water pressures at fire hydrants should be not less than 10 pounds per square inch. If sprinkler systems are used, residual pressures adequate for proper operation of the sprinkler systems must be maintained. Specific guidance as to fire flows and pressure required for various structures and types of fire protection systems.
- **Friction losses:** In computing head losses due to friction in a distribution system, the Hazen Williams formula, as given below, will be used.

$$V = 1.318CR^{0.63}S^{0.54}$$

Where,

V = the mean velocity of the flow, in feet per second.

R = the hydraulic radius of the pipe in feet, i.e., the cross-sectional area of a flow divided by the wetted perimeter of the pipe. For a circular pipe flowing full, the hydraulic radius is equal to one-fourth the pipe diameter.

S = the friction head loss per unit length of pipe (feet per feet).

C = a roughness coefficient, values of which depend on the type and condition of pipe. Typical values of this coefficient are shown in Table 9.1

Table 9.1: **Pipe materials and valves**

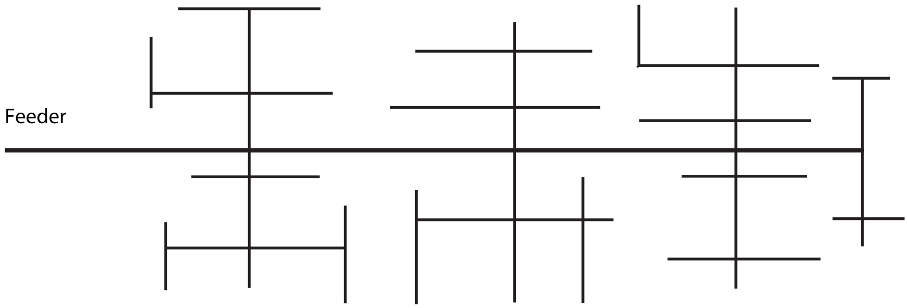
Pipe Materials	C
Concrete (regardless of age)	130
Cast iron: New	130
5 years old	120
20 years old	100
Welded steel, new	120
Wood stave (regardless of age)	120
Asbestos-cement	130
Plastic (PVC, Fiberglass)	130

Value as high as 150 are claimed for plastic pipe. The values shown in Table 9.1 are considered practical limits because of losses that may result due to fittings and valves, and because of improper installation. Hydraulic analyses will normally be made using a value of 100 for the roughness coefficient. However, consideration should be given to the use of coefficients greater than 100 when specifying concrete, asbestos-cement, or plastic pipe under conditions that experience has shown will not seriously reduce the carrying capacity of these pipes, within the anticipated economic life of the project. Coefficients greater than 130 should not be used. In some cases, expansions to existing distribution networks, rather than entirely new networks, must be planned. In such instances, it may be desirable to determine the roughness coefficients of the existing pipelines through a series of coefficients tests. These involve isolating sections of pipeline to the greatest extent possible, measuring the flow through the pipelines, and monitoring the changes in the hydraulic gradient between different points on the same pipes. This information can be used to derive the friction head loss per unit length of pipe, and, in turn, a roughness coefficient can be calculated.

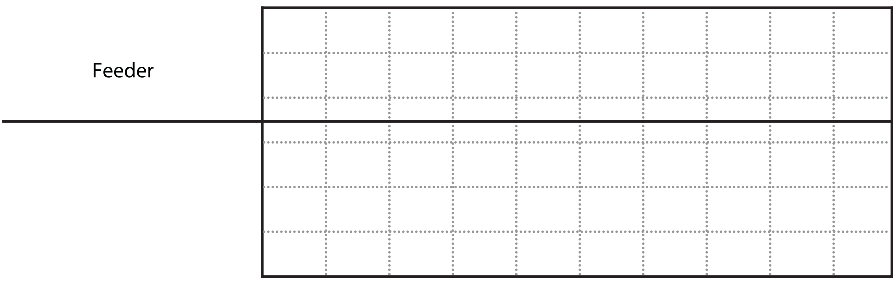
- **Fire-hydrant branches:** Fire-hydrant branches (from main to hydrant) should not be less than 6 inches in diameter and as short in length as possible, preferably not longer than 50 feet with a maximum of 300 feet.

9.8.2 Location of mains

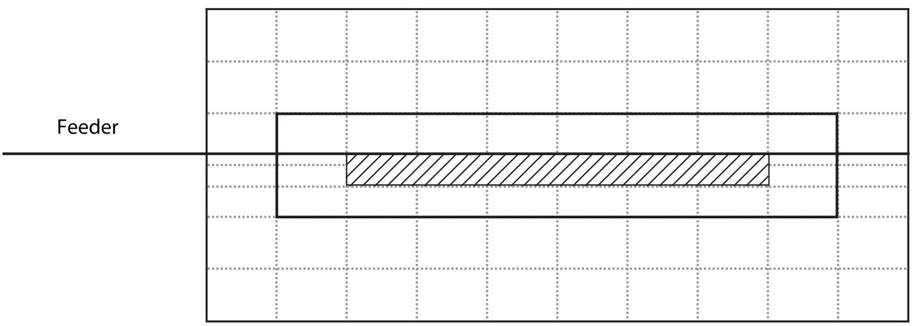
- **General:** Mains should be located along streets in order to provide short hydrant branches and service connections. Mains should not be located under paved or heavily travelled areas and should be separated from other utilities to ensure the safety of potable water supplies, and that maintenance of a utility will cause a minimum of interference with other utilities.
- **Distribution system configuration:** The configuration of the distribution system is determined primarily by size and location of water demands, street patterns, location of treatment and storage facilities, and topography. Two patterns of distribution main systems commonly used are the branching or dean end, and gridiron patterns.
  - **Branching system:** The branching system shown in Figure 9.11 evolves if distribution mains are extended along streets as the service area expands. Dead ends in the distribution system are undesirable and should be avoided to the extent possible.
  - **Gridiron system:** The second distribution configuration is the gridiron pattern shown in Figure 9.11. The gridiron system has the hydraulic advantage of delivering water to any location from more than one direction, thereby avoiding dead ends. The use of a gridiron pattern looped feeder system is preferable to the use of a gridiron pattern with a central feeder system because the looped feeder supplies water to the area of greatest demand from at least two directions. A looped feeder system should be used for water distribution systems whenever practicable. Although it is advantageous to have all water users located within a grid system, it is often impracticable to do so. Water is normally delivered to a remote water user, or a small group of users, by a single distribution main. Therefore, the majority of the water users are served within a gridiron system while the outlying water users are served by mains branching away from the gridiron system. Branching mains should be avoided to the greatest extent possible.
- **Horizontal separation between water mains and sewers:** Water mains should be laid horizontally, a minimum of 10 feet, from any point of



(A) Branching or Dead-End Pattern



(B) Gridiron Pattern With Central Feeder



Gridiron Pattern With Looped Feeder (Area of Highest Demand Cross-Hatch)

Figure 9.11: Water distribution system patterns

existing or proposed sewer or drain line. Water mains and sewers must not be installed in the same trench. If any conditions prevent a horizontal separation of 10 feet, a minimum horizontal spacing of 6 feet can be allowed, but the bottom of the water main must be at least 12 inches above the top of the sewer. Where water mains and sewers follow the same roadway, they will be installed on opposite sides of the roadway, if practicable.

- **Water main sewer crossings:** Where water mains and sewers must cross, the sewer will have no joint within 3 feet of the water main unless the sewer is encased in concrete for a distance of at least 10 feet each side of the crossing. If special conditions dictate that a water main be laid under a gravity- flow sewer, the sewer pipe should be fully encased in concrete for a distance of 10 feet each side of the crossing, or should be made of pressure pipe with no joint located within 3 feet horizontally of the water main, as measured perpendicular to the water main. Pressure sewer pipe shall always cross beneath water pipe and a minimum vertical distance of 2 feet between the bottom of water pipe and the top of pressure sewer pipe shall be maintained. The sewer must be adequately supported to prevent settling.
- **Protection in airfield pavement areas:** Water mains should not be located under airfield pavement areas if other locations are available and economically feasible. Special protections of the mains are required when alternative locations are not available and it is necessary to locate water mains under pavement areas. The amount of protection needed is dependent upon the importance of maintaining a supply of water to the area served by the main, and on the availability of emergency water supplies to the affected area. The degrees of protection should be considered as follows:
    - **Minimum protection:** The water main must be enclosed in a vented, open-end, outer conduit from which the main can be removed for repairs or replacement. The outer conduit must have sufficient strength to support all foreseeable loadings.
    - **Intermediate protection:** Intermediate protection requires the water service to be carried under the airfield pavement by dual waterlines enclosed in an outer conduit or, preferably, in separate conduits.
    - **Maximum protection:** Where more than one utility crosses the airfield pavement and individual crossings would be more expensive than a

combined crossing, the utilities will be enclosed in a utility tunnel of sufficient size for in-place repairs. Special precautions must be taken in the placement and protection of individual utility lines within the tunnel to ensure that failure of one utility does not affect the service of the others. Special protection of mains is not required where the mains are located beneath pavement areas that are not normally subject to the movement under their own power.

9.8.3 Dual water supplies

- **Applicability:** Dual water supply systems consist of independent pipe networks supplying two grades of water to users. The higher quality water is used for domestic purposes such as drinking, cooking, dishwashing, laundry, cleaning, and batching; the lower quality water may be used for toilet flushing, fire fighting, lawn and garden watering, and commercial or industrial uses not requiring high quality water. Dual water supply systems are not feasible except under unusual circumstances. A dual water supply might be utilized when the only available water supply is brackish and the cost of a dual system is less than the demineralization cost of all the water supplied to users; or when only a limited quantity of higher quality water is available, and it is more economical to construct a dual system than to implement the required treatment of the lower quality water. If a dual water supply system is established and the lower quality water use might result in human contact or ingestion (e.g., toilet flushing, lawn and garden watering), both water supplies must be disinfected.
- **Evaluation of dual water supply system:** The design of dual water supplies will be determined using results of feasibility studies which have substituted all engineering, economic, energy, and environment factors. If a dual water supply system is installed and a brackish water is used as the lower quality water, metallic pipes and plumbing facilities exposed to the brackish water may have considerably short lifespans than similar facilities exposed to water of better mineral quality. There will be no connection between the two pipe networks of a dual distribution system.

9.8.4 Recycling used water

There are operations that generate effluent water than can be reused for the same operation after minimal treatment. This does not constitute a dual system. Examples of such effluents are laundry wastes, vehicle washrack waste water, and plating operations waste water. Recycling of such water should be practiced wherever feasible.

9.9 Distribution Systems Pressures

Water distribution systems should be designed to maintain operating pressures within the system between 40 and 75 pounds per square inch at ground elevation. Minimum pressures of 30 pounds per square inch under peak domestic flow conditions can be tolerated in small areas as long as the distribution system is also capable of meeting fire flow requirements to these areas. Minimum ground-level residual pressures at fire hydrants will be at least 10 pounds per square inch while supplying fire flows. Maximum pressures of 100 pounds per square inch can be allowed in small, low-lying areas not subject to high flow rates and surge pressures. Areas of excessively high or low pressures require that the system be divided into multiple pressure levels.

9.9.1 Multiple levels

Where multiple-level systems are required, it is desirable to establish the lines of separation so that the pressures in each system will approach the optimum range of 40 to 75 pounds per square inch. Three or more levels will not be used unless distribution pressures in a large area of the two-level system fall below 30 pounds per square inch, or approach or exceed 100 pounds per square inch. In all circumstances, fire flows must be adequate.

9.9.2 Pressure reducing valves

Pressure-reducing valves will be required in areas of the distribution system that have pressures in excess of 100 pounds per square inch. The pressure-reducing valves may be installed on the mains serving these areas or on the individual building service lines in high-pressure areas. If pressure-reducing valves are to be installed on individual service lines, the preferred location is adjacent to, and upstream from, the water meter for each building or immediately inside the building being served. In some cases, it may be necessary to install pressure-reducing valves only on lines to certain plumbing or heating units which are adversely affected by excessive pressures.

9.9.3 Pressure relief valves

Pressure-relief valves should be installed in all systems which might be subjected to greater than allowable pressures. In systems with 100-pounds per square inch pumps, the pressure-relief valves should be set to discharge at 120 pounds per square inch; pressures greater than 120 pounds per square inch may be experienced for brief periods during testing or operation of these pumps. All pumps driven by variable speed motors or engines should be provided with relief valves; and if the shutoff pressure of any pump exceeds 120 pounds per square inch, the pressure-relief valves should be installed and set at approximately 120 pounds per square inch.

9.9.4 Cavitation

Cavitation is a complex phenomenon that may take place in pumps. In a centrifugal pump, as liquid flows through the suction line and enters the eye of the impeller, the velocity increases and pressure decreases. If the pressure falls below the vapour pressure corresponding to the temperature of the liquid, pockets of vapour will form. When the vapour pockets in the flowing liquid reach a region of higher pressure, the pockets collapse with a hammer effect causing noise and vibration. Tests have shown that extremely high instantaneous pressures may be developed in this manner, resulting in pitting various parts of the pump casing and impeller. Conditions may be mild or severe and mild cavitation may occur without much noise. Severe cavitation can result in reduced efficiency and ultimate failure of the pump if steps are not taken to eliminate the cause.

9.10 Distribution System Equipment

9.10.1 Valves

The types of valves most frequently used in water distribution are gate, butterfly, ball, plug, globe, and check valves. Applications of the various types of valves and the standards to be used for these valves are given in Table 9.2. All valves should have the direction to open shown on their operators.

- **Gate valves:** Gate valves may have either a single solid wedge gate or double disc. Solid wedge gates are satisfactory in sizes up to 6 inches, but double disc gates should be used for larger sizes. Because of the excessive

Table 9.2: Valve applications and standards

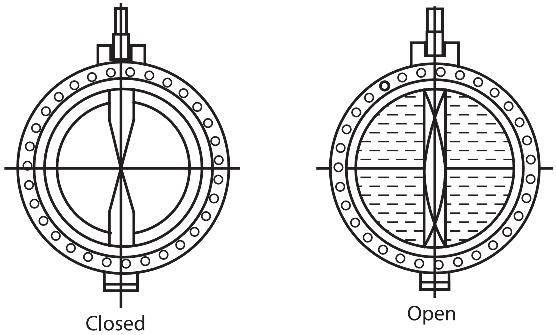
Type	Application	Size to be used (diameter, inches)	Standard
Gate	Sectionalizing distribution mains. Isolating fire-hydrant branches.	3 or larger	AWWA C500, standard for gate valves-3in. Through 48 in. For Water and Other Liquids.
Butterfly, rubber seated	Mains with water pressures less than 150 $lb\backslash in^2$	3 or larger	AWWA C504, standard for rubber-seated butterfly valves.
Ball	Applications involving throttling or frequent operation. Water service lines.	6 or less	AWWA C507, standard for valves, shaft or trunnion mounted-6in. Through 48in. For Water pressures up to 300 psi.
Plug	Application involving or frequent operation. Water service lines	6 or less	
Globe	Throttling operations. Water service lines.	2 or less	

wear and leakage of the gates and seats which may result, gate valves should not be used where frequent operation is required. If gate valves are left open for long periods, debris may accumulate in the seats and prevent complete closure, but if left closed for long periods, deposits may prevent opening. Gate valves should be operated periodically to break loose any deposits which might have formed. Large gate valves should be geared to make operation easier. A typical double disc gate valve is shown in Figure 9.12.

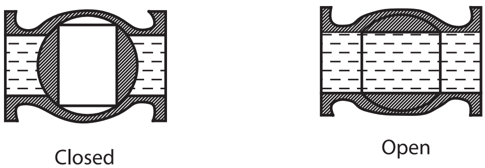
- **Butterfly valves:** The advantage of butterfly valves include easy operation, small space requirement, low cost, minimum maintenance, low head loss, driptight shutoff, suitability for throttling, and reliability. A disadvantage is that main cleaning and lining equipment cannot be used in lines containing butterfly valves without removing the valves. Mechanical valve operators will be designed to restrict the rate of closure so that water hammer will not occur in the system in which the valve is installed. A typical butterfly valve is shown in Figure 9.13.

Figure 9.12: **Double disc gate valve**

- **Ball valves:** Ball valves have the advantage of ease of operation, reliability, durability, and capability of withstanding high pressures, but are usually expensive. A typical small-diameter ball valve is shown in Figure 9.13
- **Plug valves:** Lubricated and eccentric plug valves are the types of plug valves commonly used. Lubricated plug valves normally have a cylindrical or tapered plug intersecting the flow with a rectangular port opening. Round ports can be obtained in the smaller sizes. Specially formulated greases are used both for lubrication and sealing of lubricated plug valves. When operated periodically, these valves are relatively easy to operate and provide a tight shutoff, but the plugs may freeze if not operated for a long period of time. Plug valves are especially good for high pressure applications. Eccentric plug valves are preferable to lubricated plug requirements; eccentric plug valves are also less prone to freeze. Ball and plug valves will not be used on buried pipelines, except when installed in a valve pit. The basic application for the eccentric plug valves are normally on small service lines.
- **Globe valves:** Globe valves are particularly well suited to throttling operations and most plumbing fixtures are normally equipped with these valves. Small globe valves normally have rubberized discs and metal seats to provide driptight shutoff, but special discs and seats are available for



(A) Typical Butterfly Valve



(B) Typical Small-Diameter Ball Valve

Figure 9.13: (A) Typical butterfly valve and (B) Typical small-diameter ball valve.

more severe conditions; and may be used on water service lines 2 inches or less in diameter.

- **Check valves:** Any valve used to prevent the reversal of flow is considered a check valve. Most check valves are equipped with plugs or hinged discs which close flow openings when flow is reversed. Rapid and complete valve closing is often ensured by the addition of special weights or springs to the plugs or discs. A newer type of check valve has spring-loaded, wafer-style, semicircular plates mounted on a vertical pivot through a flow port. The springs cause the plates to swing closed at the instant of flow reversal. This wafer-style check valve has the disadvantage of producing relatively high head losses and of showing excessive wear under some operating conditions.

- **Air release and vacuum relief valves:** Air release valves are required to evacuate air from the main at high points in the line when it is filled with water, and to allow the discharge of air accumulated under pressure. Excess air allowed to accumulate at high points creates a resistance to flow, and an increase in pumping power requirements results. Vacuum relief valves are needed to permit air to enter a line when it is being emptied of water or subjected to vacuum. Special valves and vacuum relief valves should be installed at high points in the line or where a long line changes slope.

- **Valve location**
  - **Shutoff valves:** The purpose of installing shutoff valves in water mains at various locations within the distribution system is to allow sections of the system to be taken out of service for repairs or maintenance without significantly curtailing service over large areas. Valves should be installed at intervals not greater than 5,000 feet in long supply lines, and 1,200 feet in main distribution loops or feeders. All branch mains connecting to feeder mains or feeder loops should be valved as closely to the feeders as practicable so that the branch mains can be taken out of service without interrupting the supply to other locations. In the areas of greatest water demand, or where the dependability of the distribution system is particularly important, maximum valve spacing of 500 feet may be appropriate. At intersections of distribution mains, the number of valves required will normally be one less than the number of radiating mains; the one valve will be omitted from the line which principally supplies flow to the intersection. Valves are not usually installed on branches serving fire hydrants on military installations. As far as practicable, shutoff valves should be installed in standardized locations (e.g., the northeast corner of intersections or a certain distance from the centerline of streets) so they can easily be found in emergencies. For large shutoff valves (approximately 30-inch diameter and larger), it may be necessary to surround the valve operator or entire valve with a vault to allow for repair or replacement. In important installations and for deep pipe cover, pipe entrance access manholes should be provided so that valve internal parts can be serviced. If valve vaults or access manholes are not provided, all buried valves, regardless of size, should be installed with special valve boxes over the operating nut in order to permit operation from ground level by the insertion of a special long wrench into the box.
  - **Blowoff valves:** Blowoff valves or fire hydrants should be installed at the ends of dead-end mains to allow periodic flushing of the mains. Primary feeder mains and larger distribution mains should have a blowoff valve in each valved section which should be installed at low

points in the mains where the flushing water can be readily discharged to natural drainage channels. Blowoff valves must be designed so that operation which will result in erosion or destruction of wildlife is not permitted. Special care must be taken to eliminate the possibility of contaminated water entering the distribution system through blowoff valves which have not been tightly closed.

9.10.2 Fire hydrants

- **Dry and wet barrel hydrants:** The most common types of fire hydrants are the dry- and wet- barrel varieties. They are similar in configuration and operation, but in the dry-barrel hydrant (AWWA C502), provision is made for draining water from the barrel after the hydrant is shut off. This is normally accomplished by gravity drainage through special drain outlets in the base or barrel of the fire hydrant. A dry-barrel hydrant is shown in Figure 9.14. Wet-barrel hydrants (AWWA C503) can be used in areas where the temperature is always above freezing.
- **Safety hydrants:** Barrel-type hydrants ext ending aboveground are available in models which could be damaged by automobiles or trucks without disturbing the main valve. These are the safety or traffic fire hydrants and should be used near heavily travelled roads or intersections where adequate protection of the hydrant cannot be provided.
- **Flush-top hydrants:** In cases where the barrel-type aboveground hydrant would interfere with normal traffic, a flush-top hydrant can be utilized. The operating nut and hose nozzles in this type hydrant are located in a cast-iron box below ground level. The top of the box has a horizontal lid which is flush with the adjacent ground surface. However, flush-top hydrants are more difficult to locate than barrel-type hydrants, especially in areas subject to heavy snows, and once located are awkward to uncover and put into operation. Barrel-type hydrants are preferable to flush-top hydrants. Hydrants of all types should have the direction to open shown on their operators.
- **Hydrant nozzles:** Nozzles on fire hydrants are either 2½ or 4½ inches in diameter. The 2½ inch nozzle is for direct connection to fire hoses and the 4½ inch nozzle is for use with mobile fire pumper units. Unless unusual conditions dictate otherwise, hydrants with two fire hose nozzles and one pumper nozzle should be used. The outlet nozzles on most hydrants are located at 90-degree angles to each other. The pumper outlet should

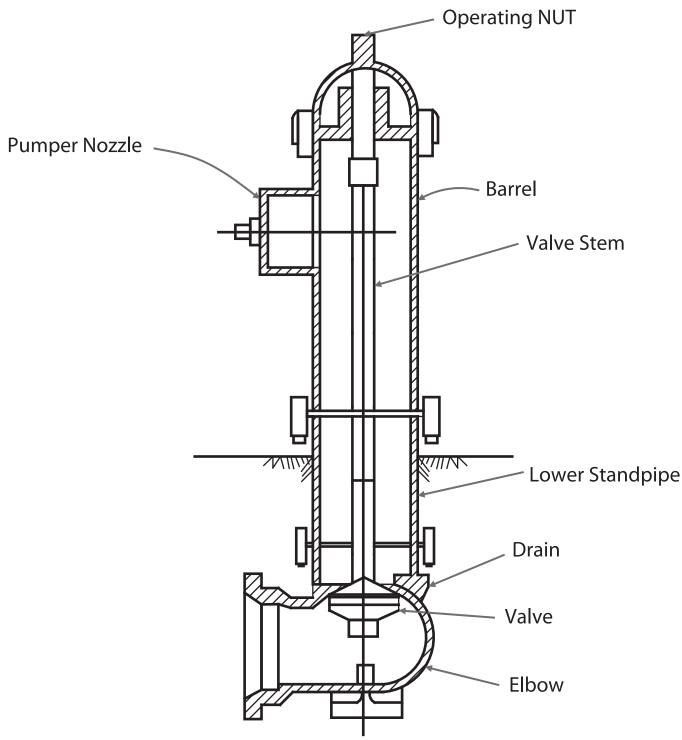


Figure 9.14: **Schematic of typical dry- barrel fire hydrant.**

normally face the street or intersection, and the two fire hose nozzles should face opposite directions, 90 degrees from the pumper nozzle. Hydrants with either more or less than three nozzles should be aliened so that the nozzles are readily accessible from the street

- **Hydrant spacing:** General Hydrant distributions will conform to the standards shown in Table 9.3.
  - **Residential areas:** The preferred location for fire hydrants in residential areas is at street intersections. Where additional hydrants are required because of the above hydrant distributions, these additional hydrants will normally be located adjacent to streets approximately halfway between intersections. Each single or duplex family unit will have at least one hydrant within 300 feet and a second hydrant within 500 feet.
  - **Airfields:** For airfield hangar areas, hydrants will be spaced

Table 9.3: Hydrant distribution

Required fire flow, gpm	Average area per hydrant, square feet	Required fire flow, gpm	Average area per hydrant, square feet
1,000 or less	160,000	6,000	80,000
1,500	150,000	6,500	75,000
2,000	140,000	7,000	70,000
2,500	130,000	7,500	65,000
3,000	120,000	8,000	60,000
3,500	110,000	8,500	57,000
4,000	100,000	9,000	55,000
4,500	95,000	10,000	50,000
5,000	90,000	11,000	45,000
5,500	85,000	12,000	40,000

approximately 300 feet apart, and where economically feasible will be connected to the base distribution system and not to the special system serving deluge sprinkler systems in the hangars. At double-cantilever hangar areas, hydrants will be connected only to the Base water distribution system. For mass parking, servicing, and maintenance areas, the fire hydrants will be installed along the edge of parking and servicing aprons. Hydrants will be spaced approximately 300 feet apart so that every part of the apron may be reached by approximately 500 feet of hose. One or more hydrants will be located within 300 feet of all operational service points.

- **Individually site-I buildings:** Where an adequately sized water main is available, or can be made available for an individually sited building such as a Reserve Center, two hydrants will be installed. However, one hydrant at the site is accept able if the provision of a second hydrant would require extension of a water main beyond the point necessary to serve the domestic demands of the building.
- **Remote fuel storage areas:** Fuel storage facilities that are remotely located with relation to public or military installation water systems will generally not have fire hydrant protection. However, where the facility is of a critical nature or is of a high strategic or monetary value that would justify some degree of fire protection, appropriate recommendations will be furnished with necessary supporting information.

- **Hydrant location**
  - Proper clearance should be maintained between hydrants and poles, buildings, or other obstructions so that hose lines can be readily attached and extended. Generally, hydrants will be located at least 50 feet from the buildings protected and in no case will hydrants be located closer than 25 feet to a building, except where building walls are blank firewalls. Hydrants may be located adjacent to blank portions of substantial masonry walls where the chance of falling walls is remote.
  - Street intersections are preferred location for fire hydrants because fire hoses can then be laid along any of the radiating streets. However, the likelihood of vehicular damage to hydrants is greatest at intersections, so the hydrants must be carefully located to reduce the possibility of damage. Hydrants should not be located less than 6 feet from the edge of a paved roadway surface, nor more than 7 feet. If hydrants are located more than 7 feet from the edge of the paved roadway surface and if the shoulders are such that the pumper cannot be placed within 7 feet of the hydrant, consideration may be given to stabilizing or surfacing a portion of wide shoulders adjacent to hydrants to permit the connection of the hydrant and pumper with a single 10-foot length of suction hose. In exceptional circumstances, it may not be practical to meet these criteria, and hydrants may be located to permit connection to the pumper using two lengths of suction hose (a distance not to exceed 16 feet).
  - Hydrants should not be placed closer than 3 feet to any obstruction nor in front of any entrance- ways. The center of the lower outlet should not be less than 18 inches above the surrounding grade and the operating nut should not be more than 4 feet above the surrounding grade.
  - In mass parking, servicing, and maintenance areas, the tops of hydrants will not be higher than 24 inches above the ground with the center of lowest outlet not less than 18 inches above the ground. The pumper nozzle will face the nearest roadway.
- **Hydrant installation:** Many problems of hydrant operation and maintenance can be avoided if the hydrant is properly installed. All hydrants should be installed on firm footings such as stone slabs of concrete bases to prevent settling and strains on line joints. Separation of the pipe joints in the elbow beneath the hydrant is sometimes a problem

because of forces created by the water pressure across the joint through the elbow. This problem can be alleviated by placing thrust blocks between the elbow and supporting undisturbed soil, or by tying the joint.

9.11 Water Pipe Materials

- **Types of materials:** Water distribution pipes are available in a variety of materials. Those most commonly used, and most suitable for use at military installations, are asbestos-cement, ductile iron, reinforced and prestressed concrete, steel, and plastic. All water mains and service lines should be designed for a minimum normal internal working pressure of 150 psi plus appropriate allowances for water hammer. External stresses due to earthfill and superimposed loadings will be calculated in accordance with the applicable standards of the American Water Works Association for each kind of pipe (Table 9.4), pipes with flexible joints will be used. Asbestos-cement pipe, mechanical-joint cast-iron pipe, or rubber gasket-joint pipe of various kinds (cast iron, steel, plastic, and reinforced concrete) may be used in these areas. The danger of earthquake damage can also be minimized if pipelines are laid in bedrock or coarse-grained sediments. Installation in fine-grained sediments such as clay and silt should be avoided in earthquake-prone areas if possible.
- **Selection of materials:** In selecting the material to be used for a particular application, the following items should be considered:
  - Ability to withstand maximum anticipated internal pressures and external loads or the most severe combination thereof.
  - Flow resistance of the pipe, both in new condition and after the pipe has been in service for several years.
  - Ease of installation. This involves the unit weight of the pipe, type of joints used, type of bedding required, and whether or not thrust blocking is required.
  - Resistance to external and internal corrosion.
  - Joint tightness.
  - Durability.
  - Ease of tapping for service connections.
  - Cost.

Information on pipe diameter, design, and coatings, linings and fittings for various types of pipe is given in Table 9.4 below.

Table 9.4: Pipe type comparison

Pipe type	Maximum diameter (Inches)	Pipe type	Coating, linings and fittings
Steel	96	M11	C200 series
Ductile iron	48	C150	C105 C100 series
Concrete	144	M9	C300 series
Asbestos-cement	16 42	C401 C403	C400 series
Glass fiber reinforced	144	C950	C950
Polyvinyl chloride	12	C900	C900

- **Description of materials.**
  - **Asbestos-cement pipe**

    This pipe is usually unaffected by corrosive soil conditions, and is installed in many locations where unprotected cast-iron or steel pipe would suffer excessive corrosion. Standard lengths of asbestos- cement pipe are 13 feet for pipe 8 inches or larger in diameter, and either 10 or 13 feet for pipe 4 or 6 inches in diameter. The three classes of asbestos-cement pipe are: class 100, class 150, and class 200 for pipe 4 inches through 16 inches and classes 30, 35, 40, etc., for pipe 18 inches through 42 inches. These refer to the maximum anticipated internal working pressure, not including sudden surges, to which the pipe is to be subjected. A factor of safety of 4.0 has been used in the design and manufacture of these pipes. They should theoretically be capable of withstanding internal bursting pressures of at least 400 psi (class 100), 600 psi (class 150), and 800 psi (class 200). Techniques for evaluating both intern al and external loads are given in AWWA C401, and C403. External loads include both the weight of the backfill supported by the pipe and the weight of superimposed loads, static or dynamic on the pipe. A factor of safety of 2.5 is used in designing for extern al loads.

    Asbestos-cement pipe is also grouped into two categories according to the percentage of uncombined calcium hydroxide in the pipe. Type I has no limit on the uncombined calcium hydroxide; type II has 1.0 percent or less. Inasmuch as the uncombined calcium hydroxide may be leached from the walls of a pipe, thereby reducing the strength of the pipe, type II pipe should be used whenever the sum of the pH, the logarithm (base 10) of the alkalinity (in mg/l as $CaCO_3$), and the logarithm (base 10) of the hardness (in mg/l as $CaCO_3$), and the water in the pipe is less than 12.0 but greater than 10.0.

Installation of asbestos-cement pipe will be in accordance with the provisions of AWWA C603. Direct tapping of asbestos-cement mains is permitted for service connections of 1-inch diameter or smaller. With the use of special service clamps, tapping for service connections up to 2 inches in diameter is permitted.

- **Ductile-iron pipe:** Ductile-iron pipe of equivalent thickeness is stronger, tougher, and more flexible than the now obsolete gray cast-iron pipe. The prescribed method of determining the required thicknesses of ductile-iron pipe is given in AWWA C150. Ductile iron shall be used in situations where some pipe deflection may occur, such as in earthquake-prone areas or in soil conditions where settling of the pipe may occur. Ductile- iron pipes are frequently lined with coal-tar enamel or cement mortar to reduce corrosion of interior surfaces. Cleaning and lining of corroded ductile-iron pipe can substantially reduce the head losses in the pipe; pipeline cleaning without lining is not permitted.
- **Concrete pipe:** Concrete pipe is strong, durable, corrosion-resistant, and has a smooth interior which allows high flow velocities with minimal head losses. Without special equipment or expertise, concrete pipe is more difficult to tap than cast iron and it should not be used where multiple futures tapping for building service may be required. Three types of concrete pipe commonly available are: noncylinder, nonprestressed concrete pipe; nonprestressed concrete cylinder pipe; and prestressed concrete cylinder pipe. Concrete cylinder pipe has a steel cylinder either outside the concrete or embedded in the concrete of the pipe (AWWA C300, C301)
    - **Noncylinder, nonprestressed concrete pipe:** This pipe has both longitudinal and circumferential reinforcing bars cast in the concrete. It is not as strong as prestressed concrete pipe and should be used only if internal working pressures are not anticipated to exceed 55 pounds per square inch. Information on design and manufacturing parameters for this pipe are contained in AWWA C302.
    - **Nonprestressed concrete cylinder pipe:** This pipe is most commonly used in diameters of 24 to 144 inches and lengths of 12, 16, or 20 feet. It is suitable for use when working pressures are less

than 260 pounds per square inch. Each section of pipe consists of a welded steel cylinder encased in concrete, with longitudinal and circumferential reinforcing bars in the outer portion of the concrete. This pipe will conform to the requirements of AWWA C300.

  - **Prestressed concrete cylinder pipe:** There are two types of prestressed concrete cylinder pipe available. They are the lined-cylinder type with concrete cast inside the steel cylinder, wire wrapped under tension around the steel cylinder, and a concrete or mortar covering over the wire and cylinder; and the embedded-cylinder type with the steel cylinder encased in concrete, wire wrapped on the outer concrete surface, and the wire covered with a coating of cement or mortar. Both types are characterized by high strength and relatively lightweight as compared to other kinds of concrete pipe. The lined-cylinder type is used for pressures up to 250 pounds per square inch and the embedded-cylinder type for pressures up to 350 pounds per square inch. Diameters of the pipes range from 16 to 48 inches for the lined-cylinder type and from 24 to 144 inches for the embedded-cylinder type. The design and manufacturer of both types of prestressed concrete cylinder pipes is covered in AWWA C301.
  - **Concrete pipe:** Operating experience has shown that rubber gasketed bell-and-spigot joints provide a long-lasting, water-tight seal when proper installation procedures are followed. Subsequent coating of the joint with mortar ensures water tightness. Other types of joints are also available and may be used.

- **Steel pipe:** The properties of steel pipe favouring its use are high strength, and ability to yield or deflect under a load while still resisting the load, the capability of bending without breaking, and the ability to resist shock. Like cast iron, steel pipe is susceptible to corrosion if effective coatings and linings are not applied and maintained. Corrosion products do not adhere to steel pipe and are continually sloughed off, thus allowing further corrosion. By contrast, corrosion products adhere to cast-iron pipe and offer some protection against further corrosion. Steel pipe is generally available in diameters ranging to 144 inches and greater. Maximum allowable working pressures depend on pipe wall thicknesses and may be selected for the entire range of waterworks applications using AWWA. In designing steel pipe to withstand internal pressures, a factor of safety of 1.0 is generally used; a factor of safety of 1.5 or 2.0 is recommended in designing for external loads. Steel pipe may be used for transmission, distribution and service lines with adequate protective coatings, and linings and cathodic protection as determined necessary by site conditions. The design and manufacture of steel pipe 6 inches and larger and cast-iron pipe is given in AWWA C 101 and C200.

- **Plastic pipe:**
    - Several different types of plastic have been used in the manufacture of water distribution pipes. The most commonly used plastics include ABS (polymers of acrylonitrile, butadiene, and styrene). Polyethylene (PE), and polyvinyl chloride (PVC). Inasmuch as PVC pipe is presently the most suitable type of plastic pipe for water distribution, it is the only type covered herein.
    - The advantages of PVC pipe are that it has a very low resistance to flow, it is somewhat flexible and can deflect under earth or superimposed loadings, it does not corrode from electrical or microbial action, and it is relatively lightweight and easy to install. Disadvantages are that it suffers a permanent loss of tensile strength with time, and that the tensile strength of the pipe at any time is decreased by temperature increases. PVC pipe also undergoes significant expansions and contractions with temperature changes, necessitating the use of gasket couplers.
    - PVC pipe is used in sizes of 4 to 12 inches inside diameter. It is available in pressure classes of 100, 150, and 200 pounds per square inch, which correspond to the maximum anticipated internal working pressure for the pipe. A factor of safety of approximately 3.0 is used in the design of PVC pipe for sustained internal pressures; and a factor of safety of 4.0 is used for sudden pressure surges. However, due to the loss of tensile strength with time, these factors of safety decrease correspondingly. Pipe conforming to AWWA C900 with elastomeric gasket bell and spigot joints in 4inch diameter through 12 inch diameter size, is acceptable for transmission, distribution, and service lines. Transmission, distribution, and service lines less than 4 inch diameter will require schedule 80 pipe with threaded joints. The use of plastic pipe should normally be included as an option for contractors bidding on installation of new piping systems.

9.12 Service Connections

9.12.1 Tapping of water lines

- **Asbestos-cement lines:** Asbestos-cement pipe can be tapped either wet or dry using standard waterworks equipment. The largest size corporation stop which can be tapped directly into asbestos- cement pipe is one inch. Larger outlet sizes up to 2 inches can be secured by using service clamps or bossed sleeves. Tapping sleeves and valves can be used for making taps larger than 2 inches in asbestos-cement pipe under pressure.
- **Concrete lines:** New service connections on existing concrete pipelines can be made with or without interruption of service. Concrete pipe is not necessarily more difficult to tap than other pipe materials; however, the cost of pressure tapping the pipeline is considerably greater than incorporating outlets for future connections during pipe manufacture. Fittings are available for making threaded connections from ½ to 2 inches in diameter for the various types of concrete pressure pipe. Flanged outlet taps can be made under pressure for branch lines with diameters as large as one size smaller than that of the pipe to be tapped. Step-by-step procedures for small and large pressure connections are available in most manufacturer's literature.
- **Steel lines:** Service connections to steel pipe can be readily made with commercially available equipment. This includes service connections both dry and with pipe under internal pressure. Small service connections consist of threaded couplings welded to the steel pipe surface and drilled through with standard drilling equipment. Large diameter service connections are normally made under press ure utilizing a flanged service outlet, a tapping valve, and a standard drilling machine. The service outlet may be either a bolted-in-place service saddle or a fabricated steel service saddle that is welded to the pipe.
- **Plastic lines:** Plastic pipe can be direct tapped wet or dry, using standard waterworks equipment, for insertion of corporation stops. However, a special tool has been developed which will minimize PVC shavings and retain the coupon. The largest size corporation stop which can be tapped directly into the pipe is one inch. AWWA thread recommended by these methods is 2 inches. Tapping sleeves and valves can be used for making large taps under pressure, size to size, i.e., 8 inch outlet in 8 inch pipe, etc.

Tapping sleeves should be assembled in accordance with the manufacturer's directions.

9.12.2 Service connection materials

- **Copper:** Copper has been the most widely used material for service piping due to its flexibility, ease of installation, corrosion-resistance, and the capability to withstand high pressure. Although the cost of copper pipe has risen rapidly in recent years, it is still well suited for service connection use.
- **Plastic:** Plastic pipe is frequently selected because of its relatively low cost and easy installation. The capabilities of plastic pipe to withstand maximum internal and external loadings and temperatures should be carefully examined before use.
- **Galvanized steel:** Galvanized steel pipe has been used for service connections for many years. The main advantage of galvanized steel pipe is its relatively low cost. However, since galvanized steel pipe is rigid and requires threading, it is not easily installed. Also, galvanized steel service connections may have relatively short lives if placed in soils in which corrosion is likely to occur. Galvanized steel pipe is generally not used for 2 inch or larger service connections.
- **Other materials:** Ductile-iron and asbestos- cement pipes are not generally available in the small sizes required for most service connections, but could be considered for service connections to large water users for which pipe sizes of 3 inches or larger are needed.

9.12.3 Sizes

The size of the service connection needed in any particular situation should be the minimum size through which water can flow at the maximum required rate without excessive velocity or head loss. A maximum velocity of 10 feet per second is commonly used. In general, head losses through service connections during maximum flows should be small enough to ensure that a residual pressure of 25 pounds per square inch is available for water distribution within the plumbing of each building. Head losses of 15 pounds per square inch or greater through service connections are considered excessive, even if the 25 psi residual criterion can be met.

9.12.4 Installation

Service connection will be installed in as direct a path as possible from the distribution main to the building served, and will enter the building on the side closest to the distribution main. Service connections will be installed below the frost depth. If the size and wall thickness of the main are adequate, smaller service lines may be connected to the main by direct drilling and tapping. This can be accomplished with special machines while maintaining water pressure in the main. Larger service connections (greater than 2 inch) may necessitate the installation of tees or special branch connections into distribution mains, but may be made with the main under pressure with a tapping machine, tapping valve, and sleeve in most cases.

9.12.5 Service connections at airfields

Water-service connections are required for servicing. These connections will be located adjacent to the parking apron at nondispersed stations or adjacent to the servicing apron at dispersed stations.

9.13 Forces Acting on Pipe

Pipes carrying water under pressure must be designed to withstand stresses caused by internal and external loads, and temperature changes and to satisfy the structural and hydraulic requirements. The forces are:

- Internal forces due to static head
- Internal forces due to water hammer
- Forces at bends and changes in cross-section
- Forces due to temperature changes
- External forces in the form of backfill, traffic and own weights.

9.13.1 Internal forces due to static head

Internal forces due to static head create hoop stress and longitudinal stress.

Hoop stress, $S_h = pd/2t$

Where

S_h = hoop stress per linear length in inch of the pipe.
p = intensity of static pressure in psi= wh,
in which h is the static head and w is the unit wt of water.

d = pipe diameter in inch.
t = thickness of the pipe shell in inch.

Longitudinal (tensile) stress, $S_t = pd/[4t(d+t)]$ 9.3

$= pd/4t$ (approximately) 9.4

$= pd/4t$

9.13.2 Water hammer

One of the most damaging factors to a water piping system is water hammer action. In addition to its effect on the piping system, water hammer causes banging noise in the system that is very disagreeable to occupants in the building. Water hammer occurs when a column of water flowing through a pipe line and discharging at an open outlet, suddenly stopped by closing the outlet. Since flowing water has forces, tremendous pressure result at the point of closure and pressure surges move along the pipe. The manner in which water hammer occurs is illustrated in Figure 9.15

Phase 1: The valve on the line is closed and the water contained in the line is at rest. Water pressure is thus exerted in all directions in equal way.

Phase 2: The valve on the line is open and water flows freely through the outlet. Now the water pressure is utilized top forces the water out of the open end of the pipe. Arrows indicate the direction of forces in the column of water.

Phase 3: When the valve is quickly closed, the column of freely flowing water is suddenly stopped; excessively high pressures are generated at the point of stoppage. This reaction is the same as would result of a steel bar moving through the line at the water were suddenly stopped by the valve.

Phase 4: In an effort to equalize the pressure build up of the water, a shock wave will travel back along the branch line until a larger diameter pipe is reached. This will allow the shock wave to dissipate itself. Arrows denote the direction of forces toward line valve and then its reversal as as a shock wave toward the point of relief. Since shock wave travels at speeds in excess of 4000 fps it causes a piping clatter all along its route. Often the shock wave will oscillate back and forth between the valve and the point of relief until the pressure is stabilized with the branch line.

The pressure generated by the shock wave can expand and often rupture the piping. Although piping clatter is normally associated with water hammer, you

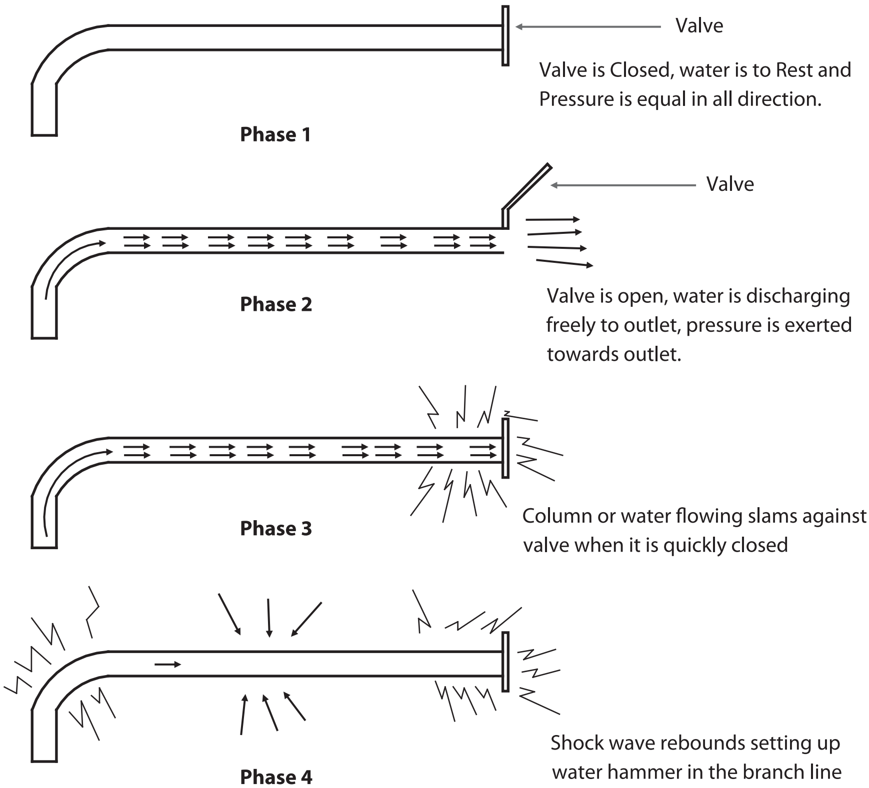


Figure 9.15: How water hammer can develop in a pipe line

cannot assume that when the noise do not occur that, the shock wave is non-existent. Quick often, water hammer takes place without any physical sound. Therefore, it is very important that piping system be designed with all due consideration given to the means that compensate for the action of water hammer.

9.13.3 Causes of water hammer

Not all the noises heard in the piping system can be attributed to water hammer. Loose faucet or valve washer can cause a ponding or chattering in the piping. Improperly supported and secured piping can created noises as the following water causes the piping to vibrate and thus rattle against steel members. This noise is easily transmitted through the piping system. Undersized water piping with excessive pressure will produce shrill sounds. Then of course, there is certain equipment as pumps that will produce noise unless the pumps are isolated and the piping connections are equipped with flexible piping connectors.

As for water hammer, it is generally impossible to predict just where in the piping system that water hammer will occur. There may be a small diameter branch line system which by the nature of its length and the fixture it supplies should produce water hammer. Yet it does not. On the other hand a larger diameter line which should not cause water hammer often will cause such noise.

One of the basic causes of water hammer is quick closing of values. These include any valve or faucet that is closed rapidly. If the same valve, were closed slowly, then the following water would have the chance to stabilize without producing the shock waves of water hammer. However since these valves must function in the manner intended, other means must be employed to minimize the violent action of water hammer.

Other factors that contribute to the positive occurrence of water hammer are excessive water pressure, inadequate piping sizes and water piping that is improperly installed.

Methods of Controlling Water Hammer: In order to eliminate the danger and piping clatter that results from water hammer, it is important that certain steps be taken in piping system design to compensate for the excessive pressure that are generated when a column of flowing water is suddenly stopped.

The consideration needed is some means or device that will provide flexibility in the system to absorb the initial shock wave of water hammer there by confining the action to a given section of piping.

Air is the most effective medium for absorbing the shock wave caused by water hammer for

- Water is non compressible
- Air can be compressed to considerable pressure when the water compresses the air, it also fills the void offered by the displaced air. Because water has this flexible means to expend its forces, the shock wave that would otherwise result, is quickly absorbed. The manner in which air serves to climate water hammer is shown in Figure 9.16

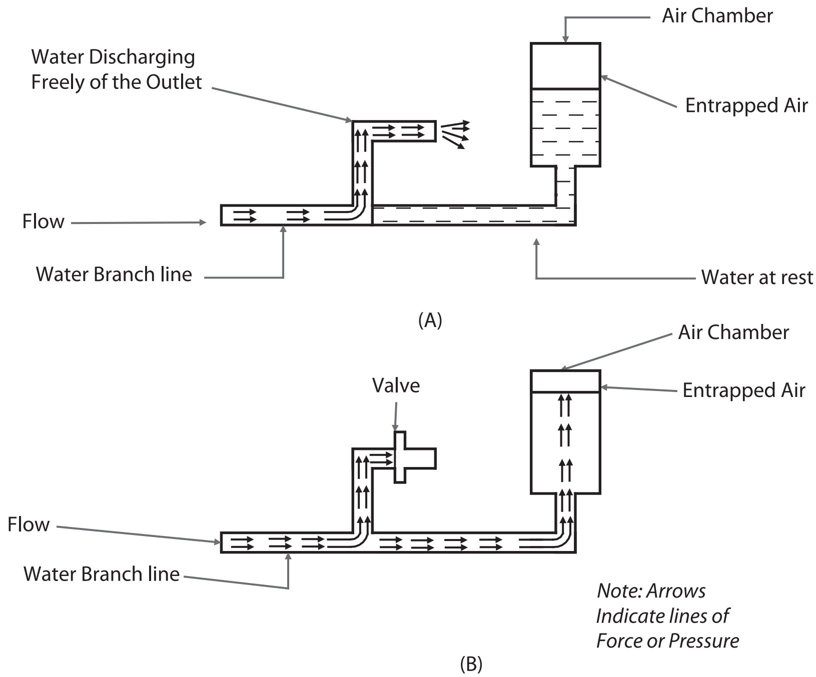


Figure 9.16: **How air chambers cushion the initial shock wave generated by water hammer.**

The Figure 9.16 (A) shows water flowing in a branch line and discharging at an open outlet. Water in the line from a to b is at rest with very little pressure exerted on the air that is contained in the chamber. The Figure 9.16 (B) illustrates the condition when the following water is suddenly stopped by the valve. The shocked wave generated rebounds but is absorbed by the air in the chamber. Thus by absorbing the initial shock wave, the water pressure is stabilized and the occasion of water hammer has been removed. Both details are diagrammatic and have been used only for example in showing how the phenomenon of water hammer can be properly controlled.

The extent of water pressure in a given line and the required sizes of air chambers to absorb the resultant shock waves can be computed.

Water hammer pressure can also greatly be reduced by the use of slow closing valves, automatic relief values and surges tanks. It is seldom that any two cause of water hammer are exactly alike. Each must be studied and one or more of the devices available for its suppression must be used alone or in combination with others.

9.13.4 Forces at bends and changes in cross section

A change in direction or magnitude of flow velocity is accompanied by a change in the momentum of water. The force required to produce this change in momentum covers from pressure variation within the water and from the forces transmitted to the water from the pipe walls. For a pipe bend of uniform section.

$$\text{Longitudinal force} = s(\pi d)t \qquad 9.5$$

Where s is the unit stress, d is the diameter of the pipe and t the wall thickness of the pipe.

Similar expression for forces developed for a pipe bend of no uniform a cross-section, pipe contraction and enlargement can be computed. These stresses can be eliminated or reduced by providing an efficient anchorage at the bend, contraction and enlargement.

9.13.5 Forces due to temperature change

Longitudinal stress of considerable magnitude may develop in pipes exposed to large in temperature. The change in length δ of a pipe length L when subjected to a temperature change ΔT is

$$\delta = \propto L \Delta T \qquad 9.6$$

when $\propto$ is the coefficient of the thermal expansion of the pipe material. If this change in length is prevented, longitudinal stresses will develop. From the principles of mechanics of materials it is known that in the elastic range.

$$\text{Deformation, } \Delta = E\varepsilon = B\delta / L \qquad 9.7$$

Where ε is the unit strain (elongation per unit length) and E is the modulus of elasticity and δ is the resulting unit stress. Combining Eqs. 9.6 and 9.7 gives

$$\Sigma = E \propto \Delta T \qquad 9.8$$

This indicates the longitudinal stress that would result when a pipe with fixed ends is subjected to a temperature change. Expansion joints are usually provided to reduce temperature stress.

9.13.6 External forces

An unsupported pipe acts as a beam with loads resulting from the weight of the pipe, weight of water in the pipe, and any other superimposed loads. The stresses resulting from beam action generally termed as flexural stress may be determined by the usual methods of analysis applied to beams. A pipe is a fairly efficient beam section, and stresses resulting from beam action alone are usually negligible except for long spans or when there are large superimposed loads. A rigorous analysis of the combined stresses resulting from internal pressures external loads temperature change and beam action involves application of the principles of elasticity.

Pipes are often placed in an excavated trench which is back filled, or they are laid on ground surface and converted with earth. In either case a vertical load is imposed on the pipe. If a load is superimposed on the fill, a portion of it will be transferred to the buried pipe, the magnitude of the load thus produced depends on the rigidity of the pipe, the pipe of bending and the character of the fill material.

Rigid pipes (concrete, cast iron and vitrified clay) cannot deform materially with our cracking. On the other hand, flexible steel pipe can deform considerably without structural damage. Pipes are usually constructed in ditches or trenches which are excavated in natural soil and then covered by refilling the trench up to the original ground level. Trench condition of pipe installation are shown diagrammatically in Figure 9.17

The vertical load to which a pipe is subjected, when so constructed, is the resultant of two major forces. The first of these is the weight of the prism of soil within the trench and above the top of the pipe, and the second is the friction or shearing forces generated between the prism of soil in the trench and the sides of the trench. The backfill soil has a tendency to settle downwards in relation to the undisturbed soil in which the trench is excavated. The downward movement or tendency for movement induces upward, shearing forces which support a part of the weight of the backfill. The resultant load and the horizontal plane at the top of the pipe and within the width of trench is equal to the weight of the backfill minus these upward shearing forces, as shown in the Figure 9.18

According to Anson Marston, for rigid pipes in narrow trenches, the load in pound per foot of pipe has been found to be

$$w = c\gamma B^2 \qquad 9.9$$

Where B is the trench width at the top of the pipe, γ the specific weight of the fill material, and c is the coefficient characteristic of the fill material, and the ratio of

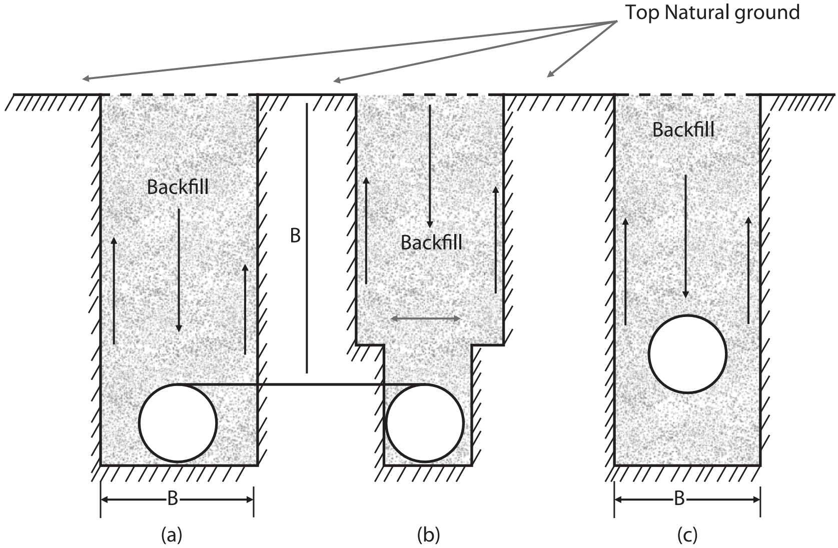


Figure 9.17: **Construction Conditions of Pipes**

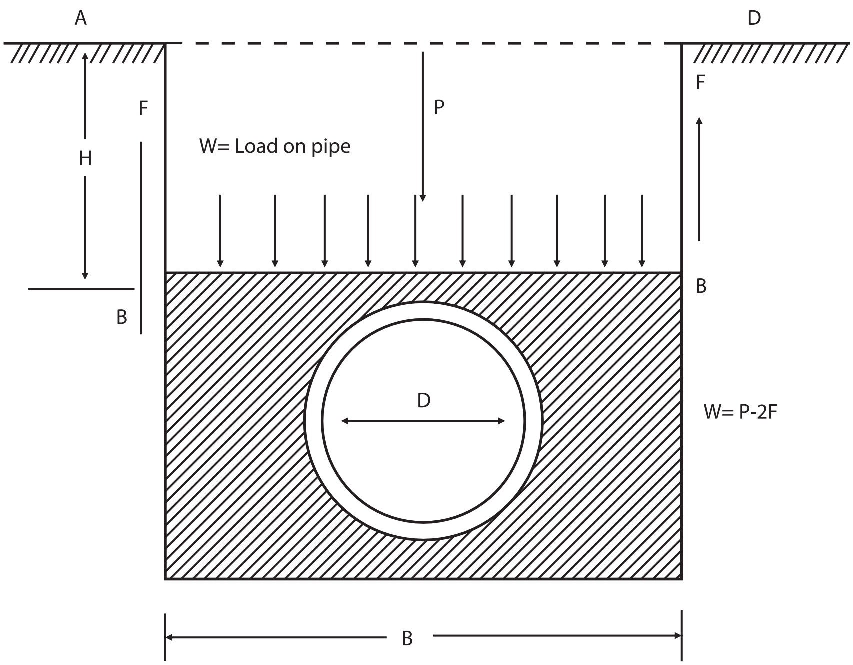


Figure 9.18: **Load Production Forces**

cover depth to the width of the trench, and is generally termed as load calculation coefficient (Table 9.5) The trench-load formula, the Eq. 9.9 gives the total vertical load on horizontal plane at the top of the pipe. Rigid pipe will carry practically all this load. If, on the other hand, the pipe is flexible type, and the soil at the sides is well compacted, the side fills may be expected to carry their proportional share of the total load. As result, the load transmitted to a flexible pipe is less than that for a rigid one. The empirical formula for the load on a buried flexible pipe in a narrow trench is.

$$w = c\gamma BD \qquad 9.10$$

Where D is the outside diameter of the pipe

If a pipe is placed on undisturbed ground and covered with fill (as high = way culvert), the fill adjacent to the pipe is deeper that over the pipe and can, therefore, settle a greater distance (Figure 9.18). Under these conditions, generally referred to as embankment or broad-fill conditions, apportion of the weight of the adjacent prisms of fill is transferred to the central prism by shear, and the load on the pipe is greater than for trench conditions. The equation for the load on a buried pipe under embankment condition is

Table 9.5: **Value of the coefficient c for Eqs. 9.9 and 9.10**

Fill Material Specific weight lb/cu ft	Sand and gravel	Saturated top soil	Clay	Saturated Clay
	100	100	120	130
$\frac{Coverdepth}{Trenchwidth} = \frac{H}{B}$		Values of c		
1.0	0.84	0.86	0.88	0.90
2.0	1.45	1.50	1.55	1.62
3.0	1.90	2.00	2.10	2.20
4.0	2.22	2.33	2.49	2.65
5.0	2.45	2.60	2.78	3.04
6.0	2.60	2.70	3.04	3.33
7.0	2.75	2.95	3.23	3.57
8.0	2.80	3.03	3.37	3.76
9.0	2.88	3.11	3.48	3.92
10.0	2.92	3.17	3.56	4.04
12.0	2.97	3.24	3.68	4.22
14.0	3.00	3.28	3.75	4.34

$$w = c_p \gamma D^2 \qquad 9.11$$

Values of c_p depend on the type of the pipe and the characteristics of the foundation and back fill. Typical values for cp are given in Table 9.6

Critical examinations of the Eq. 9.9 and 9.11 indicate the important influence which the width of the trench exerts on the load. It is seen that the width of the trench at the elevation of the op of pipe is the controlling factor; consequently, the width of the trench should be kept to an absolute minimum consistent with the provision of sufficient working space at the sides of the pipe. To this end, the engineer computing the load, the engineer supervising construction and the contractor actually installing the structure should see eye to eye with respect to the design criteria.

The load on a pipe is also influenced directly by the unit weight of the back fill materials. This value may vary widely for different soils from a minimum of about 100 lb/cu ft to a maximum of about 135 lb/cu ft. The average maximum unit weigh of the soil which with constitute the back fill over the pipe may be determined by actual density measurement in advance of the structural design of the pipe.

Table 9.6: **Values of the coefficient c_p for Eqs. 9.11**

$\frac{Coverdepth}{Trenchwidth} = \frac{H}{B}$	Rigid pipe, Unyielding base noncohesive backfill C_p	Flexible pipe average conditions C_p
1.0	1.2	1.1
2.0	2.0	2.6
3.0	4.0	4.0
4.0	6.7	5.4
6.0	11.0	8.2
8.0	16.0	11.0

Load of Pipe due to super imposed Loads. Two types of superimposed loads are commonly encountered in the structural design of pipes. They are (a) concentrated loads, and (b) distributed loads.

The formulation for load due to superimposed concentrated load is given in the following form by D. H. Holl's integration of Bousainesq's formula as

$$w_{sc} = c_1\ p_{sc}\ I/L \qquad 9.12$$

In which w_{sc} is superimposed concentrated load on the pipe in pounds per foot length p_{sc} is the superimposed concentrated loads in pounds, I is the impact factor, c_1 is the load coefficient

Which is a function of $D/2H$ and $L/2H$ shown in the Table 9.7, where H is the height from the top of the pipe to ground surface in ft, D is the diameter of the pipe in ft, and L, is the effective length of the pipe in ft (Figure 9.19)

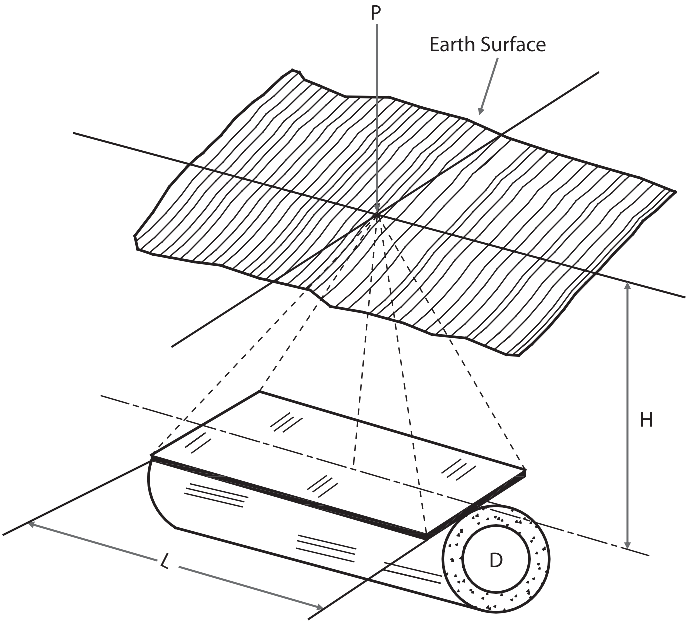


Figure 9.19: **Concentrated Superimposed load vertically centered over the pipe**

In case of a distribute superimposed load as shown in the Figure 9.20, the expression for load on the pipe is

$$w_{sd} = e_1 pID \qquad 9.13$$

Where w_{sd} is the superimposed distributed load on the pipe in Pounds per foot length, p is the intensity of distributed load in lb/sq ft, and e_1 is the load coefficient which is a function of $N/2H$ and $M/2H$ from the table 9.7, where M and N are the length and width respectively in of the over which the distributed load acts.

Table 9.7: Values of load coefficient (e_1) for concentrate and distributed superimpose loads vertically centered over pipes

D/2H or L/2H	M/2H or L/2H									
	0.1	0.2	0.4	0.6	0.8	1.0	1.2	1.5	2.0	5.0
0.1	0.019	0.037	0.053	0.089	0.103	0.112	0.117	0.121	0.124	0.128
0.2	0.037	0.072	0.109	0.252	0.203	0.219	0.229	0.238	0.244	0.248
0.4	0.067	0.131	0.190	0.320	0.373	0.405	0.425	0.440	0.454	0.460
0.6	0.089	0.174	0.252	0.428	0.499	0.544	0.572	0.596	0.613	0.460
0.8	0.103	0.202	0.292	0.499	0.584	0.639	0.674	0.703	0.725	0.740
1.0	0.112	0.219	0.318	0.544	0.639	0.701	0.740	0.773	0.800	0.816
1.2	0.117	0.229	0.333	0.572	0.674	0.740	0.783	0.820	0.849	0.868
1.5	0.121	0.238	0.345	0.596	0.703	0.774	0.820	0.961	0.894	0.916
2.0	0.124	0.244	0.355	0.613	0.725	0.800	0.849	0.896	0.930	0.956

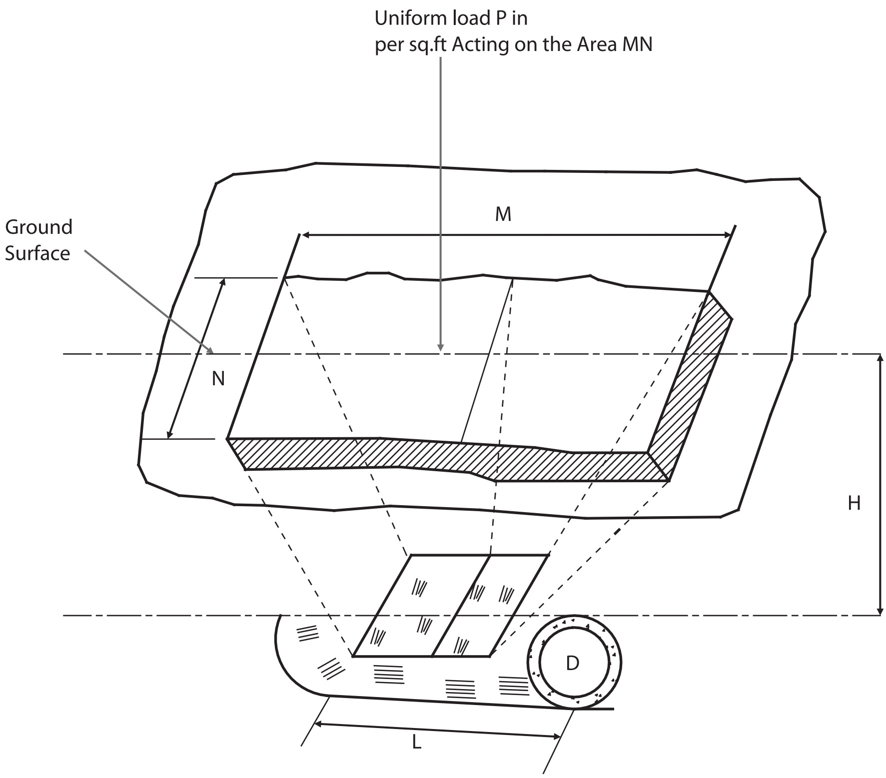


Figure 9.20: Distributed superimposed load vertically centered over pipe

Traffic vehicles which cause loads on the pipes produce mostly dynamic loads. Suggested impact factors for various kinds of traffic are as follows.

Highways	1.50
Railways	1.75
Airfields	1.00

Example 9.4 : What is the probable maximum load on a pipe laid in a trench that is 3.5 ft wide if the depth of the fill above the top of the pipe is 9 ft and the filling material is wet sand? (γ = 120 lb/cu ft.)

Solution:

Here H = 9 ft. B=3.5 ft. H/B= 9.5/3.5=257.From Table 9.5, the coefficient c for sand and for this value of H/B is 1.73 (calculated).

$$: \text{Load on the pipe} = c_{\gamma}B^2$$
$$= 1.73 \times 120 \times (3.5)^2$$
$$= 2540 \text{ pounds per linear foot.}$$

Example 9.5: A 3 ft diameter steel pipe is buried on a trench 4 ft wide. The backfill is clay (γ = 120 lb/cu ft) and the top of the pipe is 6 ft below the surface of the fill. Calculate the total load on the pipe. Take the value of c as 1.2.

Solution:

$$\text{Load on the pipe} = c_{\gamma}BD = 1.2 \times 120 \times 4 \times 3$$
$$= 1730 \text{ lbs per linear foot.}$$

Example 9.6: An 8 ft. diameter rigid concrete pipe rests on unyielding ground and is covered with sand (γ = 100 lb/cu ft) to depth of 6 ft. Calculate the pressure exerted by the fill material on the pipe. Take the value of c_p as 0.9.

Solution:

$$\text{Pressure exerted by the fill} = c_p\gamma D^2 = 0.9 \times 100 \times (8)^2$$
$$= 5760 \text{ lb/ft.}$$

9.14 Strength of Pipe

Because of the complex nature of the combained stress in pipes, the stresses are rarely analyzed in detail except for large and important pipe lines. Structurally, pressure pipes must resist the following forces singly or in combination.

- Internal pressure equal to the full head of water to which the pipe can be subjected.

- Unbalanced pressures at bends, contractions, and closures.
- Water hammer or increased internal pressure caused by sudden reduction in the velocity of water by rapid closing of a gate or valve or shutdown of pump for example.
- External load in the form of backfill, traffic and their own weights between external supports.
- Temperature induced expansion and contraction.

According to ASTM standards the pipes are tested for crushing strength by two methods

- Sand Bearing Test
- Three-edge Bearing test. Strength from sand bearing tests will be 50 percent more than those for three edge bearing test. Table 9.8 presents ASTM standards for crushing strength of various types of pipes in three edge bearing test.

Table 9.8: **Crushing strength of clay and concrete pipes by the three-edge bearing test: (All strength in pounds per linear foot)**

Internal Diameter in inch	Clay	Plain Concrete	Reinforced Concrete			
			Class I	Class II	Class III	Class IV
4	1000	1500	15000			
6	1200	1600				
8	1200	1800				
10	1400	2400				
12	1500	2500	1500	2000	3000	3750
15	1750	2750	1875	2500	3750	4700
18	2000	3300	2250	3000	4500	5600
21	2200	3700	2750	3500	5200	6500
24	2400	4000	3000	4000	6000	7500
27	2750		3300	4500	6700	8400
30	3200		3750	5000	7500	9400
36	4000		4500	5000	9000	11250
42			5250	7000	18500	13200
48			6000	8000	12000	15000
60			7500	10000	15000	18000
72			9000	12000	18000	22000

Field Supporting Strength = load factor × three edge bearing strength

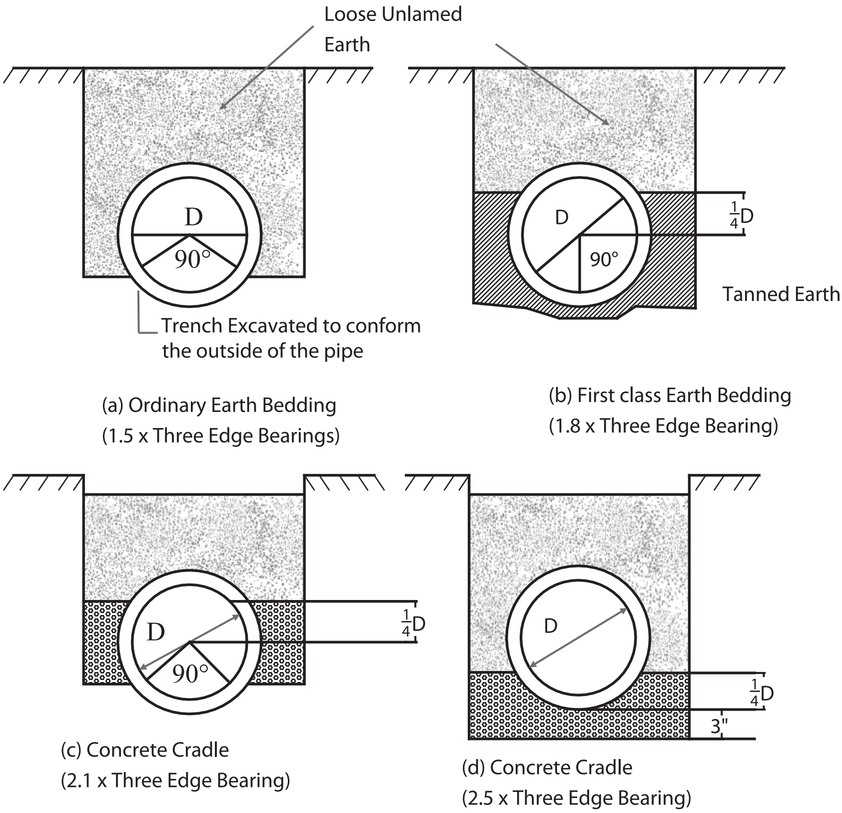


Figure 9.21: **Some Methods of Laying Pipes**

The Figure 9.21 shows several types of bending and indicates the strength ratio, i.e., the factors by which the three edge bearing strength is multiplied to find the effective strength in the field. In general, all buried pipes should be placed on a bed which has been rounded to fit the pipe and the backfill should be carefully placed and thoroughly tamed around and above the top of the pipe. This is specially true of flexible pipes since the lateral pressure of the backfill adds appreciably for the strength of the pipe.

The factor of safety against ultimate failure is generally at least 2.5 in the design of most engineering structures of monolithic concrete. The factor of safety of pipes against ultimate collapse is considerably less. It is therefore important to guarantee that loads imposed on the buried pipes are not greater than the design loads. In order to attain this objective the following procedures should be strictly followed:

- Specifications should strictly limit the width of the trench to the maximum used in design calculations. The maximum allowable clearance specified should not be exceded under any circumstances.
- Construction should be under the supervision of an experience engineer.
- Pipe testing should be done under the supervision of a skilled technician in a testing laboratory and close liaison should be maintained between the laboratory and the field engineer.
- The field engineer and the design engineer should confer time to time so that any deviation from the design specifications is immediately corrected or compensated for.

9.15 The Joints

The pipes are required to joint together pipes which are available in smaller lengths, say 6, 10, 12, 15 and 20 ft. only. The requisites of a jointing material are

- Imperviousness
- Elasticity
- Strength
- Durability
- Adhesiveness
- Availability
- Workability
- Economy

There are various types of joints of which the powered joint, spigot and socket joint, flanged joint, screwed and socketed joint are important.

9.16 Pipe Laying

Operation involved in the laying of pipe - lines include the following steps:

- **Preparation of detailed maps of roads and streets:** Showing position of curbs, gutters, other underground service lines-sewers, existing water pipe (if there is any), gas pipes, telephone and electric conduits.

- **Locating the proposed alignment on the ground:** The trench line is marked by driving centrally stakes 100 ft apart on straight reaches and 25 to 50 ft apart on curves.

- **Excavation trenches:** with width sufficient to allow the pipes to be properly laid and jointed, and with depth sufficient to give adequate protection to the pipes against impact of traffic and other factors, Width is usually, kept 12″ to 18″ more than the outside diameter of the pipe and depth such as to give a ground cover of about 3 ft. form the top of the barrel of the pipe.
- **Preparation of the bottom of the trench excavated:** The bottom of the trench should be carefully prepared so that the barrel of the pipe can bedded true to line and gradient for its entire length on a firm surface. In many cases, a bed of concrete 6″ thick would provide a hard and even surface and adequate protection against possible settlement. Joint-holes should be left in the bed at suitable intervals to assist in the jointing of pipes where necessary.
- **Lowering of pipes into the trench:** Pipes stacked on either side of the trench after transported to the site should be gently and carefully lowered into the trench so as not to damage thin outer protective coatings or their ends. Before lowering, pipes should be wiped clean to remove any dirt or foreign matter sticking to them.
- **Laying pipes:** Pipes are seldom laid with flat slope parallel to the hydraulic gradient. This is to avoid any air lock troubles. Every pipe length should, therefore, be laid either with a continuous fall to low points. Pipe laying should proceed in an uphill direction to facilitate joint making.
- **Jointing pipes:** It should conform to the operation and specification of pipe jointing.
- **Anchoring of pipes:** At all bends, tees, valves and other branch connections, it should be necessary to provide thrust blocks of concrete to transmit the hydraulic thrust and distribute it over a wider area of the ground. Where the hydraulic trust is upwards as in case of pipes on sloping grounds, anchor block of concrete would be required to be provided at regular intervals and pips should be firmly secured to them with steel straps.
- **Back filling or refilling the trench with the excavated material:** The material surrounding the pipes must be soft and laid preferably in layer of 6″ to 12″ thickness, well rammed so as to resist subsequent movement of the pipes, The reaming upper portion of upper portion of the trench may be

refilled as before with the excavated material and the brought flush with the road level or a little projecting above it for later consolidation by the traffic.

- **Pipe testing:** After laying and jointing and before backfilling the pipe is required to be tested under pressure. The test consists of filling the pipeline with water expelling all air from within, allowing it to stand full for some time and the applying the test pressure of about 70 psi. The pressure is applied by means of a manually operated test pump fitted with a pressure gauge. The test is generally carried out in sections the pipe-laying proceeds. The open end of the pipe is closed for testing by fitting a suitable water light plug.

9.17 Distribution System Design

9.17.1 Minimum pipe cover

Minimum cover over pipes will be 2½ feet in grassed areas, 3 feet under unpaved driveways or roadways, and 4 feet under railroad tracks. Where frost depths are greater than the above minimums, the cover should be at least equal to the frost depth, particularly for small lines which may not be flowing continuously. Where lines pass under railroads, pipes may be encased in concrete or enclosed in rigid conduit in accordance with the standard practice of the affected railroad. Installation of pipelines and conduits under railroad main lines is usually accomplished by carefully controlled tunnelling and jacking. For branch lines or lines used infrequently, open cut installation may be permitted by the railroad. Jacking or tunnelling procedures are usually required if a pipeline is to be installed under a major roadway with no disruption of traffic.

9.17.2 Protection of items penetrating frost zone

Water distribution equipment items penetrating the frost zone are sometimes subject to freezing if protective measures aren't taken. Air vent and vacuum relief valves, blowoff valves, or fire hydrants are par ticularly susceptible. Freezing should not be a problem with post-indicator valves and valve boxes if they are constructed and maintained so that water doesn't collect in or around them.

- **Air vent and vacuum relief valves:** These items can be protected from freezing by installation in pits deep enough to place the valves below the frost zone or by providing heat with electric space heaters, electric heating tape, or other suitable means.

- **Blowoff valves:** Blowoff valves should be installed at depths below the frost zone. If terrain conditions permit, the drain line from a blowoff valve should go to a nearby low surface area to allow gravity drainage. The valve discharge must be piped to the atmosphere and drainage provided from the line to the outlet side of the valve. If gravity drainage can't be provided, the blowoff valve should be provided a tee, with foot valve to prevent backflow, discharging into a dry well below the frost line. Alternatives permitting drainage without contamination from ground water or other nonpotable water may be used subject to approval of the Contracting Officer.

- **Fire hydrants:** Fire hydrants penetrating the frost zone will be of the dry barrel variety. Free draining backfill will be placed around the barrel to prevent frost-heave due to moisture around the barrel in the frost zone.

9.17.3 Cleaning and lining existing water mains

- Where incrustations and tubercules have formed on the inside of mains to such a degree that flow through the mains has been materially reduced, it may be advantageous to clean and line these mains. Cleaning is usually accomplished with special mechanical scraping devices which are pulled through the main with a cable or forced through the main by hydraulic pressure. Large mains, over 30 inches, can be cleaned by electrically driven, manually operated machines with rotating scraper blades.
- After a distribution main has been cleaned, it must be lined with cement mortar or a similar substance. The lining is applied by a special machine on wheels which is pulled through the main and fed the viscous lining substance under pressure. The lining is centrifugally sprayed onto the interior walls of the pipe by the machine, and the finish is smoothed by special mechanized trowels. The applied linings generally vary in thickness from $3/16$ to $1/4$ inch, but as little as $1/8$ inch might be applied to small diameter mains. In places where the lining cannot be applied and troweled by machine, hand application is necessary. During the cleaning and lining of mains, precautions must be taken to protect any valves, hydrants, or branch mains attached to the main being treated.
- The principal advantages of cleaning and lining of old mains are that the frictional resistance to flow is reduced, thereby increasing flows and pressures; and the resistance of the pipe material to corrosion is improved. However, before a program of main cleaning and relining is initiated, the

relative cost and service life must be compared to the complete replacement of deteriorated mains, and the most cost-effective alternative selected. Standards for the performance of this work are given in AWWA C602.

9.17.4 Disinfection of water supply system

Disinfection of new distribution mains and disinfection of existing distribution piping affected by construction, or system modifications by construction contract will be in accordance with AWWA C601. In no event will any of the above piping be placed in service prior to verification of disinfection by bacteriological tests as required and evaluated by the supporting medical (health) authority.

9.17.5 External corrosion

- Corrosion of the external surfaces of cast-iron or steel pipes can, under some conditions, be a significant problem. Therefore, ductile-iron or steel pipelines placed in corrosive soils must be protected by coatings of coal-tar or cement mortar. Standards for coal-tar coatings are given in AWWA C203 and AWWA C209. Cement mortar coatings may be applied by mechanical or pneumatic means and should meet the guidelines in AWWA C 205.
- The characteristics of the soil in which a pipe is placed affect the rates of corrosion, with the most corrosive soils being those having poor aeration and high values for acidity, electrical conductivity, dissolved solids, and moisture content. The relative potential for corrosion may be estimated by evaluating the degree of corrosion of pipelines or other metallic objects previously buried in that soil.
- In locations where the soils are known to be very corrosive, it may be desirable to use cathodic protection systems as a supplement to (but not in place of) the above coatings.
- Another method of avoiding corrosion of distribution mains is through the use of non-metallic pipe materials such as asbestos-cement, reinforced concrete, or plastic.

9.17.6 Layout map

An up-to-date layout map, to a suitable scale, showing the entire distribution system involved in the design will be maintained.

9.17.7 Design analysis

The design analysis will indicate the essential elements used in determining sizes and locations of mains, including:

- Projected populations and areas in which the populations are located.
- Locations and magnitudes of special water demands
- Location and magnitude of fire demands
- Location and size of pump stations
- Storage input
- Water treatment plant or other input sources

9.18 Main Concepts and Definitions

The basic hydraulic principles applied in water transport and distribution practice emerge from three main assumptions:

- The system is filled with water under pressure,
- that water is incompressible,
- that water has a steady and uniform flow

In addition, it is assumed that the deformation of the system boundaries is negligible, meaning that the water flows through a non-elastic system.

Flow Q (m^3/s) through a pipe cross-section of area A (m^2) is determined as $Q = v \times A$, where v (m/s) is the mean velocity in the cross-section. This flow is steady if the mean velocity remains constant over a period of time Δt. If the mean velocities of two consecutive cross-sections are equal at a particular moment, the flow is uniform. The earlier definitions written in the form of equations for two close moments, t_1 and t_2, and in the pipe cross-sections 1 and 2 (Figure 9.22) yield:

$$v_1^{(t_1)} = v_1^{(t_2)} \wedge v_2^{(t_1)} = v_2^{(t_2)} \qquad 9.14$$

For a steady flow, and:

$$v_1^{(t_1)} = v_2^{(t_2)} \wedge v_1^{(t_1)} = v_2^{(t_2)} \qquad 9.15$$

For a uniform flow

A steady flow in a pipe with a constant diameter is at the same time uniform, thus:

$$v_1^{(t_1)} = v_2^{(t_2)} \wedge v_1^{(t_1)} = v_2^{(t_2)} \qquad 9.16$$

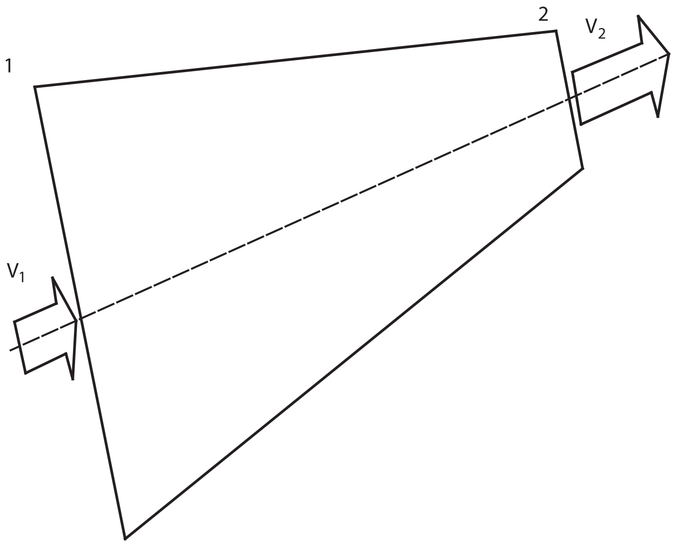


Figure 9.22: **Steady and uniform flow.**

The earlier simplifications help to describe the general hydraulic behaviour of water distribution systems assuming that the time interval between t_1 and t_2 is sufficiently short. Relatively slow changes of boundary conditions during regular operation of these systems make Δt of a few minutes acceptably short for the assumptions introduced earlier. At the same time, this interval is long enough to simulate changes in pump operation, levels in reservoirs, diurnal demand patterns, etc., without handling unnecessarily large amounts of data. If there is a sudden change in operation, for instance a situation caused by pump failure or valve closure, transitional flow conditions occur in which the assumptions of the steady and uniform flow are no longer valid. To be able to describe these phenomena in a mathematically accurate way, a more complex approach elaborated in the theory of transient flows would be required, which is out of the contents of in this book.

9.18.1 Conservation laws

The conservation laws of mass, energy and momentum are three fundamental laws related to fluid flow. These laws state:

- The Mass Conservation Law Mass m (kg) can neither be created nor destroyed; any mass that enters a system must either accumulate in that system or leave it.
- The Energy Conservation Law Energy E (J) can neither be created nor destroyed; it can only be transformed into another form.
- The Momentum Conservation Law The sum of external forces acting on a fluid system equals the change of the momentum rate M (N) of that system.

The conservation laws are translated into practice through the application of three equations, respectively:

- The Continuity Equation.
- The Energy Equation.
- The Momentum Equation.

The Continuity Equation is used when balancing the volumes and flows in distribution networks. Assuming that water is an incompressible fluid, i.e. with a mass density $p = m/V =$ const, the Mass Conservation Law can be applied to volumes. In this situation, the following is valid for tanks (see Figure 9.23):

$$Q_{inp} = Q_{out} \pm \frac{\Delta V}{\Delta t} \qquad 9.17$$

Where $\Delta V/\Delta t$ represents the change in volume V (m^3) within a time interval Δt(s). Thus, the difference between the input and output flow from a tank is the volume that is accumulated in the tank if $Q_{out} < Q_{inp}$ (sign + in Equation 9.18)

$$\sum_{i=1}^{j} Q_i - Q_n = 0 \qquad 9.18$$

Where on represents the nodal discharge. An example of the three pipes and a discharge point is shown in figure 9.24

The Energy Equation establishes the energy balance between any two cross-sections of a pipe:

$$E_1 = E_2 \pm \Delta E \qquad 9.19$$

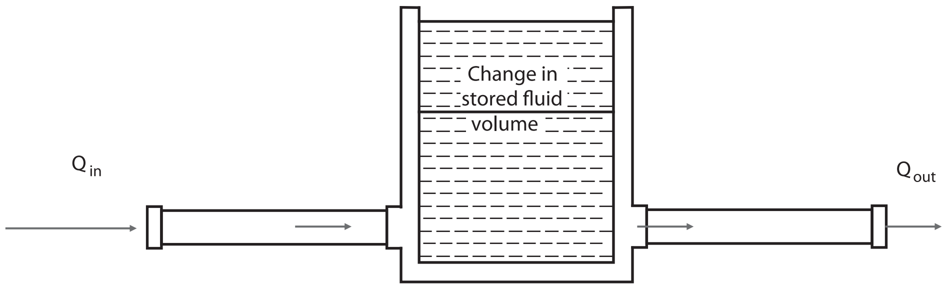


Figure 9.23: **The Continuity Equation validity in tanks.**

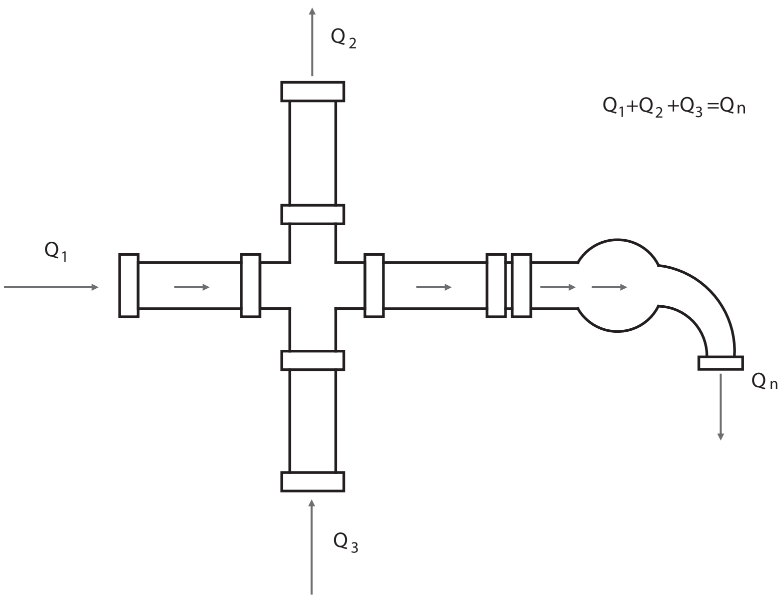


Figure 9.24: **The Continuity Equation validity in pipe junctions.**

Where, ΔE is the amount of transformed energy between cross-sections 1 and 2. It is usually the energy lost from the system (the sign + in Equation 9.19), but may also be added to it by pumping of water (sign -).

The Momentum Equation (in some literature also known as the Dynamic Equation) describes the pipe resistance to dynamic forces caused by the pressurized flow. For incompressible fluids, momentum M (N) carried across a pipe section is defined as:

$$M = \rho Q v \qquad 9.20$$

Where ρ (kg/m^3) represent the mass density of water, Q(m^3/s) is the pipe flow, v (m/s) is the mean velocity. Other forces in the equilibrium are:

- Hydrostatic force F_h (N) caused by fluid pressure p (N/m^2 or Pa); $F_h = p \times A$.
- Weight w (N) of the considered fluid volume (only acts in a vertical direction).
- Force F (N) other solid surface acting on the fluid.

The Momentum Equation as written for a horizontal direction would state:

$$\rho Q v_1 - \rho Q v_2 \cos\varphi = -p_1 A_1 + p_2 A_2 \cos\varphi + Fx \qquad 9.21$$

Whereas in a vertical direction:

$$\rho Q v_2 \sin\varphi = -p_2 A_2 \sin\varphi + w + Fy \qquad 9.22$$

The forces of the water acting on the pipe bend are the same, i.e, F_x and F_y but with an opposite direction i.e, a negative sign, in which case the total force, known as the pipe thrust will be:

$$F = \sqrt{F_x^2 + F_y^2} \qquad (9.23)$$

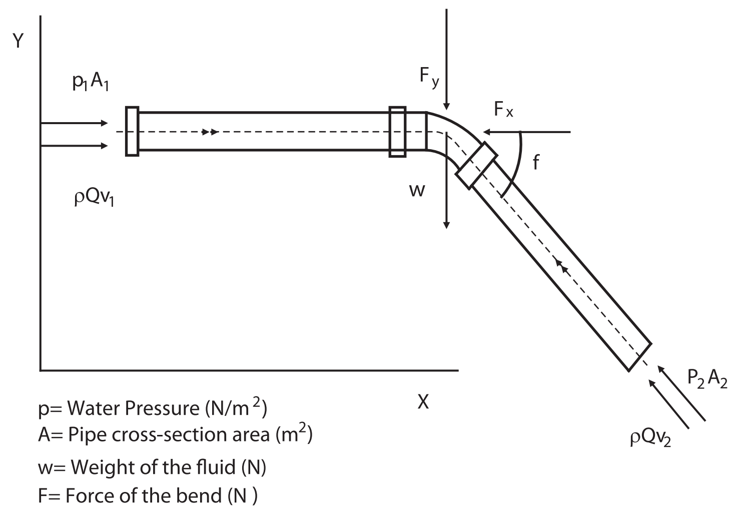


Figure 9.25: The Momentum Equation.

The Momentum Equation is applied in calculations for the additional strengthening of pipes, in locations where the flow needs to be diverted. The results are used for the design of concrete structures required for anchoring of pipe bends and elbows (Figure 9.25).

Example 9.7: A velocity of 1.2 m/s has been measured in a pipe of diameter D = 600 mm. Calculate the pipe flow.

Solution:

The cross-section of the pipe is

$$A = \frac{D^2\pi}{4} = \frac{0.6^2 \times 3.14}{4} = 0.2827 m^2$$

Which yields the flow of

$$D = vA = 1.2 \times 0.2827 = 0.339 m^3 / s \approx 3401 / s$$

Example 9.8: A circular tank with a diameter at the bottom of D = 20 m and with vertical walls has been filled with a flow of 240 m^3/h. What will be the increase of the tank depth after 15 minutes, assuming a constant flow during this period of time?

Solution:

The tank cross-section area is:

$$A = \frac{D^2\pi}{4} = \frac{20^2 \times 3.14}{4} = 314.16 m^2$$

The flow of 240 m^3/h fills the tank with an additional 60 m^3 after 15 minutes, which is going to increase the tank depth by a further 60/314.16 = 0.19 m ≈ 20 cm.

Example 9.9: For a pipe bend of 45° and a continuous diameter of D = 300 mm, calculate the pipe thrust if the water pressure in the bend is 100 kPa at a measured flow rate of 26 l/s. The weight of the fluid can be neglected. The mass density of the water equals p = 1000 kg/m^3.

Solution:

From Figure for a continuous pipe diameter:

$$A_1 = A_2 = \frac{D^2\pi}{4} = \frac{0.3^2 \times 3.14}{4} = 0.07m^2$$

Consequently, the flow velocity in the bend can be calculated as:

$$v_1 = v_2 = \frac{Q}{A} = \frac{0.026}{0.07} = 0.37m/s$$

Furthermore, for the angle $\varphi = 45^\circ$, $\sin\varphi = \cos\varphi = 0.71$. Assuming also that $p_1 = p_2$ = 100 kpa (or 100,000 N/m^2), the thrust force in the X-direction becomes:

$$-F_x = 0.29 \times (pA + pQv) = 0.29 \times (100{,}000 \times 0.07 + 1000 \times 0.026 \times 0.37) \approx 2030N = 2kN$$

While in the Y-direction:

$$-F_y = 0.71 \times (pA + \rho Qv) \approx 5kN$$

The total force will therefore be:

$$F = \sqrt{2^2 + 5^2} \approx 5.4kN$$

The calculation shows that the impact of water pressure is much more significant that the one of the flow/velocity.

9.18.2 Energy and hydraulic grade lines

The energy balance Equation stands for total energies in two cross-sections of a pipe. The total energy in each cross-section comprises three components, which is generally written as:

$$E_{tot} = mgZ + m\frac{p}{\rho} + \frac{mv^2}{2} \qquad 9.24$$

expressed in J or more commonly in kWh. Written per unit weight, the equation looks as follows:

$$E_{tot} = Z + \frac{p}{\rho g} + \frac{v^2}{2g} \qquad 9.25$$

where the energy obtained will be expressed in *metres water column* (mwc). Parameter g in both these equations stands for gravity (9.81 m/s^2).

The first term in Equations 9.24 and 9.25 determines the *potential energy*, which is entirely dependent on the elevation of the mass/volume. The second term stands for the flow energy that comes from the ability of a fluid mass $m = \rho \times V$ to do work W (N) generated by the earlier mentioned pressure forces $F = p \times A$. At pipe length L, these forces create the work that can be described per unit mass as:

$$W = FL = \frac{pAL}{\rho V} = \frac{p}{\rho} \qquad 9.26$$

Finally, the third term in the equations represents the kinetic energy generated by the mass/volume motion.

By plugging 9.25 into 9.19, it becomes:

$$Z_1 + \frac{p_1}{\rho g} + \frac{v_1^2}{2g} = Z_2 + \frac{p_2}{\rho g} + \frac{v_2^2}{2g} \pm \Delta E \qquad 9.27$$

In this from, the energy equation is known as the *Bernoulli Equation*. The equation parameters are shown in Figure 9.26. The following terminology is in common use:

- Elevation head: $Z_{1(2)}$
- Pressure head: $p_{1(2)}/\rho g$
- Piezometric head: $H_{1(2)} = Z_{1(2)} + p_{1(2)/\rho g}$
- Velocity head: $v^2_{1(2)}/2g$
- Energy head: $E_{1(2)} = H_{1(2)} + v^2_{1(2)}/2g$

The pressure and velocity heads are expressed in mwc, which gives a good visual impression while talking about 'high-' or 'low' pressures or energies. The elevation-, piezometric- and energy heads are compared to a reference or 'zero' level. Any level can be taken as a reference; it is commonly the mean sea level suggesting the units for *Z, H* and *E* in metres above mean sea level (msl). Alternatively, the street level can also be taken as a reference level.

To provide a link with the SI-units, the following is valid, where mwc represents meter of water column:

- 1 mwc of the pressure head corresponds to 9.81 kPa in SI-units, which for practical reasons is often rounded off to 10 kPa.

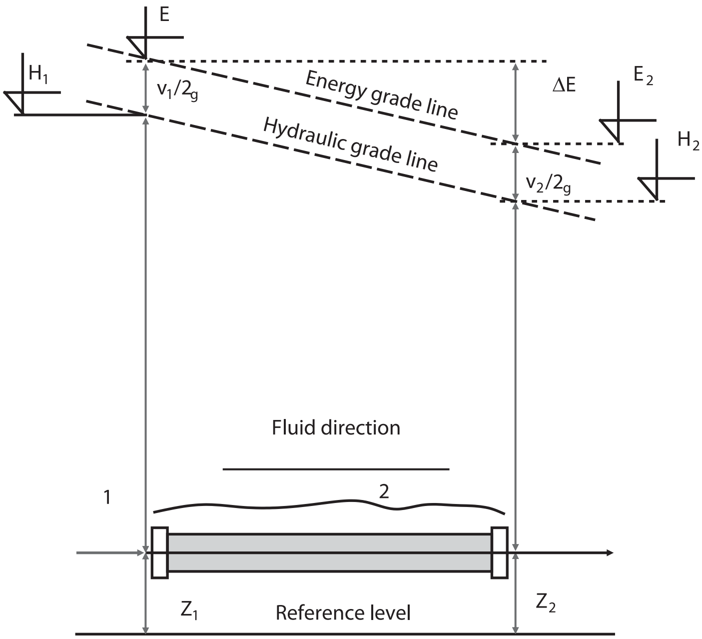


Figure 9.26: **The Bernoulli Equation.**

- 1 mwc of the potential energy corresponds to 9.81 (≈10) kJ in SI-units; for instance, this energy will be possessed by 1 m^3 of the water volume elevated 1 m above the reference level.
- 1 mwc of the kinetic energy corresponds to 9.81 (≈10) kJ in SI-units; for instance, this energy will be possessed by 1 m^3 of the water volume flowing at a velocity of 1 m/s.

In reservoirs with a surface level in contact with the atmospherepressure p equals the atmospheric pressure.

Therefore, $p = p_{atm} \approx 0$.

Furthermore, the velocity throughout the reservoir volume can be neglected ($v \approx 0$ m/s). As a result, both the energy and piezometric head will be positioned at the surface of the water. Hence, $E_{tot} = H = Z$. The lines that indicate the energy- and piezometric-head levels in consecutive cross-sections of a pipe are called the energy grade line and the hydraulic grade line, respectively.

The energy and hydraulic grade line are parallel for uniform flow conditions.

Furthermore, the velocity head is in reality considerably smaller than the pressure head. For example, for a common pipe velocity of 1 m/s, $v^2/2g = 0.05$ mwc, while the pressure heads are often in the order of tens of metres of water column. Hence, the real difference between these two lines is, with a few exceptions, negligible and the hydraulic grade line is predominantly considered while solving practical problems. Its position and slope indicate:

- the pressures existing in the pipe, and
- the flow direction.

The hydraulic grade line is generally not parallel to the slope of the pipe that normally varies from section to section. In hilly terrains, the energy level may even drop below the pipe invert causing negative pressure (below atmospheric), as Figure 9.27a shows.

The slope of the hydraulic grade line is called the hydraulic gradient, $S = \Delta E/L = \Delta H/L$, where L (m) is the length of the pipe section. This parameter reflects the pipe conveyance (Figure 9.27b).

The flow rate in pipes under pressure is related to the hydraulic gradient and not to the slope of the pipe. More energy is needed for a pipe to convey more water, which is expressed in the higher value of the hydraulic gradient.

Example 9.10: For the pipe bend in Example 9.9, calculate the total energy- and piezometric head in the cross-section of the bend if it is located at $Z = 158$ msl. Express the result in msl, J and kWh.

Solution:

In Example 9.9, the pressure indicated in the pipe bend was $p = 100$ kPa, while the velocity, calculated from the flow rate and the pipe diameter, was $v = 0.37$ m/s. The total energy can be determined from Equation 9.25:

$$E_{tot} = Z + \frac{p}{\rho g} + \frac{v^2}{2g} = 158 + \frac{100,000}{1000 \times 9.81} + \frac{0.37^2}{2 \times 9.81}$$

$$= 158 + 10.194 + 0.007 = 168.2 \text{ msl}$$

As can be seen, the impact of the kinetic energy is minimal and the difference between the total energy and the piezometric head can therefore be neglected. The same result in J and kWh is as follows:

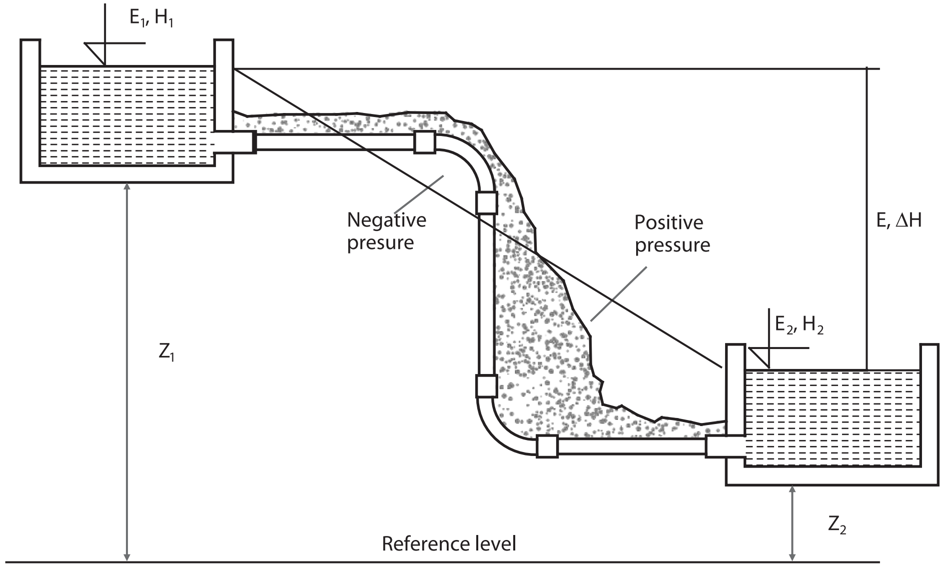


Figure 9.27a: Hydraulic grade line.

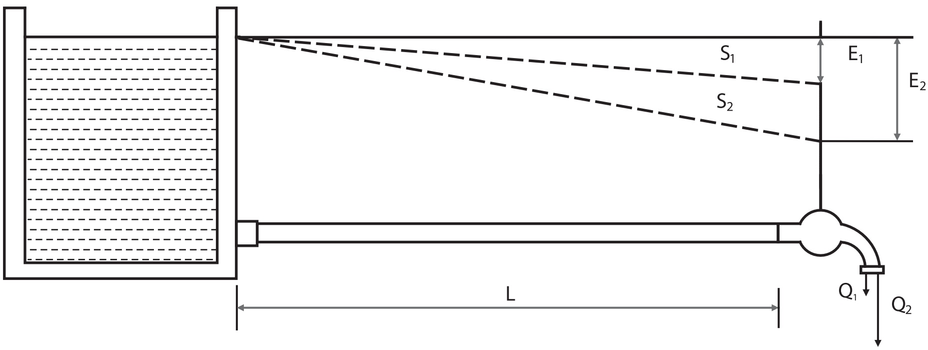


Figure 9.27b: The hydraulic gradient.

$$E_{tot} = 168.2 \times 1000 \times 9.81 = 1{,}650{,}042J \approx 1650\,kJ\,(or\,kWs)$$

$$\frac{1650}{3600} \approx 0.5\,kWh$$

For an unspecified volume, the above result represents a type of unit energy, expressed per m^3 of water. To remember the units conversion:

$$1\,N = 1\,kg \times m/s^2 \text{ and } 1\,J = 1\,N \times m.$$

9.19 Hydraulic Losses

The energy loss ΔE from Equation 9.27 is generated by:

- friction between the water and the pipe wall,
- turbulence caused by obstructions of the flow.

These causes inflict the friction- and minor losses, respectively. Both can be expressed in the same format:

$$\Delta E = h_f + h_m = R_f Q^{n_f} + R_m Q^{n_m} \qquad 9.28$$

where, R_f stands for resistance of a pipe with diameter D, along its length L. The parameter R_m can be characterised as a resistance at the pipe cross-section where obstruction occurs. Exponents n_f and n_m depend on the type of equation applied.

9.20.1 Friction losses

The most popular equations used for the determination of friction losses are:

- the Darcy–Weisbach Equation,
- the Hazen–Williams Equation,
- the Manning Equation.

Following the format in Equation 9.28:

Darcy-Weisbach

$$R_f = \frac{8\lambda L}{\pi^2 g D^5} = \frac{\lambda L}{12.1 D^5}; n_f = 2 \qquad 9.29$$

Hazen-Williams

$$R_f = \frac{10.68 L}{C_{hw}^{1.852} D^{4.87}}; n_f = 1.852 \qquad 9.30$$

Manning

$$R_f = \frac{10.29 N^2 L}{D^{16/3}}; n_f = 2 \qquad 9.31$$

In all three cases, the friction loss h_f will be calculated in mwc for the flow Q expressed in m^3/s and length L and diameter D expressed in m. The use of prescribed parameter units in Equations 9.29–9.31 is to be strictly obeyed as the constants will need to be readjusted depending on the alternative units used. In the above equations, λ, C_{hw} and N are experimentally-determined factors that describe the impact of the pipe wall roughness on the friction loss.

In the Darcy–Weisbach Equation, the friction factor λ (-) (also labeled as f in some literature) can be calculated from the equation of Colebrook– White:

$$\frac{1}{\sqrt{\lambda}} = -2\log\left[\frac{2.51}{\mathrm{Re}\sqrt{\lambda}} + \frac{k}{3.7D}\right] \qquad 9.32$$

Where k is the absolute roughness of the pipe wall (mm), D the inner diameter of the pipe (mm) and Re the Reynolds number (-).

To avoid iterative calculation, Barr (1975) suggests the following acceptable approximation, which deviates from the results obtained by the Colebrook-White Equation for ± 1%:

$$\frac{1}{\sqrt{\lambda}} = -2\log\left[\frac{5.1286}{\mathrm{Re}^{0.89}} + \frac{k}{3.7D}\right] \qquad 9.33$$

The Reynolds number describes the flow regime. It can be called as:

$$Re = \frac{vD}{\mu} \qquad 9.34$$

Where μ (m^2/s) stands for the *kinematic viscosity*. This parameter depends on the water temperature and can be determined from the following equation:

$$\mu = \frac{497\times10^{-6}}{(T+42.5)^{1.5}} \qquad 9.35$$

For T expressed in °C

The flow is:

- Laminar, if $Re < 2000$.
- Critical (in transition), for Re ≈2000 – 4000,
- Turbulent, if $Re > 4000$.

The turbulent flows are predominant in distribution networks under normal operation. For example, within a typical range for the following parameters: v = 0.5–1.5 m/s, D = 50–1500 mm and T = 10–20°C, the Reynolds number calculated by using Equations 9.34 and 9.35 has a value of between 19,000 and 225, 0000. The friction factor for the laminar flow conditions is then calculated as:

$$\lambda = \frac{64}{Re} \qquad 9.36$$

As it usually results from very low velocities, this flow regime is not favourable in any way. Once *Re*, *k* and *D* are known, the λ-factor can also be determined from the Moody diagram, shown in Figure 9.28. This diagram is in essence a graphic presentation of the Colebrook–White Equation. In the turbulent flow regime, Moody diagram shows a family of curves for different *k/D* ratios. This zone is split in two by the dashed line. The first sub-zone is called the *transitional turbulence zone*, where the effect of the pipe roughness on the friction factor is limited compared to the impact of the Reynolds number (i.e. the viscosity).

The curves in the second sub-zone of the rough (developed) turbulence are nearly parallel, which clearly indicates the opposite situation where the Reynolds number has little influence on the friction factor. As a result, in this zone the Colebrook–White Equation can be simplified:

$$\frac{1}{\sqrt{\lambda}} = -2\log\left[\frac{k}{3.7D}\right] \qquad 9.37$$

For typical values of *v, k, D* and *T*, the flow rate in distribution pipes often drops within the rough turbulence zone.

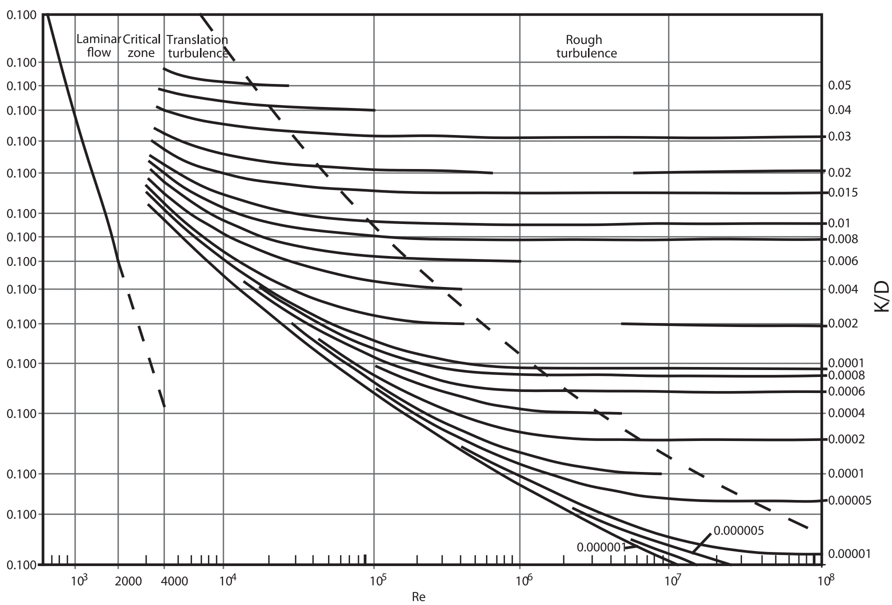


Figure 9.28: **Moody diagram.**

The *absolute roughness* is dependant upon the pipe material and age. The most commonly used values for pipes in good condition are given in Table 9.8. With the impact of corrosion, the *k*-values can increase substantially. In extreme cases, severe corrosion will be taken into consideration by reducing the inner diameter.

Table 9.8. **Absolute roughness.**

Pipe Material	k (mm)
Asbestos cement	0.015-0.03
Glavanised/coated cast iron	0.06-0.3
Uncoated cast iron	0.15-0.6
Ductile iron	0.03-0.06
Uncoated steel	0.015-0.06
Coated steel	0.03-0.15
Concrete	0.06-1.5
Plastic, PVC, PE	0.02-0.05
Glass fibre	0.06
Brass, Copper	0.003

The Hazen–Williams Equation is an empirical equation widely used in practice. It is especially applicable for smooth pipes of medium and large diameters and pipes that are not attacked by corrosion. The values of the Hazen–Williams constant, C_{hw}, for selected pipe materials and diameters are shown in Table 9.9. Bhave states that the values are experimentally determined for flow velocity of 0.9 m/s. A correction for the percentage given is therefore suggested in case the actual velocity differs significantly. Table 9.10 for example, the value of $C_{hw} = 120$ increases twice for 3% if the expected velocity is around a quarter of the reference value i.e. $C_{hw} = 127$ for v of, say, 0.22 m/s. On the other hand, for doubled velocity $v = 1.8$ m/s, $C_{hw} = 116$ i.e. 3% less than the original value of 120. However, such corrections do not significantly influence the friction loss calculation, and are, except for extreme cases, rarely applied in practice. Bhave also states that the Hazen–Williams Equation becomes less accurate for C_{hw} values below 100.

9.20 Transmission Line Design

This article includes procedures specifically related to transmission lines of 12 inches diameter and larger. Service connections are usually not permitted. Interconnections with the distribution system piping should be held to a minimum and are usually over 1,000 feet apart.

9.20.1 Design procedures

Transmission line design shall include the following procedures.

- **Layout.** The new line will be designed to fall within existing utility or street right-of-way where available. The price of acquiring easements through private property must be considered as part of the alternative cost analysis. Easements must be wide enough to allow for initial construction and future maintenance. Installation close to physical features, such as buildings or other utilities, which would cause construction problems or future access problems for maintenance, should be avoided. A set of plan and profile drawings shall be prepared which shall show as a minimum the following information:
  - Survey base line with physical control points.
  - Easements, rights-of-way, streets, and construction limits, etc.
  - Existing physical features such as buildings, fences, structures, utilities, trees and drainage systems.
  - Existing, and proposed if applicable, ground elevations along the syphon net of the pipe shall be shown on the profile.
  - In plan, the proposed pipeline and its relationship to the survey base line.
  - In profile, the syphon net elevation of the proposed pipeline.
  - Beginning and ending points of the pipeline and all appurtenances.
  - Construction details of the pipeline, connections, appurtenances, bedding and surface restoration, etc. Typical information shown on plan and profile drawings is illustrated in Figure 9.29.
- **Diameter vs pumping costs.** Pump costs are dependent upon initial cost and horsepower requirements. The total dynamic head of the system is the sum of the suction lift, discharge head, friction head and velocity head, and is represented by the following equation:

$$TDH = H_S + H_D + H_F + \frac{V^2}{2g} \qquad 9.38$$

Where,

TDH = total dynamic head

H_S = suction lift

H_D = discharge head

H_F = friction head

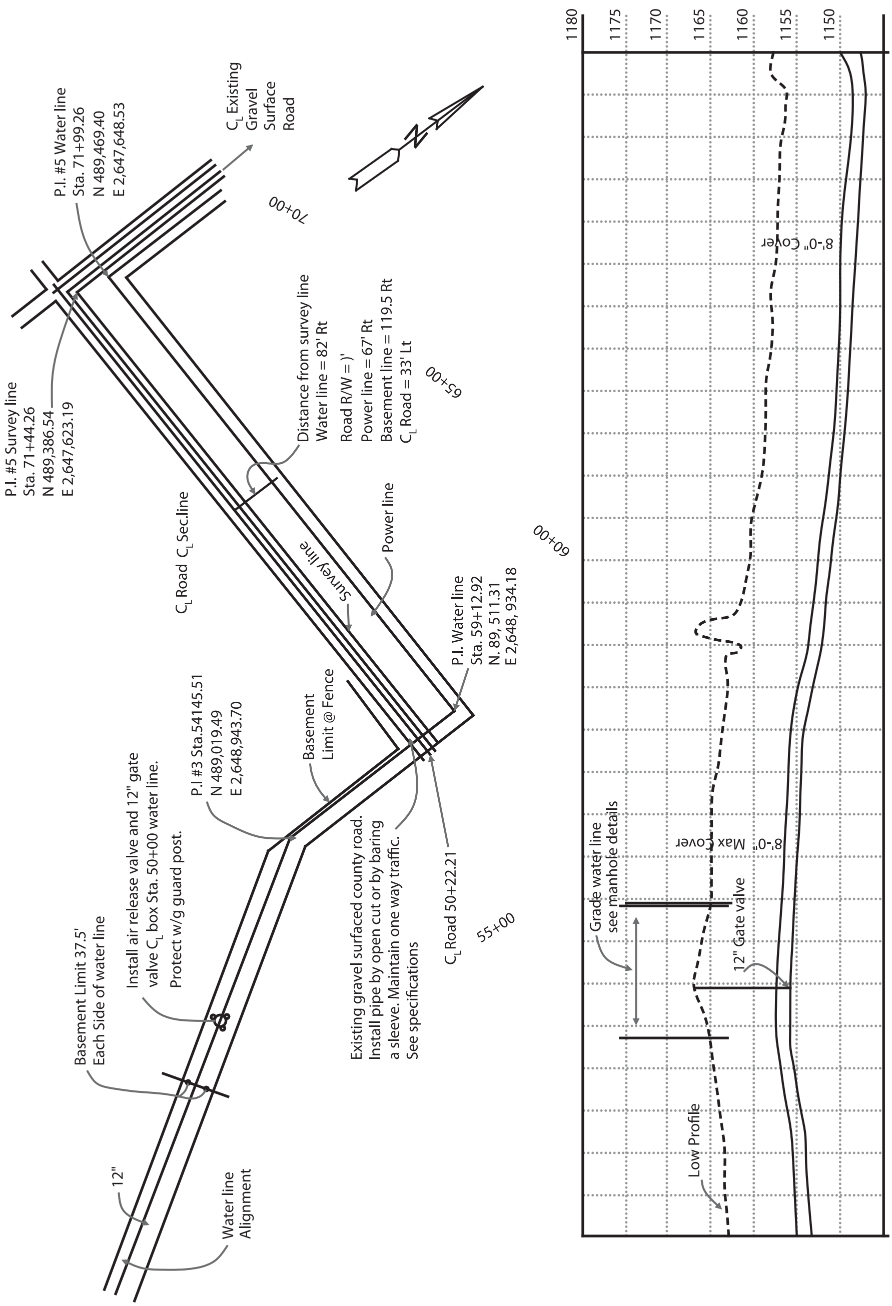


Figure 9.29: Typical plan and profile drawings.

$$\frac{V^2}{2g} = \text{velocity head}$$

The most economical pipe diameter in a pumped system is determined by comparing pumping costs for various sizes of pipe. Only standard pipeline sizes should be considered. In order to hold friction losses to a minimum and to reduce the possibility of severe waterhammer, the diameter should be sized so that the velocity is 4 feet per second (fps) or less. Under special circumstances when approved by the appropriate authority, the maximum design velocity may exceed 4 fps.

- **Hydraulic calculations.** Hydraulic calculation is necessary to design an engineering transmission pipeline. The details of hydraulic analysis are described in the Article 9.19. The Hazen-Williams formula is often used to compute flow characteristics. Also, the Darcy-Weisbach equation is used in some computer programs. Depending on requirements, one of the following forms of the HazenWilliams equation is used:

$$V = 0.55 C D^{0.63} S^{0.54} \quad 9.39$$

$$Q = 0.443 C D^{2.63} S^{0.54} \quad 9.40$$

$$S = \frac{2.32Q}{CD^{2.63}} 1.85 \quad 9.41$$

$$V = 1.318 C R^{0.63} S^{0.54} \quad 9.42$$

Where,

V = velocity of flow in feet per second
C = coefficient of roughness
D = diameter of pipe in feet
S = head loss in feet per foot of length
Q = flow in cubic feet per second
R = hydraulic radius in feet

Values of C are 150 for thermoplastic pipe, 140 for smooth lined steel pipe, very smooth concrete pipe, cement lined ductile iron pipe and asbestos-cement pipe; 130 for ordinary ductile iron pipe in good condition; 110 to 120 for ductile or cast-iron pipe in service 5 to 10 years; 100 for older cast-iron pipe; and 40 to 80 for old cast-iron or steel pipe which is severely tuberculated or any pipe with heavy deposits. A quick solution for the equations may be found by use of the nomographs in Figure 9.30 (a and b). They are prepared from the Hazen-Williams formula by using $C = 150$ and $C = 100$. For any other smaller values of C, the discharge or velocity obtained from the specific (either C= 150 of C = 100)nomograph is multiplied by the ratio of the given value of *C* 150 or 100. If the discharge or velocity is given, it should be multiplied by the ratio of 100 (using C = 100) to the known value of *C* before the nomograph is used.

The nomograph is used by lining up values on the scales by means of a ruler or straight edge. Two independent variables must be set to obtain the other values. For example line (1) indicates that 500 gallons per minute may be obtained with a 6 inch inside diameter pipe at a head loss of about 0.65 pounds per square inch at a velocity of 6.0 feet per second. Line (2) indicates that a pipe with a 2.1 inch inside diameter will give a flow of about 60 gallons per minute at a loss in head of 2 pounds per square inch per 100 feet of pipe. Line (3) and dotted line (3) show that in going from a pipe 2.1 inch inside diameter to one of 2 inches inside diameter the head loss goes from 3 to 4 pounds per square inch in obtaining a flow of 70 gallons per minute.

Example 9.11: By using Figure 9.30, determine the discharge, in cubic feet per second (cps), from a 12-inch pipe for which *C* = 120 when the loss of head is 5 feet per 1000 feet.

Solution:
The discharge corresponding to the given diameter and loss of head for a value of *C* = 100 is found first. A straightedge passing through 12 on the diameter line and 5 on the loss-of- head line will intersect the discharge line at 2.5 cfs. Therefore, the discharge is:

$$2.5 \times \frac{120}{100} = 3.0cfs$$

Example 9.12: A 30-inch pipe for which *C*=130 is to discharge 10,000 gallons per minute (gpm). By using Figure 9.30, find the loss of head per 1000 feet of pipe.

Solution:
The first step is to determine the discharge that corresponds to a value of *C* = 100. This is 10,000 × 1001130 = 7,692 gpm. A straightedge through 30 on the diameter line and 7,692 on the discharge line intersects the loss-of-head line at 2.0 feet per 1000 feet.

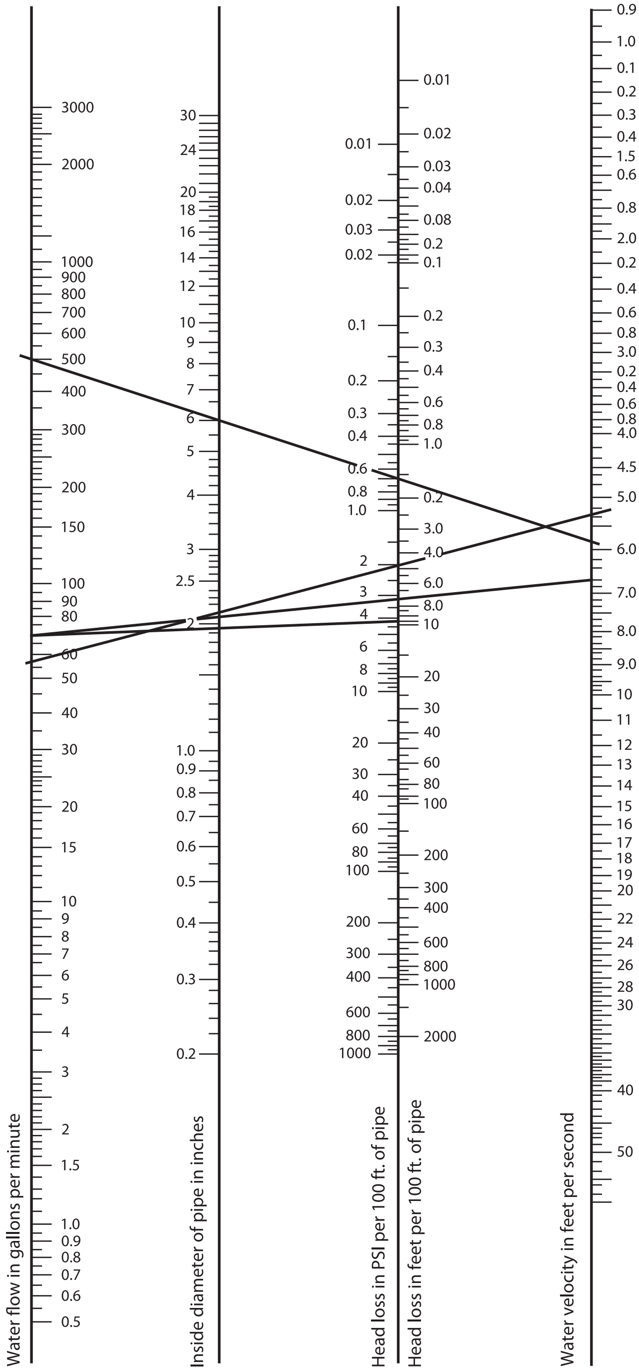


Figure 9.30 (a): **Nomograph for Hazen-Williams formula in which C=150**

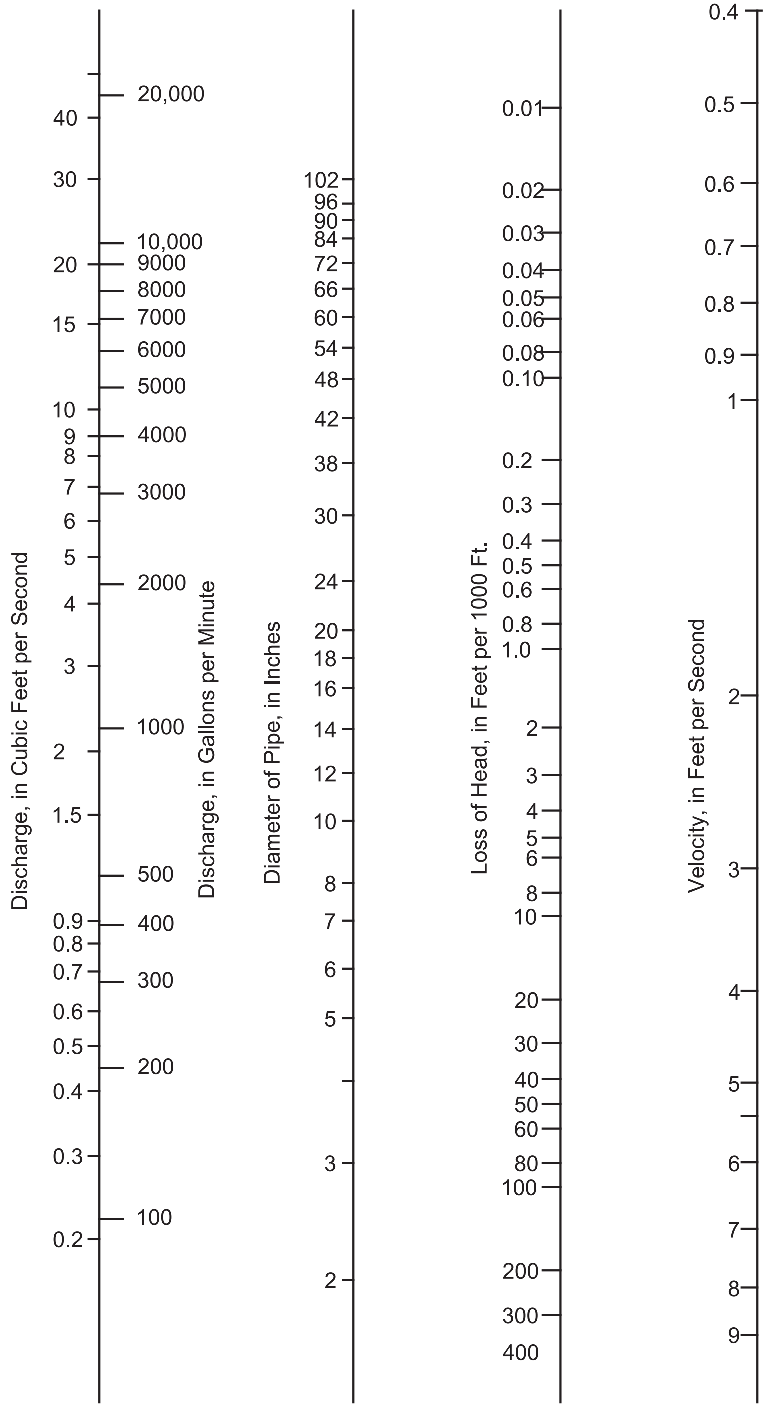


Figure 9.30 (b): **Nomograph for Hazen-Williams formula in which C = 100**

Valves, bends, and other fittings in a pipeline and sudden enlargements or contractions cause loss of head. If a valve is partly closed, there is greater resistance to the flow and greater loss of head. Table 9.11 shows the loss in pipe fittings and appurtenances, expressed as equivalent lengths of straight pipe as a multiple of the diameter, due to various valves, fittings, contractions, and enlargements.

Example 9.13: Assuming steady state conditions the pump in Figure 9.31 delivers 5,000 gpm at a discharge pressure of 140 feet of head. The pipeline is 24-inch diameter, $C = 100$, and is 3,625 feet from pump to reservoir. Calculate the internal pressure at Station 20 + 00.

Solution:
From Figure 9.30, the head loss is found to be 2.7 feet per 1,000 feet. Assume other losses are negligible.

$$
\begin{array}{ll}
H_L = 2 \times 2.7 & = 5.4 \text{ ft.} \\
\text{Elevation at pump} & = 230.0 \\
\text{Discharge pressure} & = 140.0 \text{ ft of head } (60.0 \text{ psi}) \\
\text{Elevation of HGL} & \\
\text{at pump} & = 370.0 \\
\text{Less } H_L & = -5.4 \\
\text{Elevation of HGL at} & \\
\text{Station 20} & = 365 \\
\text{Pipe Elevation} & = -310 \\
\text{Pressure} & = 55.6 \text{ft. } (24.1 \text{ psi})
\end{array}
$$

- **Internal pressure:** The internal pressure is the difference in elevation between the conduit and the hydraulic grade line (HGL). The pressure at the beginning of a main may be generated by a pump, reservoir elevation or connection from another pipeline. Pressure losses are due to friction, bends and fittings, and changes in elevation. These are measured above a horizontal datum plane as shown on Figure 9.31.
- **Pipe materials:** The materials allowed for use are steel, ductile iron, reinforced concrete, asbestos- cement, glass fibre reinforced and polyvinyl chloride pipe. Mains must be designed for the maximum internal working pressure plus an allowance for water hammer. Design must include external stress due to earthfill and superimposed loads. Reductions in wall thickness at isolated regions of lower pressure or reduced external stress

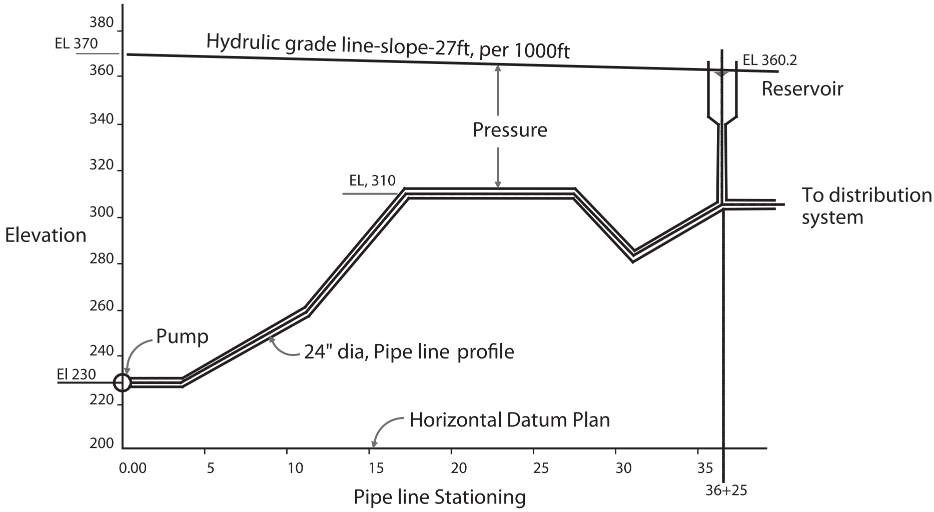


Figure 9.31: **Illustration of pipe line pressure.**

will not be made due to the possibility of construction personnel installing these sections in the wrong sections of the line. Pipe shall be designed and specified in accordance with applicable AWWA Standards, and specifically as indicated.

- **Anchorage and Expansion**
  - Thrust restraint at bends and abrupt changes in direction is required. Certain types of pipe such as welded steel are designed as continuous conduits and some ductile iron joint systems are designed as restrained joints which may not require thrust blocking.
  - Under empty conditions, lightweight conduit such as steel is buoyant and will float. This may occur if the water table is high enough even though the pipeline may be backfilled. If this is likely, the designer must add extra weight to the pipe. This may be accomplished by the use of reinforced concrete collars, poured in place, and firmly anchored to the pipe.

    For contractions and enlargements, d is diameter of smaller pipe and D is diameter of larger pipe; resistance is expressed in terms of d.
  - All pipe materials are subjected to expansion and contraction forces due to temperature changes. In colder climates, water temperature may vary from 32 degrees F. To over 80 degrees F, a range of over 50 degrees. Above ground installations, especially in hot temperature

climates may also cause significant temperature variations. In bell and spigot joints, this effect usually may be neglected. However, the effects of expansion and contraction can be significant for long, straight, continuous pipelines. The designer shall include expansion joints designed for the pipe material and/or include appropriate stresses in the calculation for pipe wall thickness where required. However, it is not usually economical to increase the strength or thickness of the pipe wall as the sole means of resisting these stresses.

- **Valving:** Sectionalizing valves are to be provided at all connections to the main. This includes pump discharge, distribution connections, fire hydrants, blowoffs, air valves and reservoir connect ions. Line valves are not usually required to be closer than one mile unless intermediate distribution connections are made. For larger size mains, the use of valves one standard size smaller than the pipeline is allowed as a cost-saving measure, provided that the velocity through the valve does not exceed 11 fps. Many large line valves have an integral by-pass arrangement. Valves may be the same as used in the distribution system. Valves shall be adequately designed for the actual internal pressure.

- **Air-vacuum valves:** Air-release valves eliminate air pocket build up which causes a flow constriction and increased head loss. They are designed to expel air from a line during filling and close automatically when water reaches the valve. Vacuum valves are designed to allow air to enter the main when it is being drained. Also, vacuum valves are required to prevent the possible collapse of thin wall conduit which may be subject to a vacuum under certain conditions such as a break in the pipeline. Combination air release and vacuum valves are to be installed at the following locations:
  - Peaks, where the pipe slope changes from positive to negative.
  - Long relatively straight stretches at ¼ to ½ mile intervals. Air valves are to be sized to exhaust air at the pipe fill rate. Vacuum valves are to be designed to admit air at a rate equal to the flow generated by gravity.

  Consult manufacturer's literature for capacity and performance data. These valves are to be installed in pairs to prevent problems due to failure at one of the valves.

- **Blow-offs:** Blow-offs, with a drain to a disposal area, should be installed near low points and other suitable locations to facilitate draining the conduit and disposal of the water. Blow-offs will be designed with an air-gap to prevent contaminated water from backing up into the main.

- **Hydrants:** Hydrants for fire fighting purposes are not normally installed on transmission mains. If they are, design should be as specified. Hydrants may be installed to facilitate filling and disinfection. For this purpose, a hydrant may be located adjacent to each line valve.
- **Access syphon:** Manholes for access to the inside of large mains facilitate the construction and inspection of pipelines large enough to be entered by a workman. The minimum pipeline size is usually 20-inch diameter. They are useful if located adjacent to air valves, blow-offs, and Line valves. Access manholes are to be designed as pipeline tees and fitted with a bolted blind flange.
- **Flow measurements:** The design should allow for measurement of the volume of flow in the main. This may be done by pitot tube which requires the installation of a 1 inch corporation in the top of the main, or by a venture installed as part of the pipeline or other commercially available equipment or methods. Sufficient straight pipe without flow interruption shall be provided ahead of and following the point of measurement as required by the manufacturer of the device.
- **External corrosion:** The design, if the same as for distribution mains. Also use references cited.
- **Area restoration:** The trench will be backfilled and compacted to prevent settlement. The surface will be brought to grade to match existing or design elevations. In previously grassy areas, the surface will be seeded or sodded. In paved or sidewalk areas, restoration shall match the original surface treatment as close as possible.

9.20.2 Filling procedures

Water should be admitted to the new transmission main at the lowest available point and be allowed to fill the pipe slowly up to higher elevations. Each section of main between line valves shall be filled separately and checked before proceeding to the next section. The progress of water in the pipeline shall be carefully and continuously monitored. It is usually neither feasible nor necessary to begin the filling process prior to the completion of construction of the entire pipeline.

- On mains longer than a few thousand feet, where it would be unwieldy to continuously refer to construction drawings, a special profile drawing may be prepared at a smaller scale; e.g., 1″= 100'or 1″ = 200'. This profile drawing should show pipeline stationing, all appurtenances and other major physical and design features.

- Prior to commencement of filling operations, all blow-offs, access manholes, other appurtenances, and temporary construction features should be checked to make sure they are closed and sealed. Check to see that air-release valves are free of debris and the control valves are open. Fire hydrants may be opened for additional air release and flushing purposes.
- The rate of fill shall be carefully monitored and controlled. Use of two-way radios is desirable on longer pipelines. The point of fill should be continuously manned and personnel should be prepared to close valves in the event of leaks or other problems.
- Water may be admitted through line valves on smaller lines. On larger mains, hydrants, distribution connections, or bypass connections of 6-inch or 8- inch diameter should be used. In any case, the valves being used must be capable of being closed under the conditions of flow with full head on one side only.
- Progress shall be monitored by checking air- release valves and flow from hydrants. Hydrants may be closed when full-barrel flow is achieved without pulsing or surging. Each appurtenance must be checked for leaks during the filling process.
- The filling process does not have to be a 24-hour-around-the-clock operation. It can be stopped any time, and later resumed. However, it cannot be carried on without being continuously monitored.
- Upon completion of the filling process, the connection used shall be closed and each appurtenance shall be checked for leaks. Each section of main between line valves shall be individually filled, checked and disinfected before proceeding to the next sect ion, and before putting it into service. Test pressures shall be in accordance with applicable AWWA Standards for the type of pipe used and the design pressure.

9.20.3 Syphons

Under special circumstances, a pipeline may be designed as an syphon. In these cases, a section of the main will be above the hydraulic grade line and therefore is under negative pressure. This design condition requires the specific approval of the installation's Facilities Engineer. Certain design details require special attention.

- **Pipe material and thickness:** The pipe material and wall thickness shall be specifically designed for negative pressure. This requires a rigid pipe wall. Thin wall pipe, such as steel, may collapse under these conditions if not specially designed for negative pressure.
- **Air-vacuum valves:** Air release and vacuum valves would defeat the design of a pipeline syphon. Their use must be carefully considered and should be limited to reaches of positive pressure if at all.

9.21 The Manning Equation

The Manning Equation is another empirical equation used for the calculation of friction losses. In a slightly modified format, it also occurs in some literature under the name of Strickler. The usual range of the *N*-values ($m^{-1/3}s$) for typical pipe materials is given in Table 9.12.

The Manning Equation is more suitable for rough pipes where *N* is greater than 0.015 $m^{-1/3}s$. It is frequently used for open channel flows rather than pressurised flows.

9.22 Comparison of the Friction Loss Equations

The straightforward calculation of pipe resistance, being the main advantage of the Hazen–Williams and Manning equations, has lost its relevance as a result of developments in computer technology. The research also shows some limitations in the application of these equations compared to the Darcy–Weisbach Equation. Nevertheless, this is not necessarily a problem for engineering practice and the Hazen–Williams Equation in particular is still widely used in some parts of the world. Figures 9.32 and 9.33 show the friction loss diagrams for a range of diameters and two roughness values calculated by each of the three equations. The flow in two pipes of different length, L = 200 and 2000 m respectively, is determined for velocity v = 1 m/s. Thus in all cases, for D in m and Q in m^3/s:

$$Q = v\frac{D^2\pi}{4} = 0.7854D^2$$

The example shows little difference between the results obtained by three different equations. Nevertheless, the same roughness parameters have a different impact on the friction loss in the case of larger and longer pipes.

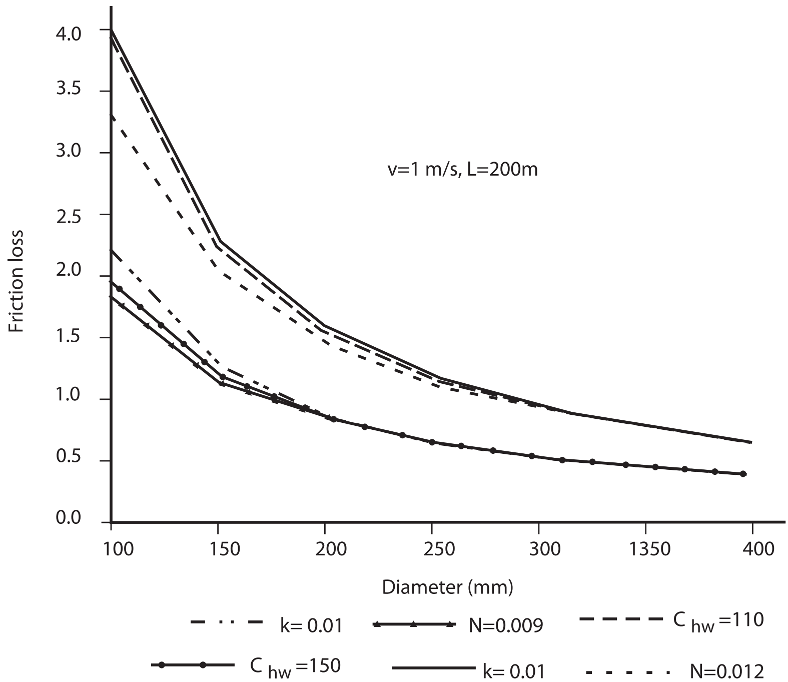


Figure 9.32: **Comparison of the friction loss equations: mid range diameters, *v* = 1 m/s, *L* = 200 m.**

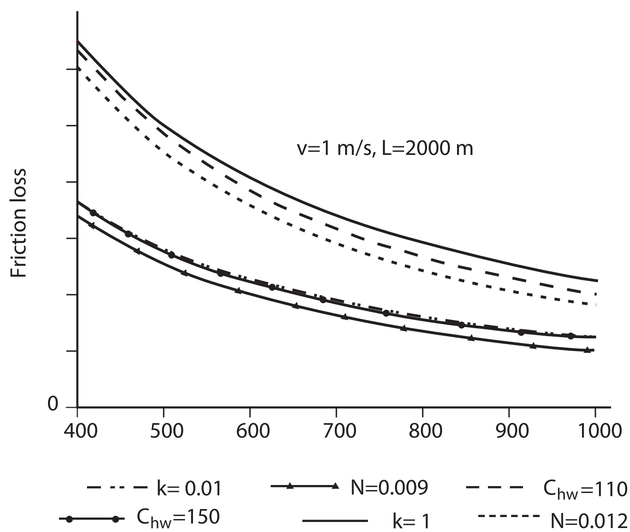


Figure 9.33: **Comparison of the friction loss equations: large diameters, *v* = 1 m/s, *L* = 2000 m.**

The difference in results becomes larger if the roughness values are not properly chosen. Figure 9.34 shows the friction loss calculated using the roughness values suggested for PVC in Tables 9.9, 9.10 and 9.12. Hence, the choice of a proper roughness value is more relevant than the choice of the friction loss equation itself. Which of the values fits the best to the particular case can be confirmed only by field measurements. In general, the friction loss will rise when there is:

- an increase in pipe discharge,
- an increase in pipe roughness,
- an increase in pipe length,
- a reduction in pipe diameter,
- a decrease in water temperature.

In reality, the situations causing this to happen are:

- higher consumption or leakage,
- corrosion growth,
- network expansion.

Table 9.9: **The Hazen–Williams factors.**

Pipe material/ Pipe diameter	C_{hw} 75 mm	C_{hw} 150 mm	C_{hw} 300 mm	C_{hw} 600 mm	C_{hw} 1200 mm
Uncoated cast iron	121	125	130	132	134
Coated cast iron	129	133	138	140	141
Uncoated steel	142	145	147	150	150
Coated steel	137	142	145	148	148
Galvanised iron	129	133	--	--	--
Uncoated asbestos cement	142	145	147	150	--
Coated asbestos cement Concrete, minimum/	147	149	150	152	--
maximum values Pre-stressed concrete	69/129	79/133	84/138	90/140	95/141
PVC, Brass, Copper,	--	--	147	150	150
Lead Wavy PVC	147	149	150	152	153
Bitumin/Cement lined	142	145	147	150	150
	147	149	150	152	153

Table 9.10: Correction of the Hazen–Williams factors.

C_{hw}	$V < 0.9$ m/s per halving	$V > 0.9$ m/s per doubling
Less than 100	+ 5%	- 5%
100-130	+3%	-3%
130-140	+1%	-1%
Greater than 140	-1%	+1%

Table 9.11: Losses in pipe fittings and appurtenances

Description of Pipe Equivalent Fitting or Appurtenances	Loss in Equivalent Length of Pipe Diameters
Gate Valve	
¾ Closed	900
½ Closed	160
¼ Closed	35
Full Open	13
Angle Valve Open	170
Globe Valve Open	340
Swing Check Valve	80
Elbows	
90° Standard	
90° Long Radius	20
45° Standard	16
Tee Flow Through Run	20
Standard Tee Take-Off	75
Run of Tee Reduce One-Half	32
Sudden Contraction:	
d/D=0.25	15
d/D=0.5	12
d/D=0.75	7
Sudden Enlargement:	
d/D=0.25	32
d/D=0.5	20
d/D =0.75	19
Entrance to Basin	75

Table 9.12. **The Manning factors.**

Pipe material	N ($m^{-1/3}s$)
PVC, Brass, Lead, Copper, Glass fibre	0.008-0.011
Pre-stressed concrete	0.09-0.012
Concrete	0.010-0.017
Welded steel	0.012-0.013
Coated cast iron	0.012-0.014
Uncoated cast iron	0.013-0.015
Galvanised iron	0.015-0.017

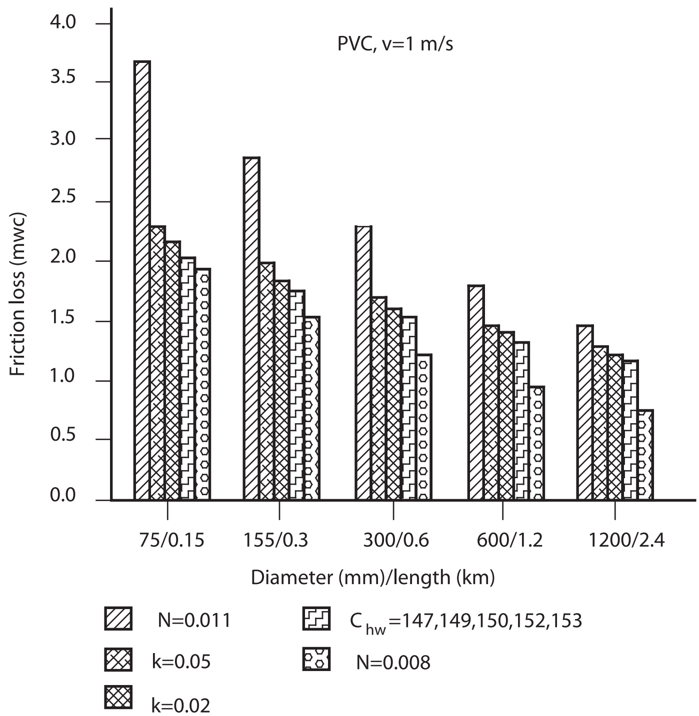


Figure 9.34: **Comparison of the friction loss equations for various PVC roughness factors.**

Table 9.13: **Hydraulic gradient in pipe *D* = 300 mm, *Q* = 80 l/s, *T* = 10°C.**

Parameter	k = 0.01 mm	k = 0.1 mm	k = 1 mm	k = 5 mm
S(m/km)	3.3	3.8	6.0	9.9
Increase (%)	--	15	58	65

The friction loss equations clearly point to the pipe diameter as the most sensitive parameter. The Darcy–Weisbach Equation shows that each halving of D (e.g. from 200 to 100 mm) increases the head-loss 25 to 32 times. Moreover, the discharge variation will have a quadratic impact on the head-losses, while these grow linearly with the increase of the pipe length. The friction losses are less sensitive to the change of the roughness factor, particularly in smooth pipes (an example is shown in Table 9.13). Finally, the impact of water temperature variation on the head-losses is marginal.

Example 9.14: For pipe L = 450m, D = 300 mm and flow rate of 120 l/s, calculate the friction loss by comparing the Darcy–Weisbach- (k = 0.2 mm), Hazen–Williams- (C_{hw} = 125) and Manning equations (N = 0.01). The water temperature can be assumed at 10°C. If the demand grows at the exponential rate of 1.8% annually, what will be the friction loss in the same pipe after 15 years? The assumed value of an increased absolute roughness in this period equals k = 0.5 mm.

Solution:

For a flow Q = 120 L/s and a diameter of 300 mm, the velocity in the pipe:

$$v = \frac{4Q}{D^2\pi} = \frac{4 \times 0.12}{0.3^2 \times 3.14} = 1.70 m/s$$

Based on the water temperature, the kinematic viscosity can be calculated from Equation

$$= \frac{497 \times 10^{-6}}{(T + 42.5)^{1.5}} = \frac{497 \times 10^{-6}}{(10 + 42.5)^{1.5}} = 1.31 \times 10^{-6} m^2/s$$

The Reynolds number then becomes:

$$Re = \frac{vD}{\mu} = \frac{1.70 \times 0.3}{1.31 \times 10^{-6}} = 3.9 \times 10^5$$

For the value of relative roughness k/D = 0.2/300 = 0.00067 and the calculated Reynolds number, the friction factor λ can be determined from the Moody diagram in ($\lambda \approx 0.019$). Based on the value of the Reynolds number (>> 4000), the flow regime is obviously turbulent. The same result can also be obtained by applying the Barr approximation. From Equation 9.32:

$$\lambda = 0.25/\log^2\left[\frac{5.1286}{Re^{0.89}} + \frac{k}{3.7D}\right]$$

$$\lambda = 0.25/\log^2 \left[\frac{5.1286}{\left(3.9 \times 10^5\right)^{0.89}} + \frac{0.2}{3.7 \times 300} \right]$$

Finally the friction loss from the Darcy-Weisbach equation can be determined:

$$h_f = \frac{\lambda L}{12.1 D^5} Q^2 = \frac{0.019 \times 450}{12.1 \times 0.3^5} 0.12^2 = 4.18 mwc$$

Applying the Hazen-Williams Equation with $C_{hw} = 125$, the friction loss becomes:

$$h_f = \frac{10.68 L}{C_{hw}^{1.825} D^{4.87}} Q^{1.825} = \frac{10.68 \times 450}{125^{1.825} \times 0.3^{4.87}} 0.12^{1.852} = 4.37 mwc$$

Introducing a correction for the C_{hw} value of 3%, as suggested in Table 9.11 based on the velocity of 1.7 m/s (almost twice the value of 0.9 m/s), yields a value of C_{hw}, which is reduced to 121. Using the same formula, the friction loss then becomes h_f =4.64 mwc, which is 6% higher than the initial figure. Finally, applying the Manning Equation with the friction factor $N = 0.01$:

$$h_f = \frac{10.29 N^2 L}{D^{16/3}} Q^{1.852} = \frac{10.29 \times 0.01^2 \times 450}{0.3^{16/3}} 0.12^2 = 4.37 mwc$$

With the annual growth rate of 1.8%, the demand after 15 years becomes:

$$Q_{15} = 120 \left(1 + \frac{1.8}{100}\right)^{15} = 156.82 l/s$$

which, with the increase of the k-value to 0.5 mm, yields the friction loss of 8.60 mwc by applying the Darcy–Weisbach Equation in the same way as shown above. The interim calculations give the following values of the parameters involved: $v = 2.22$ m/s, $Re = 5.1 \times 10^5$ and $\lambda = 0.023$. The final result represents an increase of more than 100% compared to the original value of the friction loss (at the demand increase of approximately 30%).

9.23 Minor Losses

Minor (in various literature local or turbulence) losses are usually caused by installed valves, bends, elbows, reducers, etc. Although the effect of the disturbance is spread over a short distance, the minor losses are for the sake of simplicity attributed to a cross-section of the pipe. As a result, an instant drop in the hydraulic grade line will be registered at the place of obstruction (see Figure 9.35). Factors R_m and n_m from Equation 9.43 are uniformly expressed as:

$$R_m = \frac{8\xi}{\pi^2 g D^4} = \frac{\xi}{12.1 D^4}, n_m = 2 \qquad 9.43$$

Where ξ represents the minor (local) loss coefficient. This factor is usually determined by experiments.

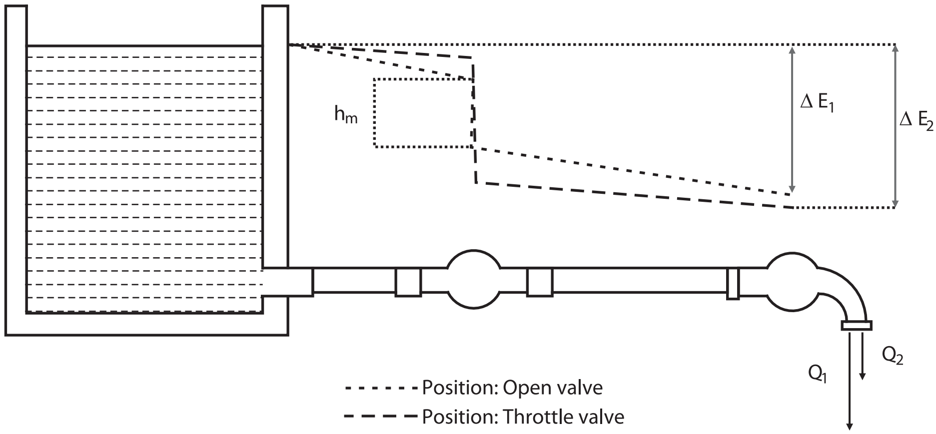


Figure 9.35: **Minor loss caused by valve operation.**

The minor loss factors for various types of valves are normally supplied together with the device. The corresponding equation may vary slightly from 3.25, mostly in order to enable a diagram that is convenient for easy reading of the values. In the example shown in Figure 9.36, the minor loss of a butterfly valve is calculated in mwc as: $h_m = 10Q^2/K_v^2$, for Q in m^3/h. The K_v-values can be determined from the diagram for different valve diameters and settings. Substantial minor losses are measured in the following cases:

- the flow velocity is high, and/or
- there is a significant valve throttling in the system.

Such conditions commonly occur in pumping stations and in pipes of larger capacities where installed valves are regularly operated; given the magnitude of the head-loss, the term 'minor' loss may not be appropriate in those situations. Within the distribution network on a large scale, the minor losses are comparatively smaller than the friction losses. Their impact on overall head-loss is typically represented through adjustment of the roughness values (increased k and N or reduced C_{hw}). In such cases, $\Delta H \approx h_f$ is an acceptable approximation and the hydraulic gradient then becomes:

$$S = \frac{\Delta H}{L} \approx \frac{h_f}{L} \qquad 9.44$$

The other possibility of considering the minor losses is to introduce so-called equivalent pipe lengths. This approach is sometimes used for the design of indoor installations where the minor loss impact is simulated by assuming an increased pipe length (for example, up to 30–40%) from the most critical end point.

9.24 Single Pipe Calculation

Summarised from the previous paragraph, the basic parameters involved in the head-loss calculation of a single pipe using the Darcy–Weisbach Equation are:

- length L,
- diameters D,
- absolute roughness k,
- discharge Q,
- piezometric head difference ΔH (i.e. the head-loss),
- water temperature T.

The parameters derived from the above are:

- velocity, $v = f(Q, D)$,
- hydraulic gradient, $S = f(\Delta H, L)$,
- kinematic viscosity, $v = f(T)$,
- Reynolds number, $Re = f(v, D, u)$,
- friction factor, $\lambda = f(k, D, Re)$.

In practice, three of the six basic parameters are always included as an input:

- L, influenced by the consumers' location,
- k, influenced by the pipe material and its overall condition,
  T, influenced by the ambient temperature.

The other three, D, Q and ΔH, are parameters of major impact on pressures and flows in the system. Any of these parameters can be considered as the overall output of the calculation after setting the other two in addition to the three initial input parameters. The result obtained in such a way answers one of the three typical questions that appear in practice:

- What is the available head-loss ΔH (and consequently the pressure) in a pipe of diameter D, when it conveys flow Q?
- What is the flow Q that a pipe of diameter D can deliver if certain maximum head-loss ΔH_{max} (i.e. the minimum pressure p_{min}) is to be maintained?
- What is the optimal diameter D of a pipe that has to deliver the required flow Q at a certain maximum head-loss ΔH_{max} (i.e. minimum pressure P_{min})?

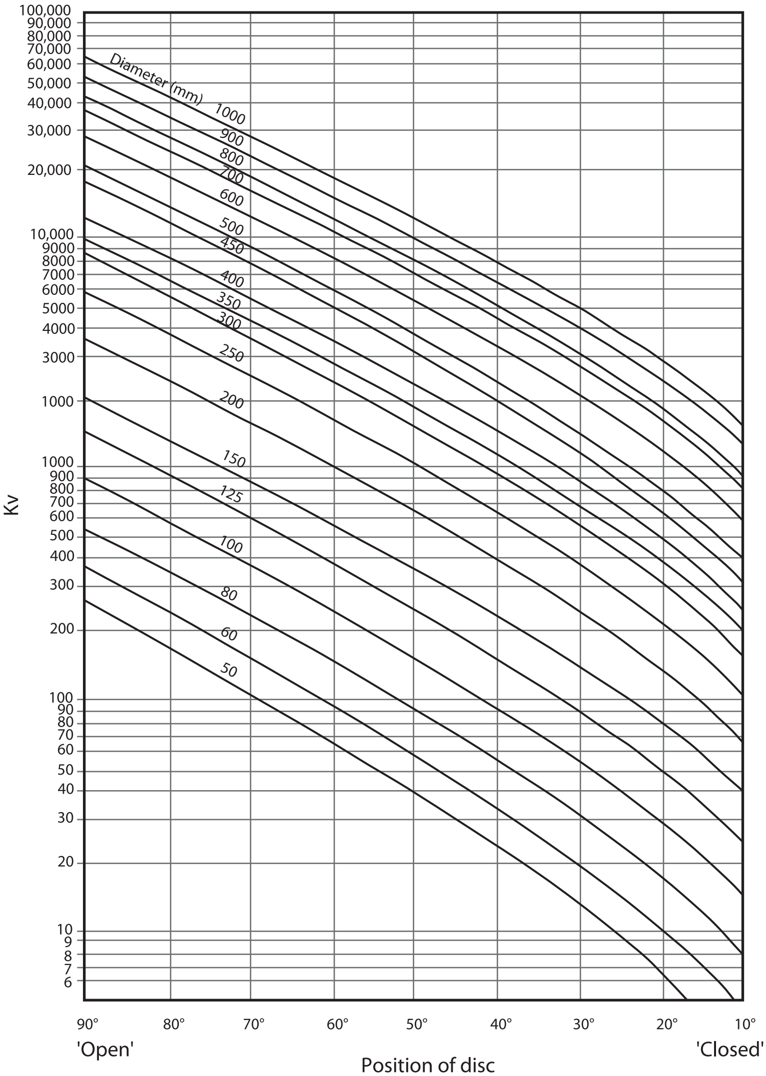


Figure 9.36: Example of minor loss diagram from valve operation.

The calculation procedure in each of these cases is explained below. The form of the Darcy–Weisbach Equation linked to kinetic energy is more suitable in this case:

$$\Delta H \approx h_f = \frac{\lambda L}{12.1D^5}Q^2 = \lambda\frac{Lv^2}{D^2 g}; S = \frac{\lambda v^2}{D^2 g} \qquad 9.45$$

9.24.1 Pipe pressure

The input data in this type of the example are: *L, D, k, Q* or *v*, and *T*, which yield ΔH (or S) as the result. The following procedure is to be applied:

- For given *Q* and *D*, find out the velocity, $v = 4Q/(D^2\pi)$.
- Calculate *Re* from Equation 9.34.
- Based on the *Re* value, choose the appropriate friction loss equation, 9.45 and determine the λ-factor. Alternatively, use the Moody diagram for an appropriate *k/D* ratio.
- Determine ΔH (or S) by Equation 9.45.

The sample calculation has already been demonstrated in Example 9.14.

To be able to define the pressure head, $p/\rho g$, an additional input is necessary:

- the pipe elevation heads, *Z*, and
- known (fixed) piezometric head, *H*, at one side.

There are two possible final outputs for the calculation:

- If the downstream (discharge) piezometric head is specified, suggesting the minimum pressure to be maintained, the final result will show the required head/pressure at the upstream side i.e. at the supply point.
- If the upstream (supply) piezometric head is specified, the final result will show the available head/pressure at the downstream side i.e. at the discharge point.

Example 9.15: The distribution area is supplied through a transportation pipe L = 750 m, D = 400 mm and k = 0.3 mm, with the average flow rate of 1260 m^3/h. For this flow, the water pressure at the end of the pipe has to be maintained at a minimum 30 mwc. What will be the required piezometric level and also the pressure on the upstream side in this situation? The average pipe elevation varies from Z_2 = 51 msl at the downstream side to Z_1 = 75 msl at the upstream side. It can be assumed that the water temperature is 10°C.

For flow Q= 1260 m^3/h

$$v = \frac{4Q}{D^2\pi} = \frac{4\times 0.35}{0.4^2\times 3.14} = 2.79 m/s$$

For temperature T= 10°C, $\mu = 1.31\times 10^{-6} m^2/s$, the Reynolds number takes the value of

$$Re = \frac{vD}{\mu} = \frac{2.79\times 0.4}{1.31\times 10^{-6}} = 8.5\times 10^5$$

And the function factor is λ from Barr's Equation equals:

$$\lambda = 0.25/\log^2\left[\frac{5.1286}{Re^{0.89}} + \frac{k}{3.7D}\right]$$

$$\lambda = 0.25/\log^2\left[\frac{5.1286}{\left(8.5\times 10^5\right)^{0.89}} + \frac{k}{3.7\times 400}\right]$$

The friction loss from the Darcy-Weisbach equation can be determined:

$$h_f = \frac{\lambda L}{12.1 D^5} Q^2 = \frac{0.019\times 750}{12.1\times 0.4^5} 0.35^2 \approx 14\, mwc$$

The downstream pipe elevation is given at Z_2 = 51 msl. By adding the minimum required pressure of 30 mwc to it, the downstream piezometric head becomes H_2 = 51 + 30 = 81 msl. On the upstream side, the piezometric head must be higher for the value of calculated friction loss, which produces a head of H_1 = 81 + 14 = 95 msl. Finally, the pressure on the upstream side will be obtained by deducting the upstream pipe elevation from this head. Hence $p1/\rho g$ = 95 - 75 = 20 mwc. Due to configuration of the terrain in this example, the upstream pressure is lower than the downstream one. For the calculated friction loss, the hydraulic gradient $S = h_f/L = 14/750 \approx 0.019$.

9.24.2 Maximum pipe capacity

For determination of the maximum pipe capacity, the input data are: *L, D, k, ΔH* (or *S*), and *T*. The result is flow *Q*. Due to the fact that the λ-factor depends on the Reynolds number i.e. the flow velocity that is not known in advance, an iterative procedure is required here. The following steps have to be executed:

- Assume the initial velocity (usually, v = 1 m/s).
- Calculate *Re* from Equation 9.34.

- Based on the *Re* value, choose the appropriate friction loss equation, 9.33 or 9.36, and calculate the λ-factor. For selected *Re* and k/D values, the Moody diagram can also be used as an alternative.
- Calculate the velocity after re-writing Equation 9.45:

$$v = \sqrt{\frac{2gDS}{\lambda}} \qquad 9.46$$

If the values of the assumed and determined velocity differ substantially, steps 2–4 should be repeated by taking the calculated velocity as the new input. When a sufficient accuracy has been reached, usually after 2–3 iterations for flows in the transitional turbulence zone, the procedure is completed and the flow can be calculated from the final velocity. If the flow is in the rough turbulence zone, the velocity obtained in the first iteration will already be the final one, as the calculated friction factor will remain constant (being independent from the value of the Reynolds number). If the Moody diagram is used, an alternative approach can be applied for determination of the friction factor. The calculation starts by assuming the rough turbulence regime:

- Read the initial λ value based on the k/D ratio (or calculate it by applying Equation 9.37).
- Calculate the velocity by applying Equation 9.46.
- Calculate *Re* from Equation 9.34. Check on the graph if the obtained Reynolds number corresponds to the assumed λ and k/D. If not, read the new λ-value for the calculated Reynolds number and repeat steps 2 and 3. Once a sufficient accuracy for the λ-value has been reached, the velocity calculated from this value will be the final velocity.

Both approaches are valid for a wide range of input parameters. The first one is numerical, i.e. suitable for computer programming. The second one is simpler for manual calculations; it is shorter and avoids estimation of the velocity in the first iteration. However, this approach relies very much on accurate reading of the values from the Moody diagram.

Example 9.16: For the system from Example 9.6, calculate the maximum capacity that can be conveyed if the pipe diameter is increased to $D = 500$ mm and the head-loss has been limited to 10 m per km of the pipe length. The roughness factor for the new pipe diameter can be assumed at $k = 0.1$ mm.

Assumed velocity 1 *m/s*= for the temperature T= 10°C, the kinematic viscosity, $\mu = 1.31\times 10^{-6}\ \mathrm{m^2/s}$. With diameter D=500 mm, the Reynolds number takes the value of:

$$Re = \frac{vD}{\mu} = \frac{1\times 0.5}{1.31\times10^{-6}} = 3.8\times 10^{5}$$

And the function factor is λ from Barr's Equation equals:

$$\lambda = 0.25/\log^{2}\left[\frac{5.1286}{\mathrm{Re}^{0.89}} + \frac{k}{3.7D}\right]$$

$$\lambda = 0.25/\log^{2}\left[\frac{5.1286}{\left(3.8\times10^{5}\right)^{0.89}} + \frac{k}{3.7\times 500}\right] \approx 0.016\ m/s$$

The new value of the velocity based on the maximum-allowance hydraulic gradient $S_{max} = 10/1000 = 0.01$ is calculated from Equation

$$v = \sqrt{\frac{2gDS}{\lambda}} = \sqrt{\frac{2\times 9.81\times 0.5\times 0.01}{0.016}} = 2.48\ m/s$$

The results differ substantially from the assumed velocity and the calculation should be repeated in the second iteration with this value as a new assumption. Hence:

$$Re = \frac{vD}{\mu} = \frac{2.48\times 0.5}{1.31\times10^{-6}} = 9.5\times 10^{5}$$

And the function factor is λ from Barr's Equation equals:

$$\lambda = 0.25/\log^{2}\left[\frac{5.1286}{\left(9.5\times10^{5}\right)^{0.89}} + \frac{k}{3.7\times 500}\right] \approx 0.015\ m/s$$

The new resulting velocity will be:

$$v = \sqrt{\frac{2gDS}{\lambda}} = \sqrt{\frac{2\times 9.81\times 0.5\times 0.01}{0.015}} = 2.57\ m/s$$

Which can be considering as a sufficiently accurate result, as any additional iteration that can be done is not going to change this value. Finally, the maximum flow can be discharged at S=0.01 equals:

$$Q = v\frac{D^{2}\pi}{4} = 2.57\frac{0.5^{2}\times 3.14}{4} = 0.5\,m^{3}/s \approx 1800\ m^{3}/h$$

In the alternative approach, the initial λ value assumes the rough turbulent zone can be read from the Moody diagram in figure 9.28. For value of $k/D = 0.1/500 = 0.0002$, it is approximately 0.014. The calculation from the equation:

$$\lambda = 0.25 \ \log^2 \left[\frac{k}{3.7D} \right] = 0.25/\log^2 \left[\frac{0.1}{3.7 \times 500} \right] = 0.0137$$

With the value:

$$v = \sqrt{\frac{2gDS}{\lambda}} = \sqrt{\frac{2 \times 9.81 \times 0.5 \times 0.01}{0.137}} = 2.66 \ m/s$$

The Reynolds number then becomes:

$$Re = \frac{vD}{\mu} = \frac{2.66 \times 0.5}{1.31 \times 10^{-6}} = 1.0 \times 10^6$$

Which means that the new reading for λ is closer to the value of 0.015 ($k/D = 0.0002$). Repeated calculation of the velocity and the Reynolds number with this figure leads to a final result as in the first approach.

9.24.3 Optimal diameter

In the calculation of optimal diameters, the input data are: L, k, Q, ΔH (or S), and T. The result is diameter D. The iteration procedure is similar to the one in the previous case, with the additional step of calculating the input diameter based on the assumed velocity:

- Assume the initial velocity (usually, $v = 1$ m/s).
- Calculate the diameter from the velocity/flow relation. $D^2 = 4Q/(v\pi)$.
- Calculate *Re* from Equation 9.34.
- Based on the *Re* value, choose the appropriate friction loss equation,9.33, and determine the λ-factor. For selected *Re* and k/D values, the Moody diagram can also be used instead.
- Calculate the velocity from Equation 9.46.

If the values of the assumed and determined velocity differ substantially, above steps 2–5 should be repeated by taking the calculated velocity as the new input. After a sufficient accuracy has been achieved, the calculated diameter can be rounded up to a first higher (manufactured) size. This procedure normally requires more iterations than for the calculation of the maximum pipe capacity. The calculation of the diameter from an assumed velocity is needed as the proper diameter assumption is often difficult and an inaccurate guess of D accumulates more errors than in the case of the assumption of velocity.

Example 9.17: In case the flow from the previous example has to be doubled to Q = 3600 m^3/h, calculate the diameter that would be sufficient to convey it without increasing the hydraulic gradient. The other input parameters remain the same as in Example 9.14.

Assume velocity v=1 m/s, based on this velocity, the diameter D,

$$D = 2 \times \sqrt{\frac{Q}{v\pi}} = 2 \times \sqrt{\frac{1}{1 \times 3.14}} = 1.128\, m$$

And the Reynolds number

$$R_e = \frac{vD}{\mu} = \frac{1 \times 1.28}{1.31 \times 10^{-6}} = 8.6 \times 10^{5}$$

And the function factor is λ from Barr's Equation equals:

$$\lambda = 0.25/\log^2 \left[\frac{5.1286}{\mathrm{Re}^{0.89}} + \frac{k}{3.7D} \right]$$

$$\lambda = 0.25/\log^2 \left[\frac{5.1286}{\left(8.6 \times 10^{5}\right)^{0.89}} + \frac{0.1}{3.7 \times 1123} \right]$$

$$\lambda \approx 0.0135$$

And at S_{max}=10/1000=0.01 the velocity equation becomes:

$$v = \sqrt{\frac{2gDS}{\lambda}} = \sqrt{\frac{2 \times 9.81 \times 1.128 \times 0.01}{0.0135}} = 4.04\, m/s$$

The result is substantially different than the assumed velocity (V_{ass}) and the calculation has to be continued with several more iterations. The results after applying the same procedure are shown in the following table:

With the final value for the diameter of D = 650 mm, the manufactured size would be, say, D = 700 mm.

Iter.	v_{ass} (m/s)	D (mm)	Re (-)	λ (-)	v_{calc} (m/s)
2	4.04	561	1.7×10^6	0.0141	2.79
3	2.79	676	1.4×10^6	0.0139	3.09
4	3.09	642	1.5×10^6	0.0139	3.01
5	3.01	650	1.5×10^6	0.0139	3.03

9.24.4 Pipe charts and tables

Straightforward determination of the required pressures, flows or diameters is possible by using the *pipe charts* or *pipe tables*. These are created by combining the Darcy–Weisbach and Colebrook–White Equations. Substituting λ and Re in Equation 9.32, by using Equations 9.28 and 9.21 respectively, yields the following equation:

$$v = -2\sqrt{2gDS}\,\log\left[\frac{2.5lu}{D\sqrt{2gDS}} + \frac{k}{3.7D}\right] \qquad 9.47$$

For a fixed k-value and the water temperature (i.e. the viscosity), the velocity can be calculated for common ranges of D and S.

The chart in Figure 9.37 shows an example of a flow rate of 20 l/s (top axis) passing through a pipe of diameter D = 200 mm (bottom axis). From the intersection of the lines connecting these two values it emerges that the corresponding velocity (left axis) and hydraulic gradient (right axis) would be around 0.6 m/s and 2 m/km, respectively. The same flow rate in a pipe D = 300 mm yields much lower values: the velocity would be below 0.3 m/s and the gradient around 0.3 m/km. It is important to note that the particular graph or table is valid for *one single* roughness value and *one single* water temperature. Although the variation of these parameters has a smaller effect on the friction loss than the variation of D, v or Q, this limits the application of the tables and graphs if the values specifically for k differ substantially from those used in the creation of the table/graph. As an example, Table 9.14 shows the difference in the calculation of hydraulic gradients for the range of values for k and T. In former times, the pipe charts and tables were widely used for hydraulic calculations. Since the development of PC-spreadsheet programmes, their relevance has somewhat diminished. Nevertheless, they are a useful help in providing quick and straightforward estimates of pipe discharges for given design layouts.

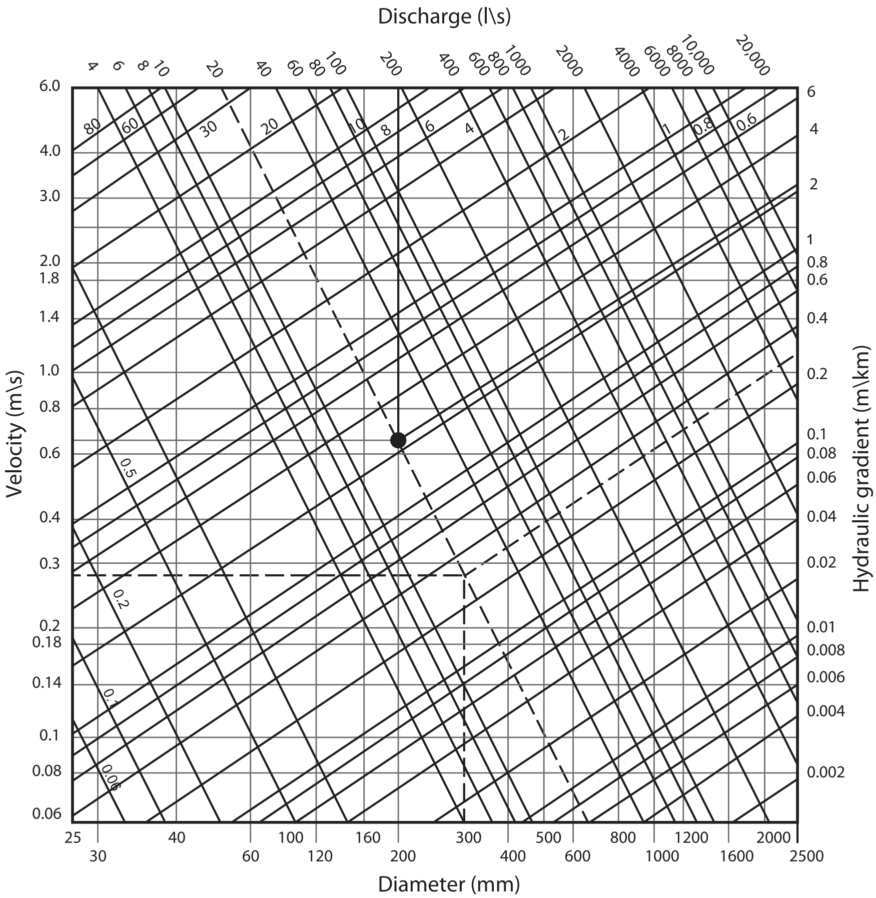


Figure 9.37: **Example of a pipe chart.**

Table 9.14: **Hydraulic gradient *S* (-) in pipe *D* = 400 mm at *Q* = 200 l/s.**

Parameter	k = 0.01 mm	k = 0.1 mm	k = 01 mm	k = 5 mm
T = 10 °C	0.0044	0.00	0.00	0.00
T = 20 °C	0.0042	0.0051	0.0081	0.0132
T = 40 °C	0.0040	0.0049	0.0081	0.0132

Example 9.18: Using the pipe tables, determine the maximum discharge capacity for pipe D = 800 mm for the following roughness values: k = 0.01, 0.5, 1 and 5 mm and the maximum-allowed hydraulic gradients of S = 0.001, 0.005, 0.01 and 0.02, respectively. The water temperature can be assumed at T =10°C.

Solution:

The following table shows the results read for pipe $D = 800$ mm from the tables:

Discharge flows Q (l/s) for pipe $D = 800$ mm (for $T = 10°C$)

Parameter	k = 0.01 mm	k = 0.1 mm	k = 01 mm	k = 5 mm
S = 0.001	559.5	465.6	432.3	348.3
S = 0.005	1336.2	1052.5	972.8	780.0
S = 0.01	1936.8	1492.6	1377.8	1103.6
S = 0.02	2800.1	2115.1	1950.6	1561.1

The results suggest the following two conclusions:

- For fixed values of S, the discharge capacity is reduced by the increase of the roughness value. In other words, the pipes start to loose their conveying capacity as they get older, which is reflected in reality by the drop of demand and/or pressure.
- The discharge at the fixed k-value will increase by allowing the higher hydraulic gradient. In other words, if more of a friction loss is allowed in the network, more water will be distributed but at higher operational costs (because of additional pumping).

9.24.5 Equivalent diameters

During planning of network extensions or renovations, the alternative of laying single pipe or pipes connected in parallel or series is sometimes compared. To provide a hydraulically equivalent system, the capacity and hydraulic gradient along the considered section should remain unchanged in all options. Those pipes are then of equivalent diameters (see Figure 9.38). Each pipe in the parallel arrangement creates the same friction loss, which is equal to the total loss at the section. The total capacity is the sum of the flows in all pipes. Hence, for n pipes it is possible to write:

$$\Delta H_{eqv} = \Delta H_1 = \Delta H_2 = ... = \Delta H_n$$

$$Q_{eqv} = Q_1 + Q_2 + ... + Q_n$$

Pipes in parallel are more frequently of the same diameter, allowing for easier maintenance and handling of irregular situations. Furthermore, they will often be laid in the same trench i.e. along the same route and can therefore be assumed to be of the same length in which case the slope of the hydraulic grade line for all pipes will be equal. Nevertheless, the equation $S_{eqv} = S_1 = S_2 \ldots = S_n$ is not always

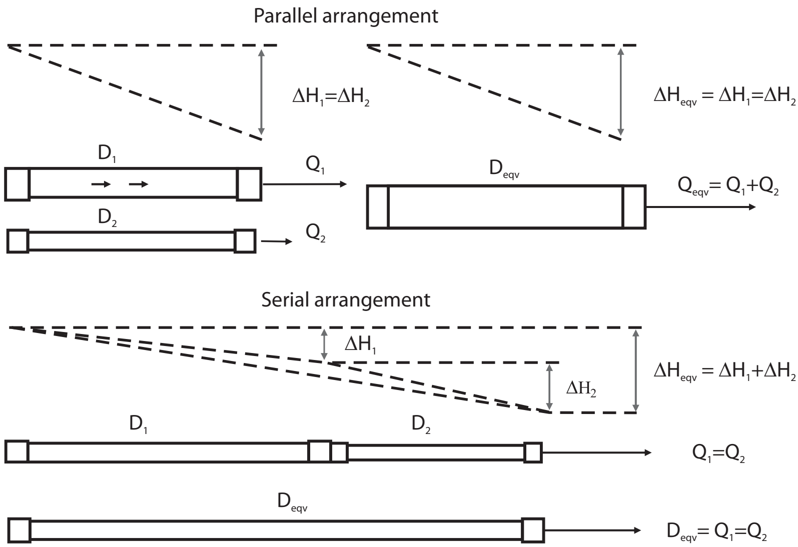


Figure 9.38: **Equivalent diameters.**

true as the pipes connected in parallel need not necessarily be of identical length. For pipes in series, the basic hydraulic condition is that each pipe carries the same flow rate. The total energy loss is the sum of the losses in all pipes. If written for n pipes:

$$\Delta H_{eqv} = \Delta H_1 = \Delta H_2 = \dots = \Delta H_n$$

$$Q_{eqv} = Q_1 + Q_2 + \dots + Q_n$$

Equation $S_{eqv} = S_1 + S_2 + \dots \dots + S_n$, will not normally be true except in the hypothetical case of $S_1 = S_2 = \dots = S_n$

The hydraulic calculation of the equivalent diameters further proceeds based on the principles of the single pipe calculation.

Example 9.19: A pipe $L = 550$ m, $D = 400$ mm, and $k = 1$ mm transports the flow of 170 l/s. By an extension of the system this capacity is expected to grow to 250 l/s. Two alternatives to solve this example are considered:

- To lay a parallel pipe of smaller diameter on the same route, or
- To lay a parallel pipe of the same diameter on a separate route with a total length $L = 800$ m.

Using the hydraulic tables for water temperature $T = 10^{\circ}$C:

- Determine the diameter of the pipe required to supply the surplus capacity of 80 l/s in the first alternative,
- Determine the discharge of the second pipe D = 400 mm in the second alternative.

In both cases, the absolute roughness of the new pipes can be assumed to be k = 0.1 mm.

Solution:

In the hydraulic tables (for $T = 10^{\circ}C$), the diameter D = 400 mm conveys the flow Q = 156.6 l/s for the hydraulic gradient S = 0.005 and Q = 171.7 l/s for S = 0.006. Assuming linear interpolation (which introduces negligible error), the flow of 170 l/s will be conveyed at S = 0.0059, leading to a friction loss $h_f = S \times L = 0.0059 \times 550 = 3.25$ mwc. This value is to be maintained in the design of the new parallel pipe. Laying the second pipe in the same trench (i.e. with the same length) should provide an additional flow of 80 l/s. From the hydraulic tables for k = 0.1 mm the following closest discharge values can be read:

Discharge flows (l/s) for pipe k = 0.1 mm

Parameter	D = 250 mm	D = 300 mm
S = 0.005	57.2	92.5
S = 0.006	62.9	101.7

Which suggests that the manufactured diameter of 300 mm is the final solution. The flow rate to be conveyed at S = 0.0059 would be Q = 100.8 l/s (after interpolation) leading to a total supply capacity of 270.8 l/s.

In the second case, the parallel pipe D = 400 mm follows an alternative route with a total length of L = 800 m. The value of the hydraulic gradient will be consequently reduced to S = 3.25 / 800 = 0.0041. The hydraulic tables give the following readings closest to this value:

Discharge flows (l/s) for pipe k = 0.1 mm

Parameter	D = 400 mm
S = 0.004	175.6
S = 0.005	197.3

Despite the longer route, this pipe is sufficiently large to convey capacities far beyond the required 80 l/s. For S = 0.0041, discharge Q = 177.8 l/s and the total supplying capacity from both pipes equals 347.8 l/s. Hence, more water but at higher investment costs.

9.25 Serial and Branched Networks

Calculation of serial and branched networks is entirely based on the methods used for single pipes. The differences in hydraulic performance occur between the branched systems with one supply point and those that have more than one supply point.

9.25.1 Supply at one point

With known nodal demands, the flows in all pipes can easily be determined by applying the Continuity Equation, starting from the end points of the system (Figure 9.39).

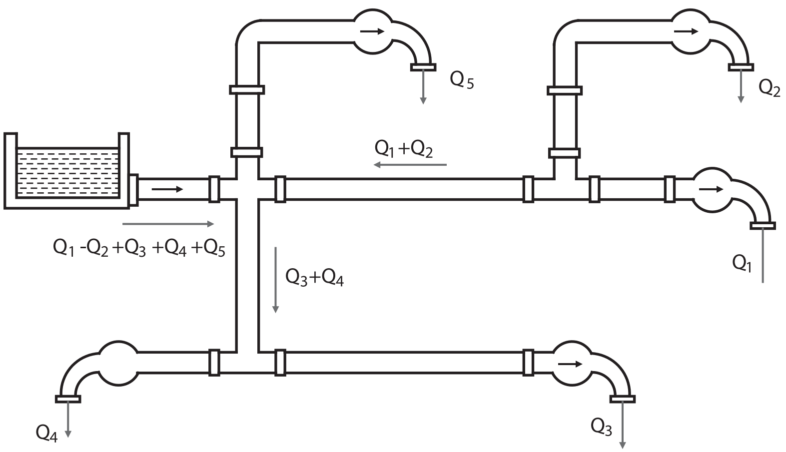


Figure 9.39: **Branched network with a single supply point.**

If the diameters of the pipes are also known, the head-loss calculation resulted in the hydraulic gradient S for each pipe. In the next step the piezometric heads, and consequently the pressures, will be calculated for each node starting from the node assumed to have the minimum pressure. In this respect potentially critical nodes are those with either high elevation and/or nodes located faraway from the source. Adding or subtracting the head-losses for each pipe, depending on the flow direction, will determine all other heads including the required piezometric head at the supply point. Calculation of the piezometric heads in the opposite direction, starting from the known value at the source, is also possible; this shows the pressures in the system available for specified head at the supplying point. In situations where pipe diameters have to be designed, the maximum allowed hydraulic gradient must be included in the calculation input. The pipe charts/tables are required here, leading to actual values of the hydraulic gradient for each pipe based on the best available (manufactured) diameter. Finally, the pressures in the system will also be determined either by setting the minimum pressure criterion or the head available at the supply point.

9.25.2 Supply at several points

For more than one supply point, the contribution from each source may differ depending on its piezometric head and distribution of nodal demands in the system. In this case, flows in the pipes connecting the sources are not directly known from the Continuity Equation. These flows can change their rate and even reverse the direction based on the variation of nodal demands. Figures 9.40 and 9.41 show an example of anticipated demand increase in node one.

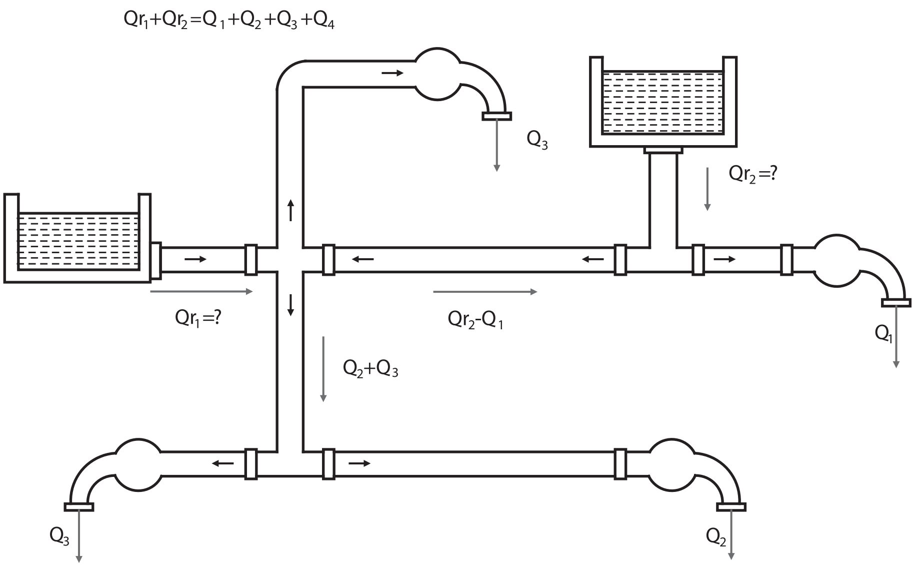


Figure 9.40: **Branched network with two supply points.**

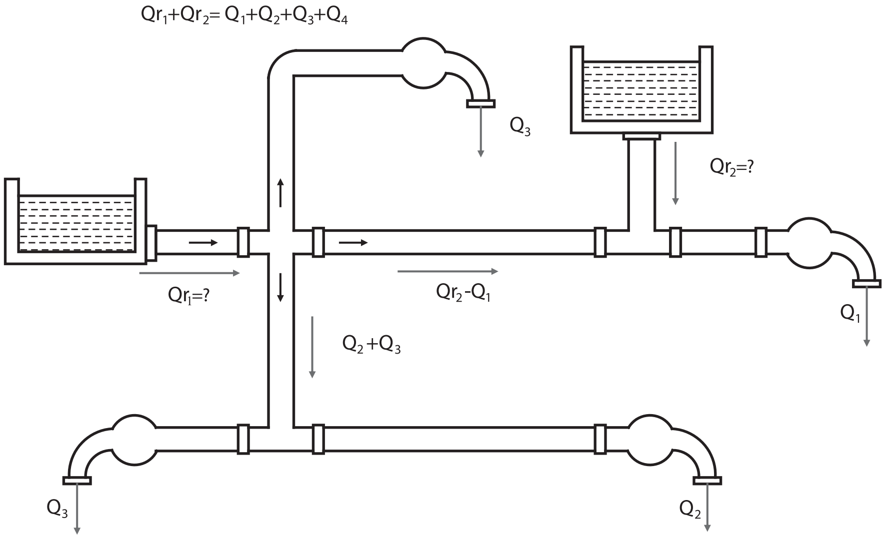
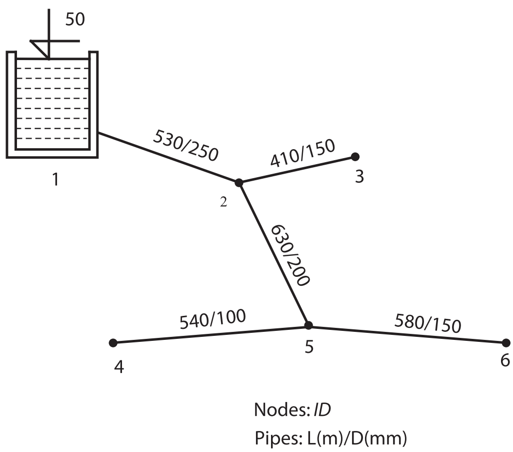


Figure 9.41: **Branched network with two supply points, showing an increase of nodal flow Q_1.**

Except for the chosen source, fixed conditions are required for all other sources existing in the system: a head, discharge or the hydraulic gradient of the connecting pipe(s). For the remainder, the calculation proceeds in precisely the same manner as in the case of one supply point.

Example 9.20: For the following branched system, calculate the pipe flows and nodal pressures for surface level in the reservoir of H = 50 msl. Assume for all pipes k = 0.1 mm, and water temperature of 10°C.



	1	2	3	4	5	6
Z (msl)	-	-	22	17	25	20
Q (l/s)	75.6	75.6	22.1	10.2	18.5	14.4

Solution:

The total supply from the reservoir equals the sum of all nodal demands, which is 75.6 l/s. Applying the Continuity Equation in each node, both the flow rate and its direction can be determined; each pipe conveys the flow that is the sum of all downstream nodal demands. The pipe friction loss will be further calculated by the approach discussed in Example 9.6. If the hydraulic tables are used, the friction loss will be calculated from interpolated hydraulic gradients at a given diameter and flow rate (for fixed k and T). The results of the calculation applying the Darcy Weisbach Equation are shown in the following table.

Pipe	D(mm)	Q (l/s)	v (m/s)	Re (-)	λ (-)	S (-)	L (m)	h_f (mwc)
1-2	250	75.6	1.54	2.9×10^5	0.018	0.0086	530	4.55
2-3	150	22.1	1.25	1.4×10^5	0.020	0.00108	410	4.43
2-5	200	43.1	1.37	2.1×10^5	0.19	0.0090	630	5.70
5-4	100	10.2	1.30	9.9×10^5	0.022	0.00192	540	10.38
5-6	150	14.4	0.81	9.4×10^5	0.021	0.0048	580	2.78

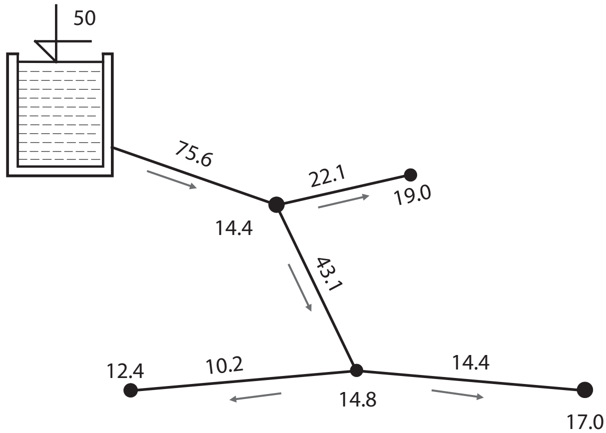
Finally, the pressure in each node is calculated by subtracting the friction losses starting from the reservoir surface level and further deducting the nodal elevation from the piezometric heads obtained in this way. The final results are shown in the following table and figure.

	1	2	3	4	5	6
H (msl)	50	45.45	41.02	29.37	39.75	36.96
p (mwc)	-	33.45	19.02	12.37	14.75	16.96

The lowest pressure appears to be in node 4 (12.4 mwc) resulting from a relatively small diameter (causing large friction loss) of pipe 5–4.

9.26 Looped Networks

Kirchoff's Laws The principles of calculation as applied to single pipes are not sufficient in case of looped networks. Instead, a system of equations is required which can be solved by numerical algorithm. This system of equations is based on the analogy with two electricity laws known in physics as Kirchoff's Laws.



Translated to water distribution networks, these laws state that:

- The sum of all ingoing and outgoing flows in each node equals zero ($\sum Q_i = 0$).
- The sum of all head-losses along pipes that compose a complete loop equals zero ($\sum \Delta H_i = 0$).

The first law is essentially the mass conservation law, resulting in the Continuity Equation that must be valid for each node in the system. From the second law, it emerges that the hydraulic grade line along one loop is also continuous, just as the flow in any node is. The number of equations that can be formulated applying this law equals the number of loops. For example, in the simple network from Figure 9.42 in the clockwise direction, this yields:

$$(H_r - H_1) + (H_1 - H_2) + (H_2 - H_3) - (H_r - H_3) = 0$$

9.26.1 Hardy Cross methods

Two similar iterative procedures can be derived from Kirchoff's Laws:

- The method of balancing heads.
- The method of balancing flows.

These methods, known in literature under the name of Hardy Cross (published in 1936 and developed further by Cornish in 1939), calculate the pipe flows and nodal piezometric heads in looped systems for a given input, which is:

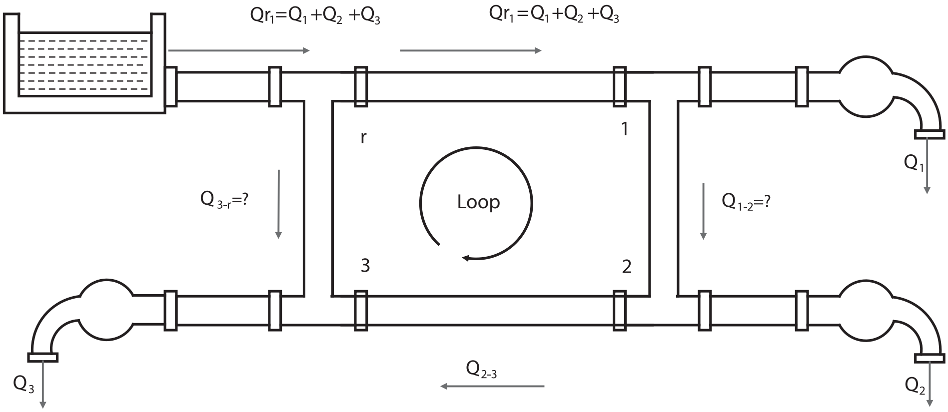


Figure 9.42: **Looped network.**

- for pipes: length *L*, diameter *D*, absolute roughness *k* and minor loss factor ξ,
- for nodes: nodal discharge *Q* and elevation *Z*.

Pressure in at least one node has to be fixed, which will influence the pressure in the rest of the system. This is usually a supply point. Successive calculation of the loops (nodes) is executed by following the following steps:

Method of balancing heads

- Flows from an initial guess are assigned to each pipe. However, these must satisfy the Continuity Equation in all nodes.
- Head-loss in each pipe is then calculated starting from Equation 9.28.
- The sum of the head-losses along each loop is checked.
- If the head-loss sum at any loop is outside of the required accuracy range, 0 $\varepsilon_{\Delta H}$ mwc, the following flow correction has to be introduced for each pipe within that loop (total n pipes):

$$\delta Q_j = \frac{-\sum_{j=1}^{n} \Delta H_j}{2\sum_{j=1}^{n} \left| \Delta H_j / Q_j \right|} \qquad 9.47$$

- Correction δ*Q* is applied throughout the loop taking consistent orientation: clockwise or anti-clockwise. This has implications for the value of the pipe flows, which will be negative if their direction counters the adopted orientation. The positive/negative sign of the correction should also be taken into account while adding it to the current pipe flow.

- The iteration procedure is carried out for the new flows, $Q + \delta Q$, repeated in above steps 2–5, until $\varepsilon_{\Delta H}$ is satisfied for all loops.
- After the iteration of flows and head-losses is completed, the pressures in the nodes can be determined from the reference node with fixed pressure, taking into account the flow directions.

The calculation proceeds simultaneously for all loops in the network, with their corresponding corrections δQ being applied in the same iteration. In case of the pipes shared between the two neighboring loops, the sum of the two δQ corrections should be applied. The flow continuity in the nodes will not be affected in this case; assuming uniform orientation for both loops will reverse the sign of the composite δQ in one of them.

If the system is supplied from more than one source, the number of unknowns increases. Dummy loops have to be created by connecting the sources with dummy pipes of fictitious *L, D* and *k,* but with a fixed ΔH equal to the surface elevation difference between the connected reservoirs. This value has to be maintained throughout the entire iteration process.

Method of balancing flows

- The estimated piezometric heads are initially assigned to each node in the system, except for the reference i.e. fixed pressure node(s). An arbitrary distribution is allowed in this case.
- The piezometric head difference is determined for each pipe.
- Flow in each pipe is determined starting from the head-loss Equation 9.28.
- The Continuity Equation is checked in each node.
- If the sum of flows in any node is out of the requested accuracy range, 0 ε_Q m^3/s, the following piezometric head correction has to be introduced in that node (n is the number of pipes connected in the node):

$$\delta H_i = \frac{2\sum_{i=1}^{n} \Delta H_j}{2\sum_{i=1}^{n} \left| \Delta H_j / Q_j \right|} \qquad 9.48$$

- The iteration procedure is continued with the new heads, $H + \delta H$, repeated in above steps 2–5, until ε_Q is satisfied for all nodes. Unlike in the method of balancing heads, faster convergence in the method of balancing flows is reached by applying the corrections consecutively.

As a consequence, the flow continuity in some nodes will include the pipe flows calculated from the piezometric heads of the surrounding nodes from the same iteration. The required calculation time for both methods is influenced by the size of the network. The balancing head method involves systems with a smaller number of equations, equal to the number of loops, which saves time while doing the calculation manually. The balancing flow method requires a larger system of equations, equal to the number of nodes.

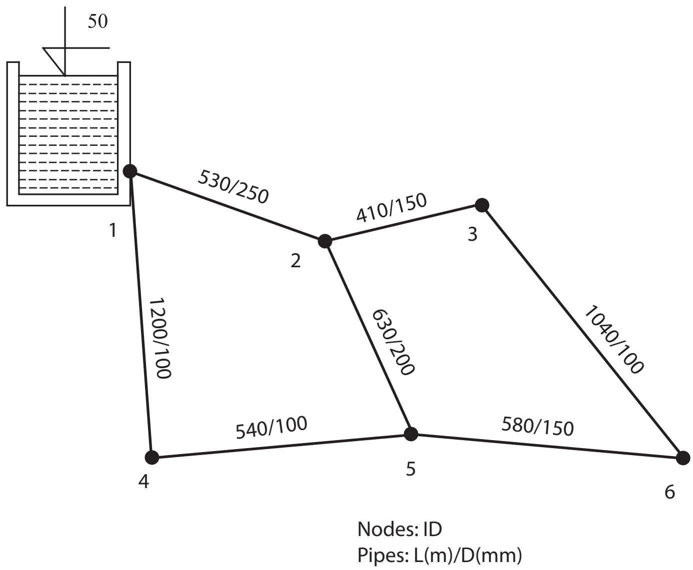
However, this method excludes the identification of loops, which is of some advantage for computer programming. The layout and operation of the system may have an impact on convergence in both methods. In general, faster convergence is reached by the balancing head method.

The Hardy Cross methods were widely used in the pre-computer era. The first hydraulic modeling software in water distribution was also based on these methods. Several modifications have been introduced in the meantime. The balancing flow method was first developed for computer applications while the balancing head method still remains a preferred approach for manual calculations of simple looped networks. Both methods are programmable in spreadsheet form, which helps in reducing the calculation time in such cases.

Example 9.21: To improve the conveyance of the system from Example 9.20, nodes one and four as well as nodes three and six have been connected with pipes D = 100 mm and L = 1200 and 1040 m, respectively (k = 0.1 mm in both cases). Calculate the pipe flows and nodal pressures for such a system by applying the balancing head method.

Solution:

Two loops are created from the branched system after adding the new pipes. The calculation starts by distributing the pipe flows arbitrarily, but satisfying the Continuity Equation in each node. The next step is to calculate the friction losses in each loop, as the following tables show (negative values mean the reverse direction, from the right node to the left one):



Loop one- Iteration one

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
1-2	250	50.6	1.03	2.12
2-5	200	20.2	0.64	1.36
5-4	100	-14.8	-1.88	-21.17
4-1	100	-25.0	-3.18	-129.89

The sum of all friction losses, which should be equal to 0 for correct flow rate values, is in this case $\sum h_f = -147.59$ mwc (selecting the clockwise direction). Thus, the correction of all pipe flows in Loop One will be required in the new iteration. From Equation 9.47 (for $\Delta H = h_f$) this correction becomes $\delta Q = 10.961$/s.

In the case of Loop Two:

Loop two- Iteration one

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
5-2	250	-20.20	-0.64	-1.36
2-3	150	20.00	1.13	3.66
3-6	100	-2.108	-0.27	-1.06
6-5	150	-16.50	-0.93	-3.60

The sum of all friction losses in this case is $\sum h_f$=-2.35 mwc, which is closer to the final result but also requires another flow correction. After applying Equation 9.47, δQ = 1.21 l/s. In the second iteration, the following results were achieved after applying the pipe flows $Q+\delta Q$:

Loop one- Iteration two

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
1-2	250	62.56	1.25	3.07
2-5	200	29.95	0.95	2.85
5-4	100	-3.84	-0.49	-1.66
4-1	100	-14.04	-1.79	-42.53

$\sum h_f$ = -38.27 mwc and therefore δQ = 5.31 l/s

Loop two- Iteration two

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
5-2	250	-29.95	-0.95	-2.85
2-3	150	21.21	1.20	4.10
3-6	100	-0.89	-0.11	-0.23
6-5	150	-15.29	-0.87	-3.12

$\sum h_f$= -2.10 mwc and therefore δQ = 1.40 l/s

The new flow in pipes 2–5 shared between the loops has been obtained by applying the correction δQ of both loops, i.e. 20.20 + 10.96 - 1.21 = 29.95 l/s. This pipe in Loop Two has a reversed order of nodes and therefore $Q_{5\text{-}2}$ = - 20.20 + 1.21 - 10.96 = - 29.95 l/s. Hence, the corrected flow of the shared pipe maintains the same value in both loops, once with a positive and once with a negative sign. In the rest of the calculations:

Loop one- Iteration three

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
1-2	250	66.86	1.36	3.60
2-5	200	33.85	1.80	3.60
5-4	100	1.46	0.19	-0.29
4-1	100	-8.74	-1.11	-17.17

$\sum h_f$ = -9.69 mwc and therefore δQ = 2.09 l/s

$\sum h_f$ = -0.02 mwc and therefore δQ = 0.01 l/s

Loop two- Iteration two

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
5-2	250	-33.85	-1.08	-3.60
2-3	150	22.61	1.28	4.63
3-6	100	0.51	0.06	0.09
6-5	150	-13.89	-0.79	-2.60

$\sum h_f = -1.48$ mwc and therefore $\delta Q = 1.11$ l/s

Loop one- Iteration four

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
1-2	250	68.95	1.40	3.82
2-5	200	34.83	1.11	3.80
5-4	100	3.55	0.45	1.43
4-1	100	-6.65	-0.85	-10.25

$\sum h_f = -1.20$ mwc and therefore $\delta Q = 0.29$ l/s

Loop two- Iteration four

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
5-2	250	-34.83	-1.11	-3.80
2-3	150	23.72	1.34	5.07
3-6	100	1.62	0.21	0.66
6-5	150	-12.78	-0.72	-2.22

$\sum h_f = -0.29$ mwc and therefore $\delta Q = 0.16$ l/s

Loop one - Iteration five

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
1-2	250	69.24	1.41	33.85
2-5	200	34.96	1.11	3.82
5-4	100	3.84	0.49	1.65
4-1	100	-6.36	-0.81	-9.44

$\sum h_f = -0.12$ mwc and therefore $\delta Q = 0.03$ l/s

Loop one- Iteration six

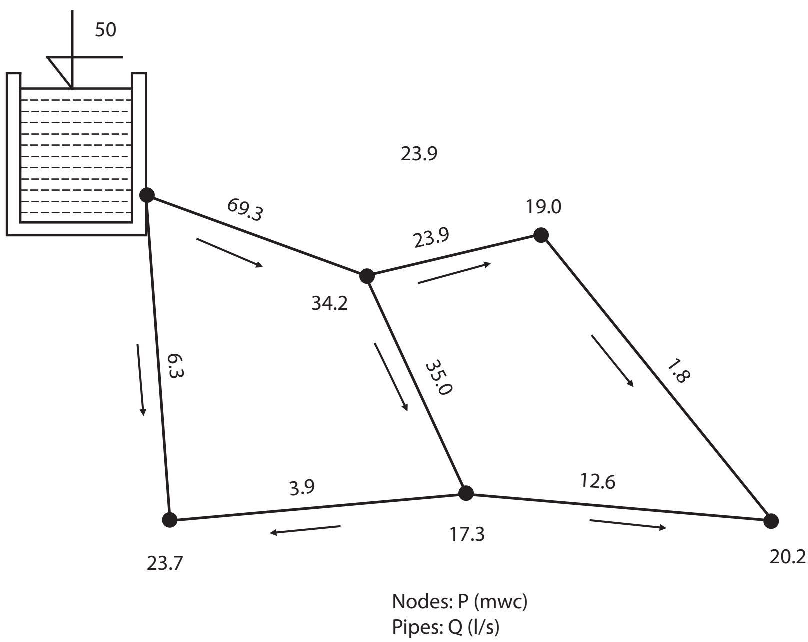
Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
1-2	250	69.26	1.41	3.85
2-5	200	34.94	1.11	3.82
5-4	100	3.86	0.49	1.68
4-1	100	-6.34	-0.81	-9.36

Loop two- Iteration six

Pipe	D(mm)	Q (l/s)	v (m/s)	h_f (mwc)
5-2	250	-34.94	-1.11	-3.82
2-3	150	23.92	1.35	5.15
3-6	100	1.82	0.23	0.82
6-5	150	-12.58	-0.71	-2.16

$\sum h_f$ = -0.01 mwc and therefore δQ = 0.01 l/s

As the tables show, the method already provides fast convergence after the first two to three iterations and continuation of the calculations does not add much to the accuracy of the results while it takes time, specifically in the case of manual calculations. Determination of the nodal pressures will be done in the same way as in the case of branched systems: starting from the supply point with a fixed piezometric head or from the point with minimum pressure required in the system. In each pipe, the unknown piezometric head on one side is obtained either by adding the pipe friction loss to the known downstream piezometric head, or by subtracting the pipe friction loss from the known upstream piezometric head. The final results are shown in the following figure:



9.26.2 Linear theory

The Newton Raphson and the Linear Theory methods succeeded the Hardy-Cross methods, as the main approach for solving the non-linear network governing equations. The linear theory method was developed by and involves a remarkably simple linearization compared with the Newton Raphson method. When using the Darcy–Weisbach Equation:

$$\Delta H = \left(R_f + R_m\right)|Q|Q = UQ \qquad 9.49$$

Where:

$$U = \frac{(\lambda L + \xi D)|Q|}{12.1D^5} \qquad 9.50$$

The absolute value of Q in the equation 9.49 and 9.50 helps to distinguish between different flows directions (+/- sign). The following can be written for node i, assuming the inflow to the node has a negative sign (Figure 9.43):

$$Q_i - \sum_{j=1}^{n} Q_{ij} = 0 \qquad 9.51$$

Index n represents the total number of pipes connected to node i, while Q_i is the nodal demand (outflow).

Equation 9.51 is the satisfied in the iteration procedure with specified accuracy ε_i, for each node. Thus:

$$Q_i - \sum_{j=1}^{n} Q_{ij} = \pm\varepsilon_i = f(H_i) \qquad 9.52$$

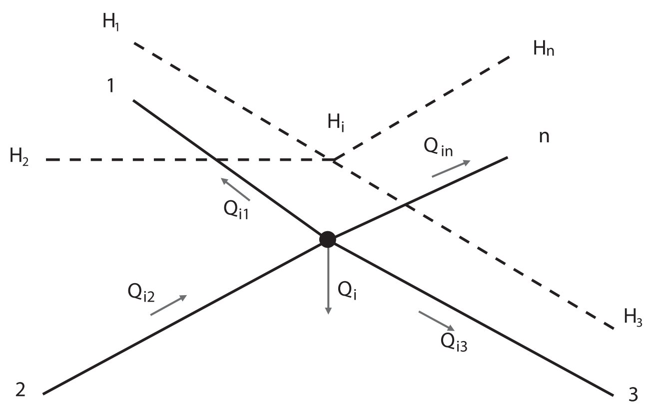


Figure 9.43: Linear theory.

and after combining Equations 9.49 and 9.52:

$$\pm\varepsilon_i = f(H_i) = Q_i - \sum_{j=1}^{n} \frac{H_j}{U_{ij}} + H_i \sum_{j=1}^{n} \frac{1}{Uij} \qquad 9.53$$

The system of linear equations given in equation 9.53, equals the total number of nodes in the network. The solution of these linear equations can be achieved by any standard procedure (i.e. algorithms for inversion and decomposition). For example, one could apply the Newton Raphson method that would give the nodal head in the $(k+1)^{th}$ iteration as follows:

$$H_i^{(k+1)} = H_i^{(k)} - \frac{f(H_i^{(k)})}{f'(H_f^{(k)})} \qquad 9.54$$

Plugging 9.53 into 9.54 yields:

$$f(H_i^{(k)}) = Q_i - \sum_{j=1}^{n} \frac{H_j^{(k)}}{U_{ij}^{(k)}} + H_i^{(k)} \sum_{j=1}^{n} \frac{1}{U_{ij}^{(k)}} \qquad 9.55$$

$$f'(H_i^{(k)}) = \sum_{j=1}^{n} \frac{1}{U_{ij}^{(k)}} \qquad 9.56$$

$$H_i^{(k+1)} = H_i^{(k)} - \left[\frac{Q_i - \sum_{j=1}^{n} H_j^{(k)} / U_{ij}^{(k)} + H_i^{(k)} \sum_{j=1}^{n} 1 / U_{ij}^{(k)}}{\sum_{j=1}^{n} 1 / U_{ij}^{(k)}} \right] \qquad 9.57$$

Finally, a factor ω that takes values between one and two can be added to improve the convergence:

$$H_i^{(k+1)} = H_i^{(k)} (1-\omega) + \left[\frac{\sum_{j=1}^{n} H_j^{(k)} / U_{ij}^{(k)} - Q_i}{\sum_{j=1}^{n} H_j^{(k)} / U_{ij}^{(k)}} \right] \omega \qquad 9.58$$

This method, known as the Successive Over-Relaxation Method (SOR), was recommended by. Equation 9.58 shows that the piezometric head in node i depend on:

- piezometric heads at the surrounding nodes $j = 1$ to n, and
- resistance U of the pipes that connect node i with the surrounding nodes.

The size of the equation matrix in this approach is $m \times n_{max}$, where m is the total number of nodes and n_{max} is a specified maximum number of nodes connected to each node in the system.

The iteration procedure is executed separately for nodes and pipes and consists of internal and external cycles.

The preparation steps are:

- Setting the initial values of the flow in each pipe. This is usually based on the velocity of 1 m/s and a given pipe diameter.
- Calculation of the U-values based on the initial flows.
- Setting the initial values of the piezometric head in each node.

As in the case of the Hardy Cross methods, at least one node must be chosen as a reference (fixed head) node, to allow determination of other nodal heads in the system. The initial head in all other nodes can be selected in relation to the fixed head value.

The iteration starts in the internal cycle, where the nodal heads are determined by Equation 9.58. The calculation for each node is repeated until $H^{(k+1)} - H^{(k)} < \varepsilon_H$ is satisfied for all nodes in the system. Thereafter, the flow will be calculated as $Q = \Delta H/U$ for each pipe, in the $(l+1)^{th}$ iteration of the external cycle. If $Q^{(l+1)} - Q^{(l)} > \varepsilon_Q$ for any of the pipes, the internal cycle will have to be repeated using the new U-values calculated from the latest flow rates. The iteration stops when the requested flow accuracy has been achieved for all pipes or the specified maximum number of iterations was reached. No matter how complicated the method looks at first glance, it is convenient for computer programming. Used for manual calculations, it will require lots of time even for a network of very few pipes. A spreadsheet application can reduce this but is by no means an alternative to a full-scale computer programme. It can however serve as a useful tool for better understanding of the principle.

The most recently used method to solve the network analysis problem, is the Gradient method that was first introduced by Todini and Pilati (1987). The linear system of equations formulated by their approach is solved using a matrix calculation described by George and Liu (1981). Next to its robustness, the additional advantage of this method is that it can handle the change of the status of system components (pumps and valves) without changing the structure of the equation matrix. The basic steps of the calculation procedure can be found in Rossman (2000).

9.27 Pressure-related Demand

It is a known fact that more water is consumed when there is a higher pressure in the system, resulting in higher water flows at the outlets. In addition to this, the leakage levels are very much pressure-sensitive, as can be seen from the British experience shown in Figure 9.44. All calculation procedures explained in the previous sections deal with pipe flows against the hydraulic gradients, with the pressures calculated afterwards. As the reference head (pressure) is set independently from the flows, some error is introduced by neglecting the relation between the demand and pressure in the system. The mathematical relation between these two is quadratic, which can be derived from the analogy with the discharge through an orifice (see Figure 9.45). The water pressure at the orifice is assumed to be atmospheric and applying the Bernoulli Equation for the cross-section just before and just after the orifice leads to the equation:

$$Q = CA\sqrt{2gh} \qquad 9.59$$

where A is the surface area of the orifice and C is a factor (<1) related to its shape. With free surface level in the reservoir, water depth h above the orifice can be compared with energy head difference ($h \approx \Delta E \approx \Delta H$), while the C-factor corresponds to the minor loss factor,

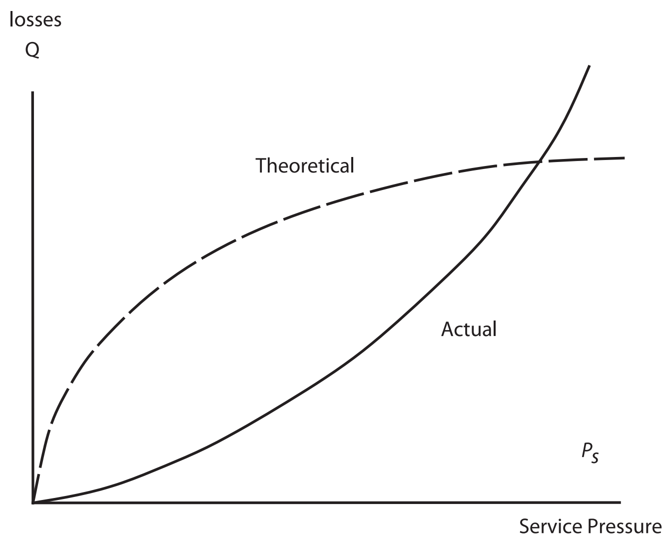


Figure 9.44: **Pressure-related leakage (PLC 1993) .**

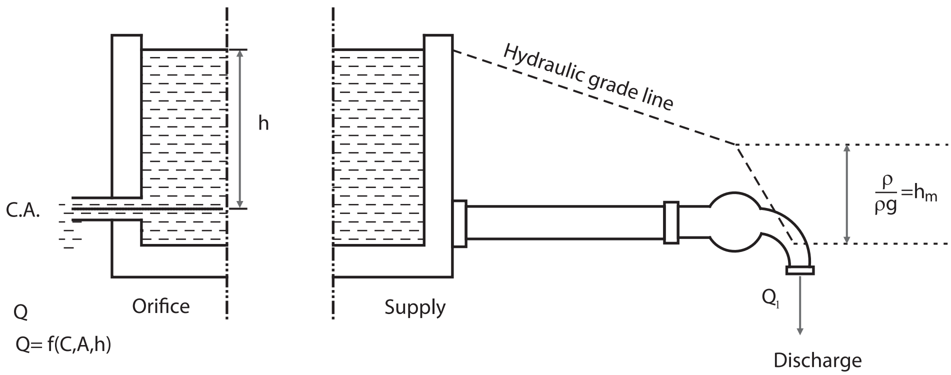


Figure 9.45: **Discharge through an orifice**

ξ. Neglecting the friction, Equation 9.59 actually has a format comparable to $h_m = R_m Q^2$ shown in Equation 9.28 and further elaborated by Equation 9.43.

Finally, it shows that the residual pressure in water distribution systems is destroyed at the tap, i.e. has in essence the status of a minor loss ($h_m = p/\rho g$). In reality, applying this logic creates two potential problems:

- Demand-driven hydraulic calculations will require a correction of the nodal discharges according to the calculated pressure, which may significantly increase the calculation time leading to an unstable iteration procedure. In other words, an input parameter (demand) becomes dependent on an output parameter (pressure).
- Resistance R_m is, in the case of hundreds of nodes supplying thousands of consumers, virtually impossible to determine. In the best possible case, a general pressure-related diagram may be created from a series of field measurements.

In Netherland, KIWA suggests a linear relation between the pressure and demand for calculations carried out for the assessment of distribution network reliability. The demand is considered independently of the pressure above a certain threshold, which is typically a pressure of 20–30 mwc (Figure 9.46). Running hydraulic calculations for low-pressure conditions without taking the pressure-related demand into consideration might result in negative pressures in some nodes. This happens if for example the supply head is too low (see Figure 9.47) or the head-losses in the system are exceptionally high. Proper interpretation of the results is necessary in this case in order to avoid false conclusions about the system operation. Applying the pressure-related demand mode in calculations causes a gradual reduction of the nodal discharges and the hydraulic gradient values, resulting in the slower drop of the reservoir levels, as Figure 9.48 shows.

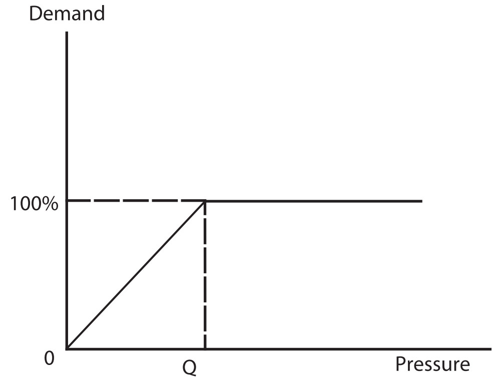


Figure 9.46: **Pressure-related demand relation.**

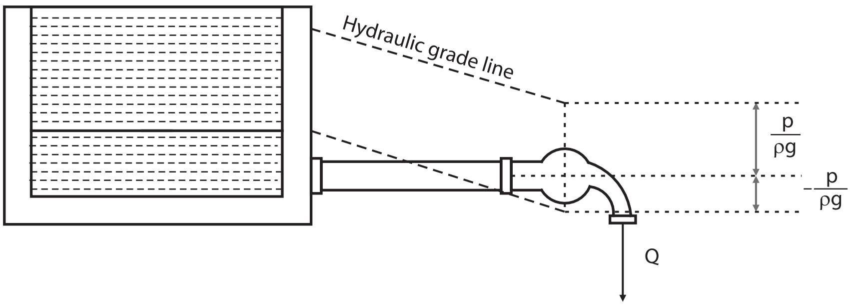


Figure 9.47: **Negative pressures - as a result of a calculation without pressure-related demand.**

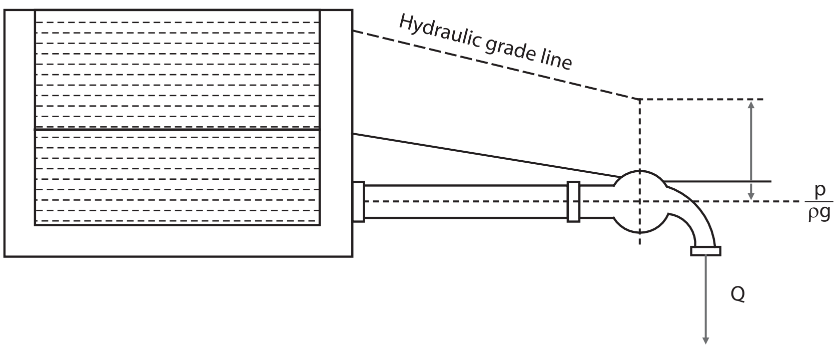


Figure 9.48: **Pressures as the result of the calculation with pressure-related demand.**

9.28 Thrust Resistant

Thrust forces occur in water mains when the pipeline changes directions, stops, or changes size. On pipelines with unrestrained joints, as used in ductile iron pipe installations, thrust blocks or restrained joints are required. For welded steel pipelines, flanged joints and lugged joints in concrete and ductile iron pipelines, other forms of anchorage are not usually required. All thrust anchorages shall be designed for a safety factor of not less than 1.5 under maximum pressure loading. The magnitude of hydrostatic thrust may be determined from the following formula:

At bends: $T = 2\pi r^2 p \sin\frac{\Delta}{2}$ 9.60

At dead end or branch: $T = \pi r^2 p$

where,

T = thrust in pounds

r = radius of pipe joints in inches

p = water pressure in psi = bend deflection angle

Δ = bend deflection angle

9.28.1 Thrust Blocks

Thrust block size is calculated based on the bearing capacity of soil:

Area of block = $L \times D = \frac{T}{a}(F.S)$ 9.61

Where,

L = length of block in feet

D = depth of block in feet

T = thrust in pounds

a = safe bearing valve for soil in psf

$F.S$ = factor of safety

Example 9.22: Calculate the thrust block bearing area required for a 12-inch pipe of a 90 degree bend, internal pressure of 120 psi including surge, allowable soil bearing pressure of 3000 psf

Solution:

$$T = 2\,(3.14)\,(36)\,(120)\,(\sin 45)$$

$T = 19{,}180\ lbs$

$$L \times D \frac{19180}{3000}(1.5) \qquad 9.62$$

$L \times D = 9.59 s.f.$

For $D = 2.5$ ft, $L = 3.8$ ft

Design of thrust blocks for vertical bends is the same as for horizontal bend. For top bends, the block must be sized to adequately resist the vertical component of thrust with the dead weight of the block, bend water in the bend and overburden. Steel straps are used to tie the pipe to the thrust block when the block is placed below the pipe and reinforcing steel may be necessary to resist tensile forces within the block. Figure 9.49 shows the typical thrust blocking.

9.29.2 Restrained Joints

Restrained joints are used as an alternate to thrust blocks to avoid uncertainties such as excavation behind a block. For ductile iron pipe, the length to be restrained is calculated as follows:

$$L = \frac{4\pi r^2 p \tan\frac{\Delta}{2}}{(2f \tan\frac{\Delta}{2}) + (Ds)} \qquad 9.63$$

where:

- L = length to be restrained on each side of bend in feet
- r = radius of pipe in inches
- p = water pressure in psi
- Δ = bend deflection angle
- D = pipe outside diameter in feet
- f = conduit frictional resistance in plf
- s = passive soil pressure in psf

Conduit frictional resistance is calculated from the following equation, using values in Table 9.15 and Table 9.16.

$$f = Af_c C_s + \pi wRHD \tan(f_p \Delta) \qquad 9.64$$

Where:

C_s = soil cohesion in psf
w = soil unit weight in pcf
R = reduction factor
H = cover above conduit in feet
f_p = ratio of pipe friction angl
Δ = bend deflection angle
A = area of block
f_c = friction co-efficient
D = pipe out side diameter in feet

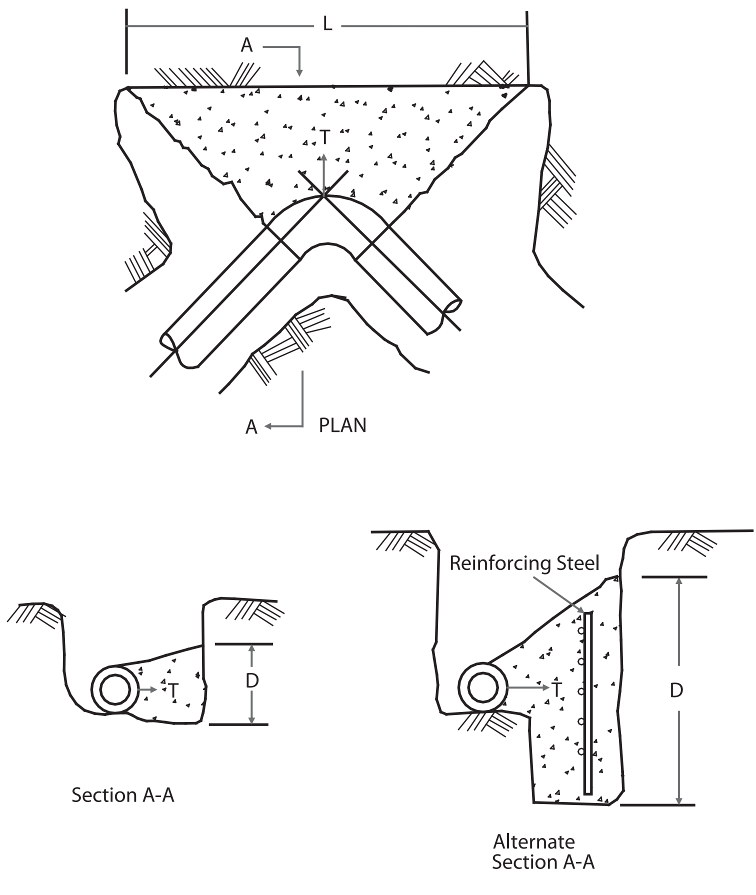


Figure 9.49: Typical thrust blocking.

Passive soil pressure is calculated according to the Rankine Theory by the following equation:

$$s = w(H + \frac{D}{2})N + 2C_s\sqrt{N}$$

$$N = \tan^2(45^\circ + \frac{\Delta}{2})$$

$$\frac{(4)(3.14)(157)(250)(1.5)(.414)}{(4)(2273)(.414) + (2.15)(5360)}$$

Table 9.15: Soil friction and cohesion factors

Soil Description	Friction Angles (degrees)	Cs	f_p	f_c
Sand, dry well graded	44.5	0	.76	0
Sand, saturated, well graded	39	0	.80	0
Silt, dry, passing 200 sieve	40	0	.95	0
Silt, saturated, passing 200 sieve	32	0	.75	0
Cohesive granular soil wet to moist	13-22	385-920	.65	.35
Clay, wet to moist at maximum compaction	11.5-16.5	460-1175	.50	.50 .50

Table 9.16: Reduction factors

Existing Conditions	Reduction factor
General Construction backfill soils compacted to critical void ratio	2/3
Well compacted backfill and selected backfill	3/4
Shallow cover H less than one half of D	½

9.29 Hydraulics of Storage and Pumps

Reservoirs and pumps are constructed to maintain the energy levels needed for water to reach the discharge points. Factors that directly influence the position, capacity and operation of these components are:

- topographical conditions,
- location of supply and demand points,
- patterns of demand variation,
- conveyance capacity of the network.

9.29.1 System characteristics

The conveyance capacity of a pipe with a known length, diameter, roughness factor and slope is described by the pipe characteristics diagram. This diagram shows the required heads at the upstream side of the pipe, which enables the supply of a range of flows while maintaining constant pressure at the pipe end. The total head required for flow *Q*, in particular, consists of a dynamic and static component (see Figure 9.50).

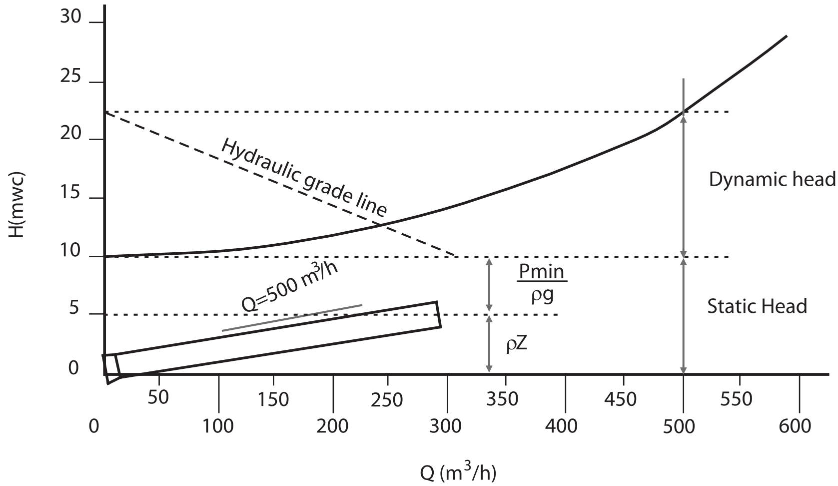


Figure 9.50: **Pipe characteristics.**

The dynamic head covers the head-losses i.e. the pipe resistance:

$$H_{dyn} = \Delta E = \Delta H = h_f + h_m \tag{9.70}$$

The static head is independent of the flow:

$$H_{st} = \frac{p_{end}}{\rho g} \pm \Delta Z \tag{9.71}$$

Where, P_{end} stands for the remaining pressure at the pipe end. In design problems where the required head is to be determined for the maximum flow expected in the pipe, the pressure at the end will be fixed at a critical value, i.e. $P_{end} = P_{min}$. Maintaining the specified minimum pressure at any flow $Q < Q_{max}$ will result in the least energy input required for water conveyance at given pipe characteristics. ΔZ in Figure 9.51 is related to the pipe slope and represents the elevation difference between the end and supplying point of the pipe. Positive ΔZ indicates the necessity of pumping while, if there is a negative value, the gravity may partly or entirely be involved in water conveyance. In the example from the figure, the static head $H_{st} = 10$ mwc comprises the minimum downstream pressure head of 5 mwc and 5 m of the elevation difference between the pipe ends. Such a pipe could deliver a maximum flow of 500 m^3/h if the head at the supply point was raised up to 22.5 mwc.

The curve for a single (transportation) pipe will be drawn for fixed *L, D* and k values, and a range of flows covering the demand variations on a maximum consumption day. The minor losses are usually ignored and the curve will be plotted using the results of the friction loss calculation for various flow rates.

System characteristics In cases when the diagram in Figure 9.50 represents a network of pipes, it will be called the system characteristics. A quadratic relation between the system heads and flows can be assumed in this case:

$$H_{dyn} = R_s Q^2 \tag{9.72}$$

where R_s is the system resistance determined from pressure measurements for various demand scenarios. The static head of the system will have the same meaning as in Eq. 9.57 except that the end in this context means the most critical point i.e. with the lowest expected pressure. That point can exist at the physical end of the system, faraway from the source, but can also be within the system if located at a high elevation (thus, inflicting high ΔZ). The system/pipe curves change their shape as the head-loss varies resulting from modification of the pipe roughness, diameter or length. This can be a consequence of:

- pipe ageing, and/or
- system rehabilitation/extension.

The curve becomes steeper in all cases as the head-loss increases, which results in the reduction of the conveying capacity. The original capacity can be restored by laying new pipes of a larger diameter, or in a parallel arrangement. Alternatively,

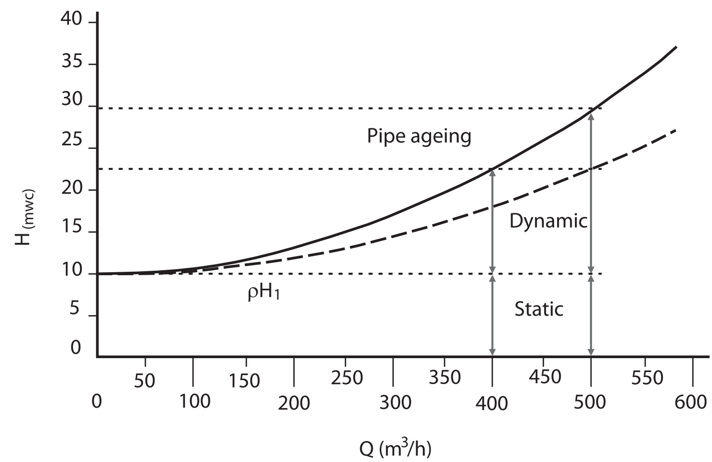


Figure 9.51: **Capacity reduction of the system.**

more energy, i.e. the higher head at the supplying side, will be needed in order to meet the demand. The example in Figure 9.51 shows how the pipe from Figure 9.50 requires the upstream head of nearly 30 mwc after it loses its initial conveyance due to ageing, in order to keep the same supply of 500 m^3/h. Maintaining the initial supply head of 22.5 mwc would otherwise cause a reduction in supply to 400 m^3/h.

9.29.2 Gravity systems

In the case of gravity systems, the entire energy needed for water flow is provided from the elevation difference ΔZ. The pressure variation in the z system is influenced exclusively by the demand variation (see Figure 9.52). Hence:

$$\Delta Z = H_{dyn} + H_{st} = \Delta H + \frac{P_{end}}{\rho g} \qquad 9.73$$

For known pressure at the end of the system, the maximum capacity can be determined from the system curve, as shown in Figure 9.53. The figure shows that the lower demand overnight causes smaller head-loss and therefore the minimum pressure in the system will be higher than during the daytime when the demand and head-losses are higher. In theory, this has implications for the value of the static head that is changing with the variable minimum pressure. *The static head used for design purposes is always fixed based on the minimum pressure that is to be maintained in the system during the maximum consumption hour.* When an area that is to be supplied from a single source starts to grow considerably, demand increases and longer pipe routes can lead to a pressure drop in the network affecting the newly-constructed areas. In theory, this problem can be solved by

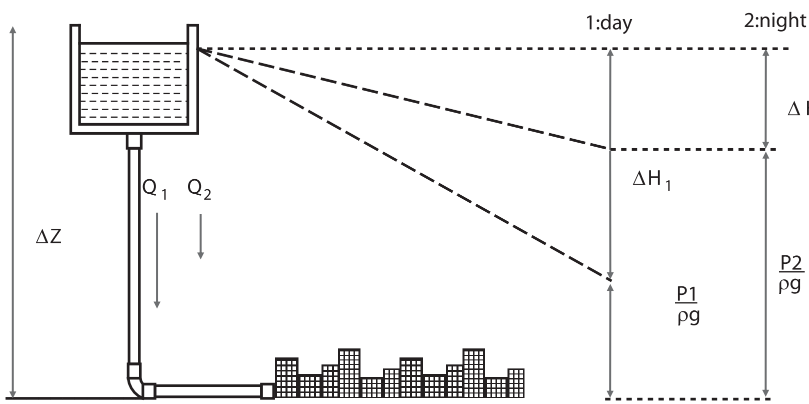


Figure 9.52: **Gravity system: regular supply**

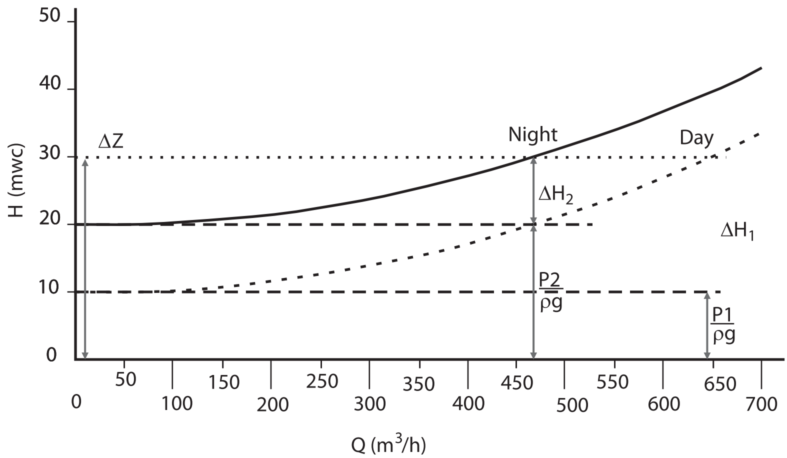


Figure 9.53: **System characteristics: regular operation.**

enlarging the pipes and/or elevating the reservoir (Figures 9.54 and 9.55). Nonetheless, the latter is often impossible due to the fixed position of the source and additional head will probably have to be provided by pumping.

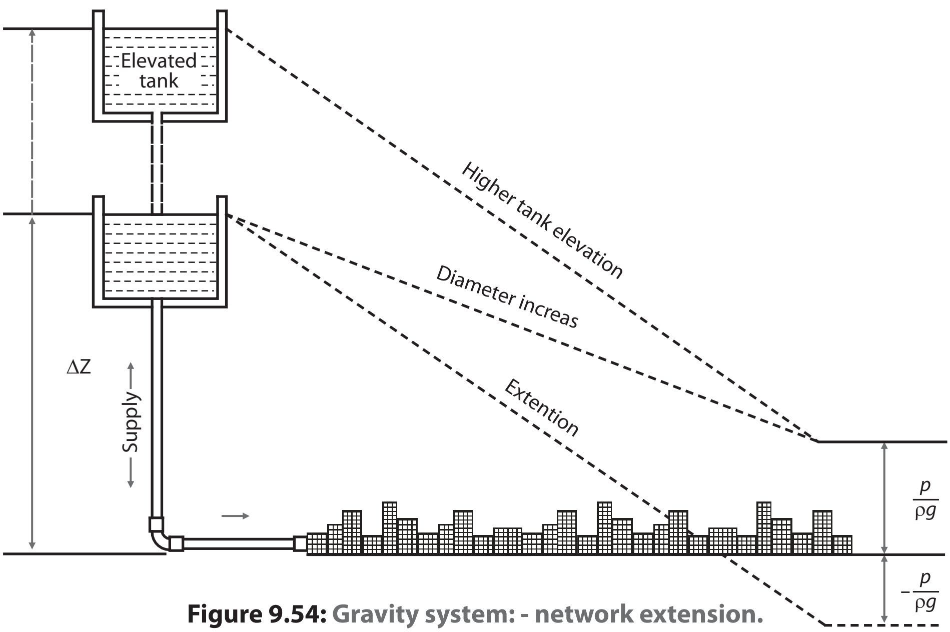


Figure 9.54: **Gravity system: - network extension.**

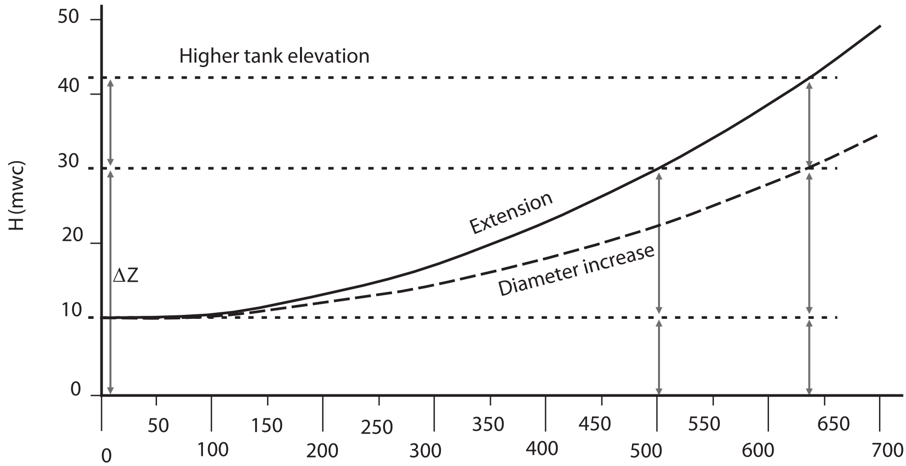


Figure 9.55: **System characteristics: network extension.**

If the system is supplied from more than one side, the storage that is at the higher elevation will normally provide more water i.e. the coverage of the larger part of the distribution area. The intersection between the hydraulic grade lines shows the line of separation between the areas covered by different reservoirs, the so-called *zero-line* (Figure 9.56). Hydraulic conditions in the vicinity of the zero-line are unfavorable:

- the pressure is lower than in other parts of the network,
- the flow velocities are also low, leading to water stagnation and potential water quality problems.

Example 9.24: For the gravity system shown in the Figure 9.57, find the diameter of the pipe L = 2000 m that can deliver a flow of 6000 m^3/h with a pressure of 40 mwc at the entrance of the city. Absolute roughness of the pipe can be assumed at k = 1 mm and the water temperature equals 10°C. What will the increase in capacity of the system be if the pressure at the entrance of the city drops to 30 mwc?

Solution:

At the elevation of Z = 10 msl and the pressure of $p/\rho g$ = 40 mwc, the piezometric head at the entrance of the city becomes H = 50 msl. The surface

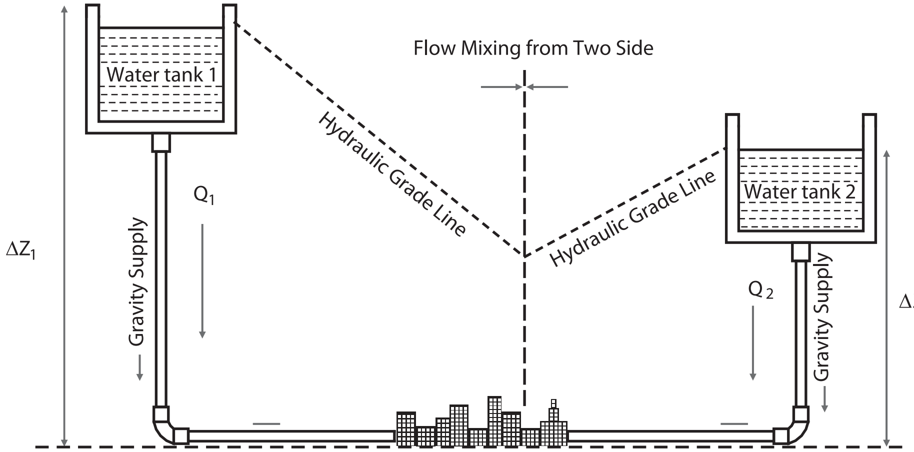


Figure 9.56: **Gravity system: supply from two sides**

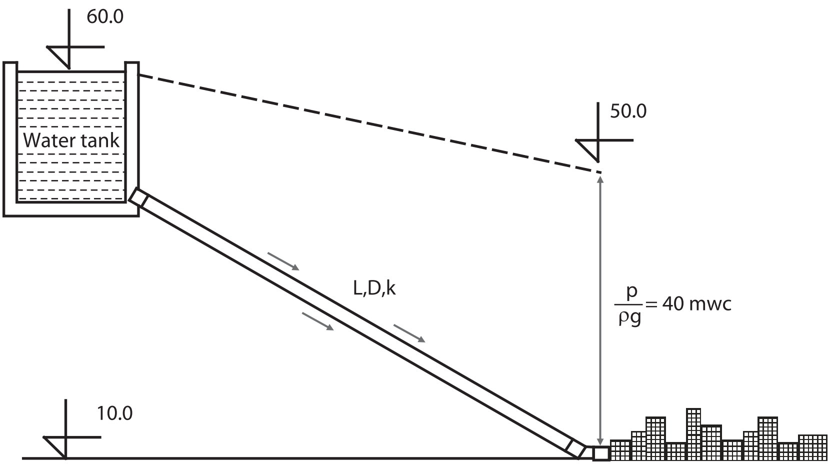


Figure 9.57: **Gravity system (for Example 9.24)**

level/piezometric head of the reservoir is 10 metres higher, and this difference can be utilised as friction loss. The hydraulic gradient then becomes $S = h_f/L = 10/2000 = 0.005$. From the hydraulic tables, for $k = 1$ mm and $T = 10^{\circ}$C:

Discharge flows (l/s), $k = 1$ mm, $S = 0.005$

D (mm)	Q (m^3/h)
900	4772.0
1000	6292.6

If the pressure drops to $p/\rho g = 30$ mwc, the available friction loss increases to hf = 20 mwc and therefore $S = 0.01$. From the same tables, for $D = 1000$ mm, the flow that can be supplied for the given hydraulic gradient increases to $Q = 8911.1\ m^3/h$.

9.29.3 Pumped systems

In pumped systems, the energy needed for water conveyance is obtained from the pump operation. This energy, generated by the pump impeller, is usually expressed as a head of water column (in mwc) and is called the pumping head (or pump lift), hp. It represents the difference between the energy levels at the pump entrance i.e. at the suction pipe and at the pump exit, i.e. at the discharge (or pressure) pipe (Figure 9.58). In the case of a single pump unit, the higher the pumping head hp is, the smaller the pumped flow Q will be. For a combination of Q-h_p values, the power N (W) required to lift the water is calculated as:

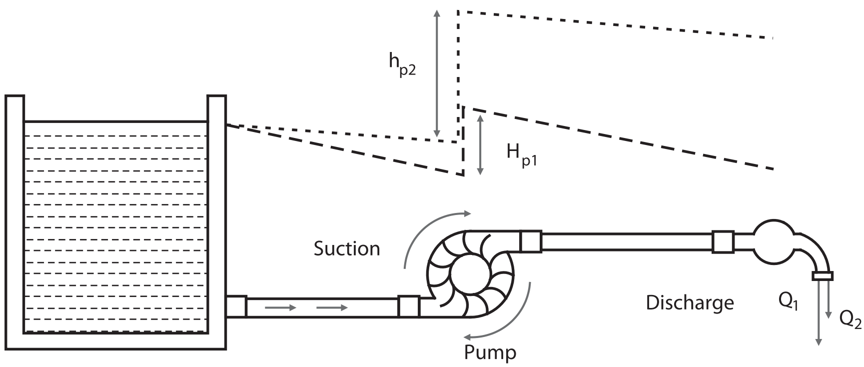


Figure 9.58: Pumping head

$$N = \rho g Q h_p \qquad 9.74$$

Where, Q (m^3/s) is the pump discharge. The power to drive the pump will be higker, due to energy losses in the pump:

$$N_p = \frac{\rho g Q h_p}{\eta_p} \qquad 9.75(a)$$

Where, η_p is the pump efficiency dependant on the pump model and working regime. Finally, the power required for the pump motor will be:

$$N_m = \frac{N_p}{\eta_m} \qquad 9.75(b)$$

Where, η_m indicates the motor efficiency.

The hydraulic performance of pumps is described by the pump characteristics. This diagram shows the relation between the pump discharge and delivered head (Figure 9.59). For centrifugal pumps, a very good approximation of the pump curve is achieved by the following equation:

$$h_p = aQ^2 + bQ + c \qquad 9.76$$

Where factors a, b and c depend on the pump model and flow units. The alternative equation can also be used:

$$h_p = c - aQ^b \qquad 9.77$$

This equation allows for the pump curve definition with a single set of Q-h_p point. These are known as the duty flow (Q_d) and duty head (H_d) and indicate the optimal operational regime of the pump, i.e. the one in which the maximum efficiency η_p will be achieved. As a convention, for exponent $b = 2$:

$$h_{p(Q=0)} = c = \frac{4}{3} H_d ; Q_{(h_p=0)} = 2Q_d \Rightarrow a = \frac{1}{3} \frac{H_d}{Q_d^2} \qquad 9.78$$

Pump manufacturers regularly supply pump characteristics diagrams for each model; a typical format showing a range of impeller diameters and efficiencies η_p, is given in Figure 9.60. Therefore, the pumping head required at the supply side of the system to maintain certain minimum pressure at its end will be:

$$h_p^{req} = H_{dyn} + H_{st} = \Delta H + \frac{P_{\min}}{\rho g} \pm \Delta Z \qquad 9.79$$

This required head is normally higher during daytime than overnight, resulting from higher demand i.e. higher head-losses. Operating the same pump (curve) over 24 hours is therefore unfavourable as it results in the opposite effect: low heads during the daytime and high heads overnight (Figure 9.61). Apart from that, using a single pump in a pumping station is unjustified for reasons of low reliability, high-energy consumption/ low efficiency and problematic maintenance. In practice, several pumps are commonly combined in one pumping station.

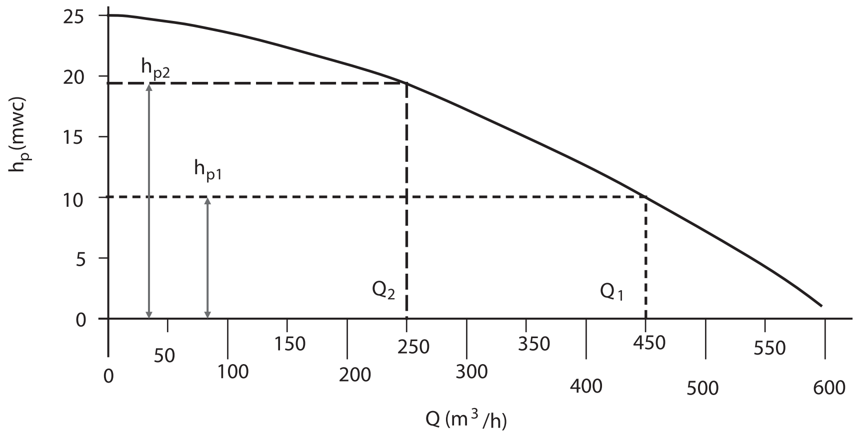


Figure 9.59: Pumping characteristics

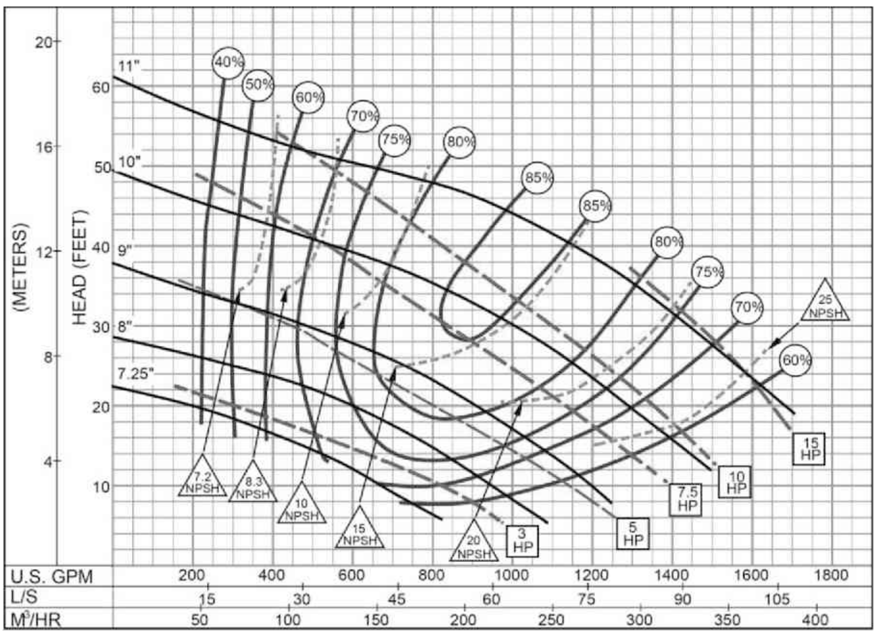


Figure 9.60: Typical pump characteristics curve

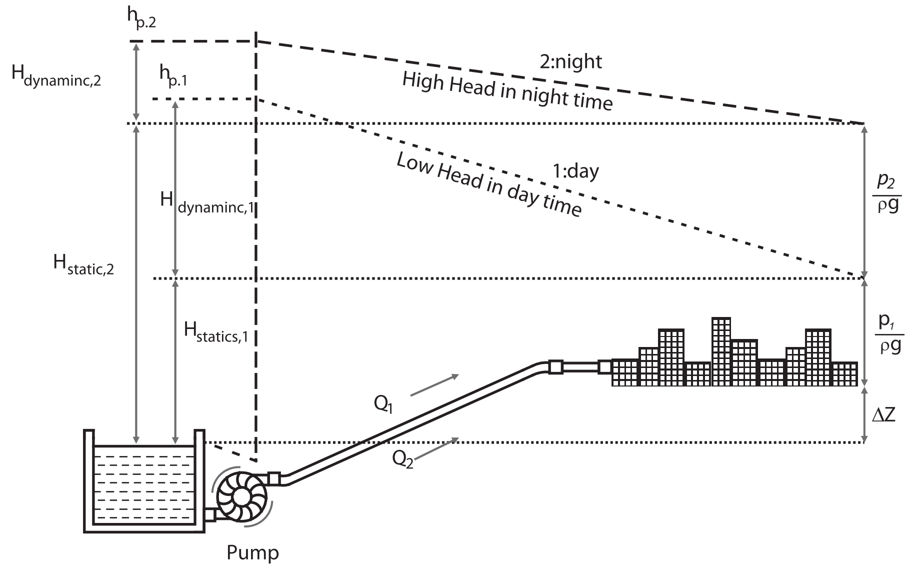


Figure 9.61: **Operation of one pump.**

Flows and pressures that can be delivered by pump operation are determined from the system and pump characteristics. The intersection of these two curves, the *working point*, indicates the required pumping head that provides the flow and the static head, as shown on the graph in Figure 9.62.

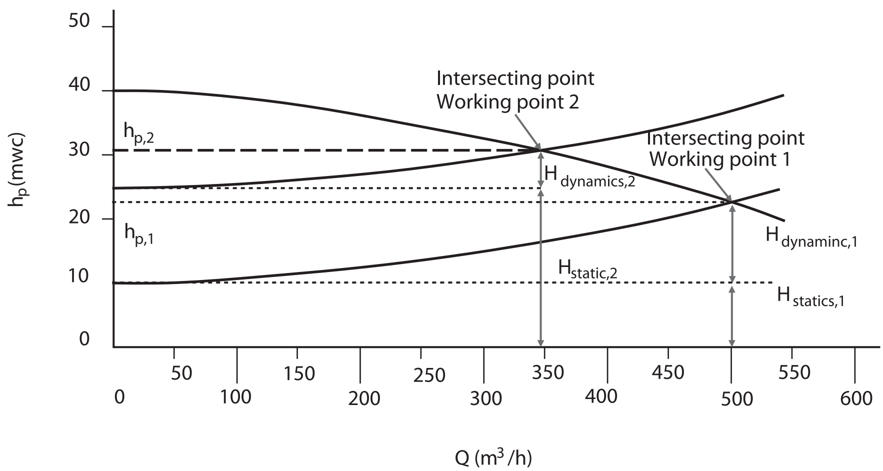


Figure 9.62: **Operation of one pump: day and night time flows**

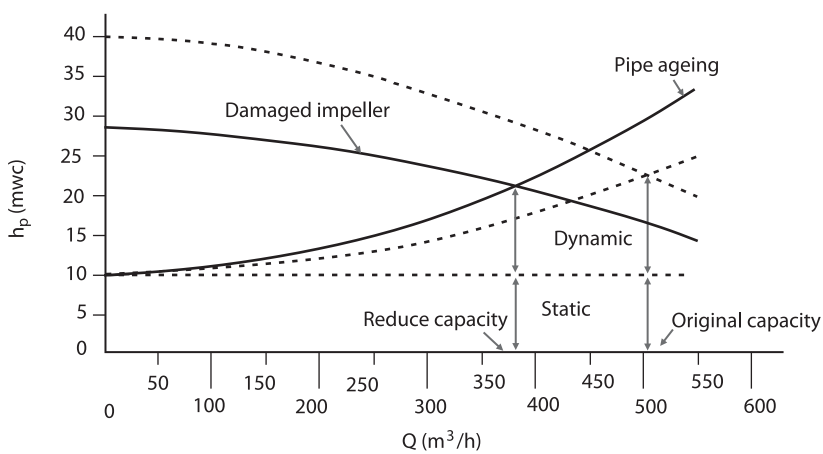


Figure 9.63: Operation of one pump: flow reduction

As in the case of the gravity supply discussed in the previous section, the pressure variation in the distribution area has implications for the value of the static head in this case as well. For design purposes, however, the working point obtained from the system characteristics plotted at the lowest static head (i.e. the lowest pressure required in the system) will be used to determine the maximum pump capacity.

The maximum pumping capacity may vary in time. Ageing of the pump impeller, pipe corrosion, increase of leakage, etc. will cause reduction of the maximum flow that can be delivered by the pump while maintaining the same static head (Figure 9.63).

Decisions on the number and size of pumps in a pumping station are made with the general intention of keeping the pressure variations in the system at the lowest acceptable level in order to minimize the required pumping energy. For this reason, several pumps connected to the same delivery main can be installed in parallel. Their operation will be represented by a composite pump curve, which is obtained by adding the single pump discharges at the same pumping head. Hence, for n pumps:

$$h_p = h_1 = h_2 = h_3 = \ldots\ldots = h_n$$
$$Q_p = Q_1 + Q_2 + Q_3 + \ldots\ldots + Q_n$$

Figure 9.64 shows the operation of two equal pumping units in parallel arrangement. The system should preferably operate at any point along the curve

A_1-A_2-B_1-B_2, between the minimum and maximum flow. The shaded area on the figure indicates excessive pumping, which is unavoidable when fixed speed pumps are used. A properly-selected combination of pump units should reduce this area. This is often achieved by installing pumps of different capacities; the example in Figure 9.65 shows the combination of three equal units with one stronger pump. Introducing variable speed pumps can completely eliminate the excessive head. The flow variation is in this case met by adjusting the impeller rotation, keeping the discharge pressure constant (Figure 9.66). The pump characteristics diagram will consist of a family of curves for various pump frequencies, n (rpm). The relation between the various pumping heads and flows of any two curves is proportional to the frequencies in the following way:

$$\frac{Q_2}{Q_1} = \frac{n_2}{n_1}; \frac{h_{p,2}}{h_{p,1}} = \left(\frac{n_2}{n_1}\right)^2 \qquad 9.80$$

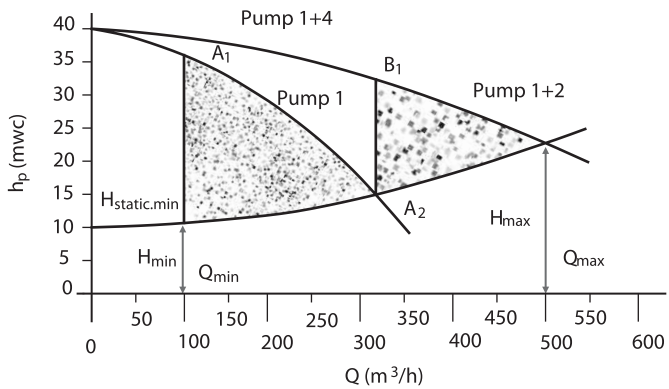


Figure 9.64: **Equal pumps in parallel arrangement.**

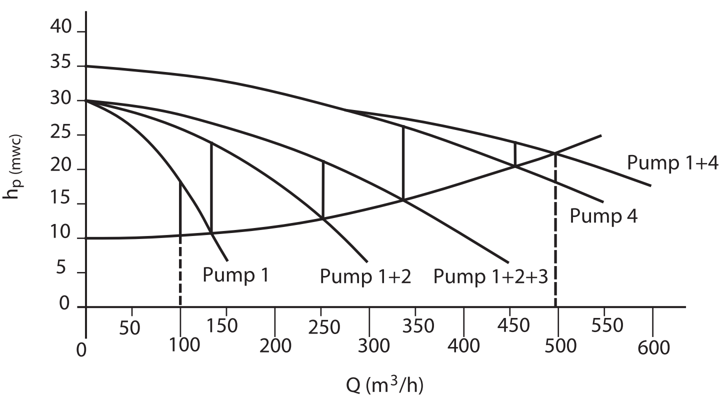


Figure 9.65: **Various pump sizes in a parallel arrangement.**

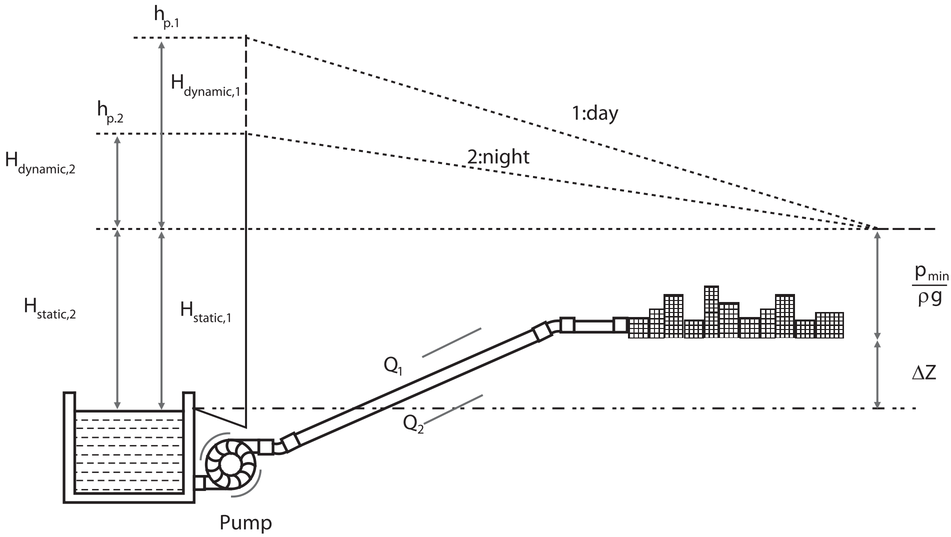


Figure 9.66: **Operation of variable speed pumps.**

Nevertheless, one variable speed unit alone can hardly cover the entire range of flows and several units in parallel will therefore be used. Variable speed pumps can also be combined with fixed speed pumps, controlling only the peak flows (Figure 9.67).

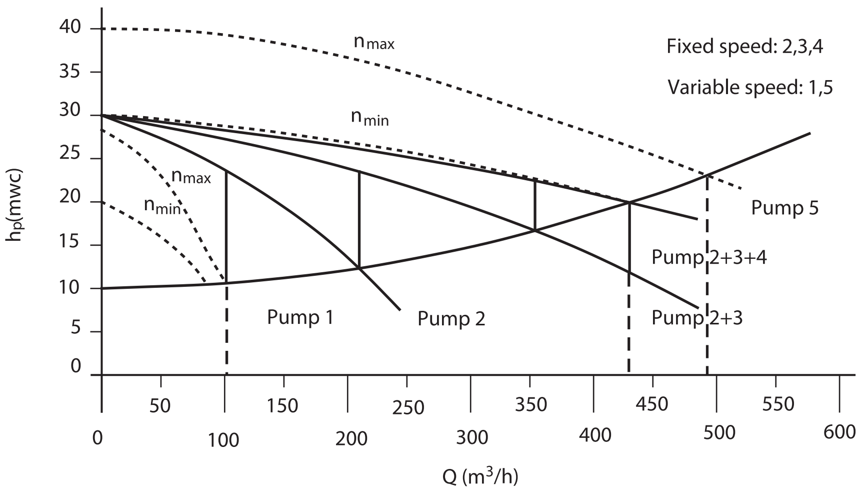


Figure 9.67: **Combined operation of variable and fixed speed pumps.**

In case of large pressure variations in the system, the pumps have to be installed in a serial arrangement. The total head is in this case equal to the sum of heads for each pump. Figure 9.68 shows the curves for two pumps in operation. For *n* equal units:

$$h_p = h_1 = h_2 = h_3 = \ldots\ldots = h_n$$
$$Q_p = Q_1 + Q_2 + Q_3 + \ldots\ldots + Q_n$$

Therefore, pumping from more than one supply point will cause similar problems as with gravity systems in areas where the water from different sources is mixed. However, modifying the pump regimes can shift the zones of minimum pressure, as shown in Figure 9.69.

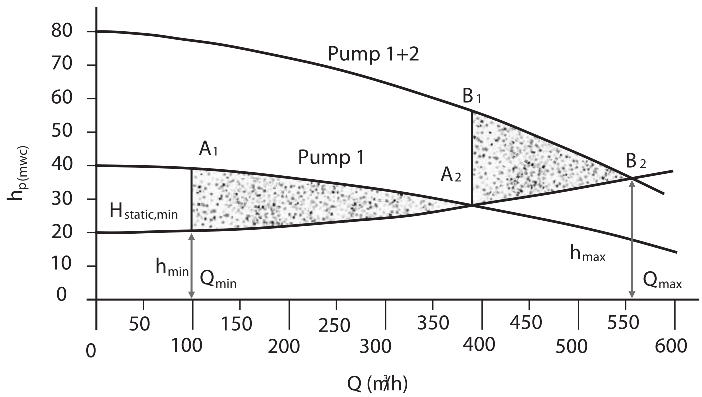


Figure 9.68: Pumps in series.

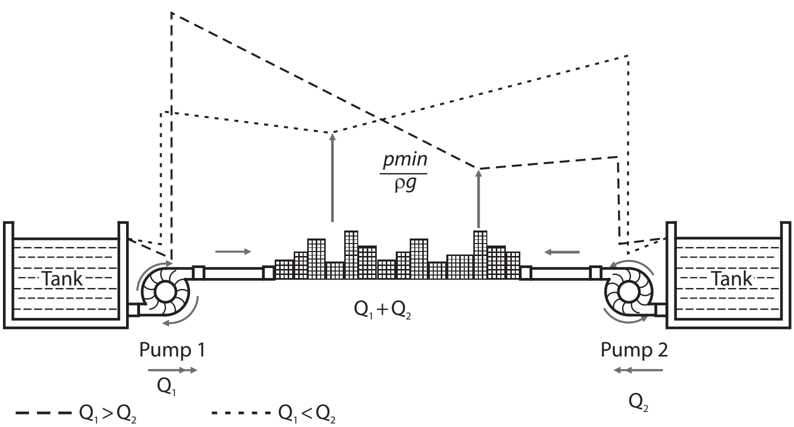


Figure 9.69: Pumping from two sources.

Example 9.25: For the pumped system shown in the following Figure 9.70, determine the required pumping head to deliver flow $Q = 4000\ m^3/h$ through pipe $L = 1200$ m and $D = 800$ mm, while maintaining the pressure of 50 mwc at the entrance of the city. Absolute roughness of the pipe can be assumed at $k = 0.5$ mm and the water temperature equals 10°C. Find the equation of the pump curve assuming the earlier operation happens at maximum pump efficiency. What will be the pressure at the entrance of the city when there is a demand growth of 25%?

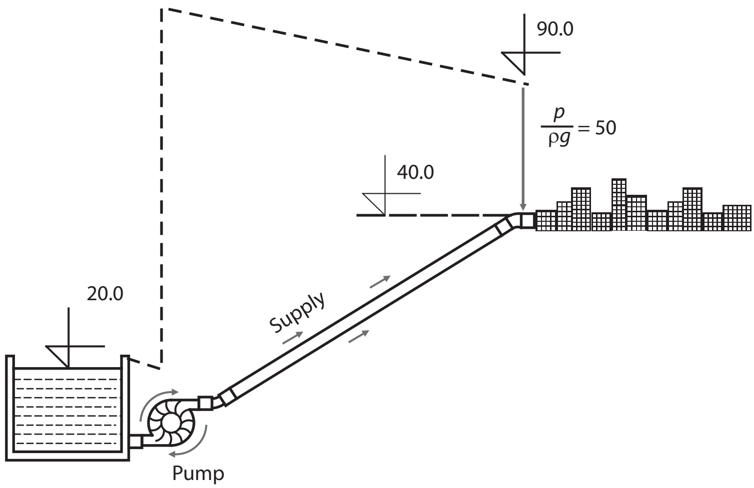


Figure 9.70: Pumped system (for the example 9.25)

Solution:

At the entrance of the city, elevation $Z = 40$ msl. With a required pressure of $p/\rho g = 50$ mwc, the piezometric head becomes $H = 90$ msl. As the surface level/piezometric head of the reservoir is set at 20 metres, the total static head $H_{st} = 90–20 = 70$ mwc. The losses between the reservoir and the pump can be ignored. For the following parameters: $Q = 4000\ m^3/h$, $L = 1200$ m, $D = 800$ mm, $k = 0.5$ mm and $T = 10°C$, the pipe friction loss will be calculat.ed as follows:

$$v = \frac{4Q}{D^2\pi} = \frac{4 \times 4000}{0.8^2 \times 3.14 \times 3600} = 2.21\ m/s$$

For temperature $T = 10°C$, Kinematic viscosity, $\mu = 1.31 \times 10^{-6}\ m^2/s$, the Reynolds number takes the value of:

$$Re = \frac{vD}{\mu} = \frac{2.21 \times 0.8}{1.31 \times 10^{-6}} = 1.4 \times 10^6$$

And the function factor is λ from Barr's Equation equals:

$$\lambda = 0.25/\log^2\left[\frac{5.1286}{Re^{0.89}} + \frac{k}{3.7D}\right]$$

$$\lambda = 0.25/\log^2\left[\frac{5.1286}{\left(1.4 \times 10^6\right)^{0.89}} + \frac{0.5}{3.7 \times 800}\right] \approx 0.018$$

The friction loss from the Darcy-Weisbach equation can be determined:

$$h_f = \frac{\lambda L}{12.1 D^5} Q^2 = \frac{0.018 \times 1200}{12.1 \times 0.8^5} 1.11^2 \approx 7mwc$$

The total required pumping head is therefore $h_p = H_{st} + H_{dys}$ = 70+7 =77 mwc. Given the maximum pumping efficiency this is also the duty head at the flow of 4000 m^3/h and the equation becomes:

$$a = \frac{1}{3}\frac{H_d}{Q_d^2} = \frac{77}{3 \times 4000^2} = 1.604 \times 10^{-6} \quad \text{And}$$

$$c = \frac{4}{3} H_d = \frac{4}{3} \times 77 = 102.67$$

Hence the pumping curve can be appropriate with the following equation (exponent b = 2):

$$h_p = c - aQ^b \approx 103 - 1.6 \times 10^{-6} Q^2$$

If demand grows by 25% i.e. to 5000 m3/h the pumping head that can be provided will be:

$$h_p = 103 - 1.6 \times 10^{-6} 5000^2 \approx 63mwc$$

The friction loss calculation in the same way as above is going to increase to approximately 11 wmc, leading to residual pressure at the entrance to the city of $p/\rho g$=20+60-11-40≈32 mwc

9.29.4 Combined systems

Consumers in combined systems are partly supplied by gravity and partly by pumping. Three basic concepts can be distinguished:

- The water is pumped from a reservoir into the distribution area (tank-pump- network).
- The water is pumped to a reservoir and thereafter supplied by gravity (pump-tank-network).
- Pump and reservoir are at the opposite sides of the distribution area (pump-network-tank).

Tank-pump-network

This scheme is suitable for mild terrains where a favorable location for the reservoir is difficult to find, either due to insufficiently-high elevations or because of a large distance from the distribution area. Essentially this is the same concept as that of direct pumping, except that the required pumping head can be reduced on account of the elevation difference in the system. Hence:

$$h_p + \Delta Z = H_{dyn} + H_{st} = \Delta H + \frac{P_{end}}{\rho g} \qquad 9.81$$

Both the dynamic and static head are supplied partly by gravity and partly by pumping, depending on the elevation difference and the pressure at the end of the system (Figures 9.71 and 9.72).

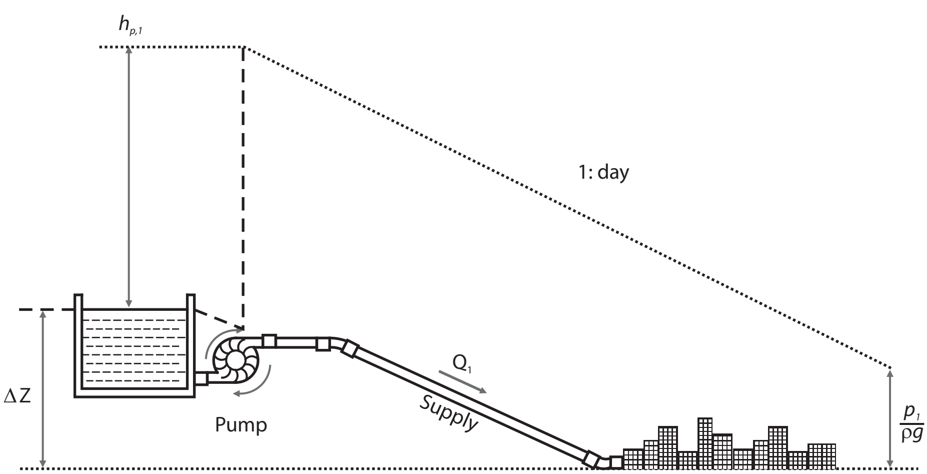


Figure 9.71: Combined supply by gravity and pumping: daytime flows.

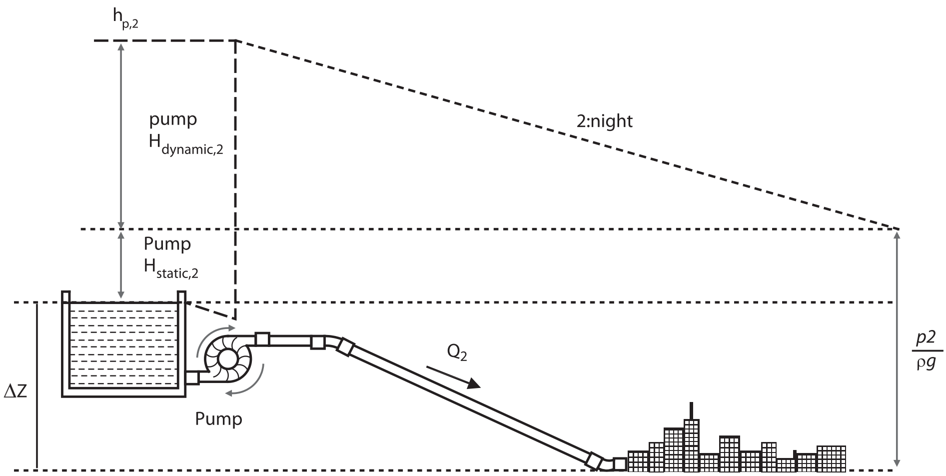


Figure 9.72: **Combined supply by gravity and pumping: night time flows**

Pumping stations need not necessarily to be located at the supply point. When positioned within the system, they are commonly called booster stations. Such a layout is attractive if high pressures are to be avoided (Figure 9.73).

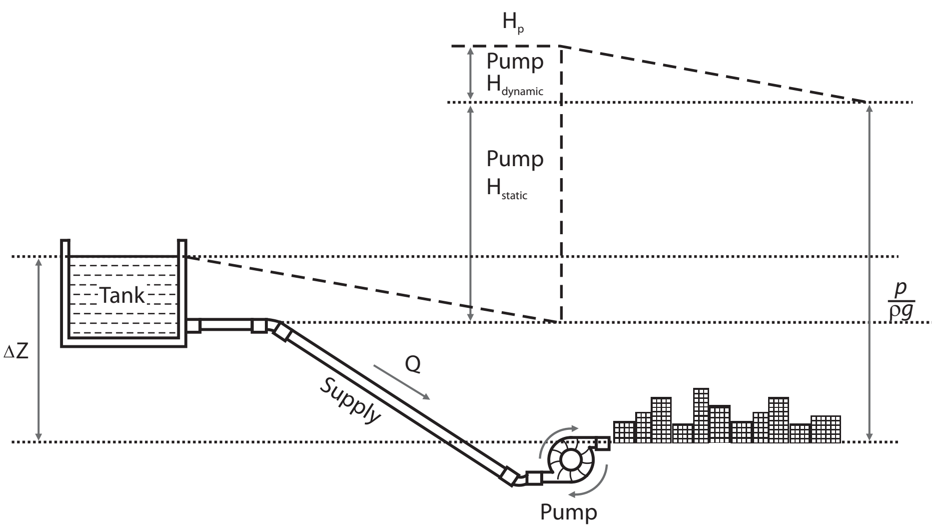


Figure 9.73: **Booster stations.**

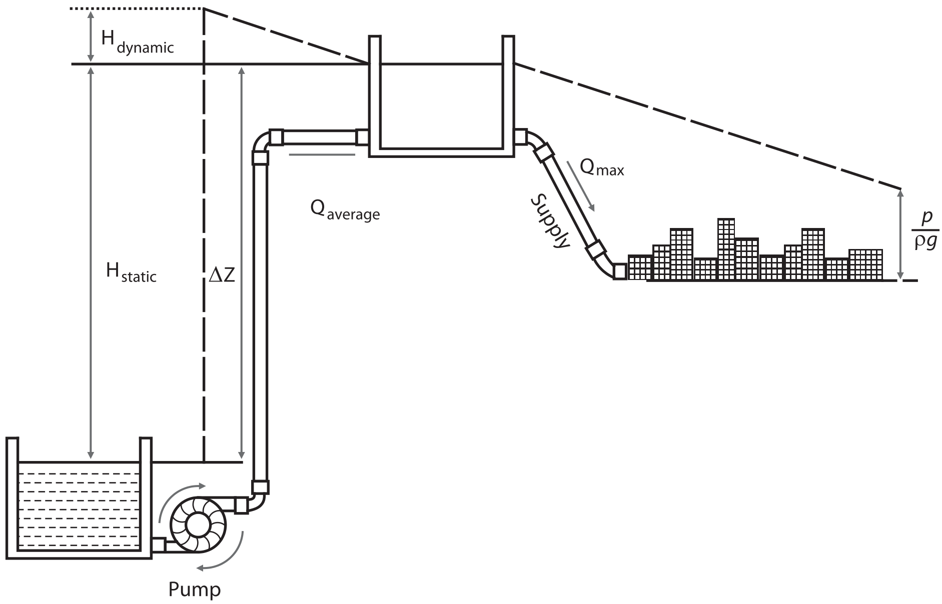


Figure 9.74: Gravity supply supported by pumping

Pump-tank-network

This scheme is typical for hilly terrains. When pumps deliver water to the reservoir, the static head will only comprise the elevation head ΔZ, which equals the elevation difference between the surface levels in the two reservoirs (Figure 9.74). Thus:

$$h_p = H_{dyn} + H_{st} = \Delta H + \Delta Z \qquad 9.82$$

The advantages of this scheme are:

- stable operation of the pumping station,
- a buffer supply capacity in case of pump failure.

A similar hydraulic pattern is valid if water towers are put into the system (Figure 9.75). However, their predominant role is to maintain stable

operation of the pumps, rather than to provide buffer or large balancing volumes. While supplying tanks, the pumps often operate automatically, based on monitoring of water levels in the reservoirs. Pump throttling may be required in order to adjust the flow. The effects on the system characteristics are shown in Figure 9.76.

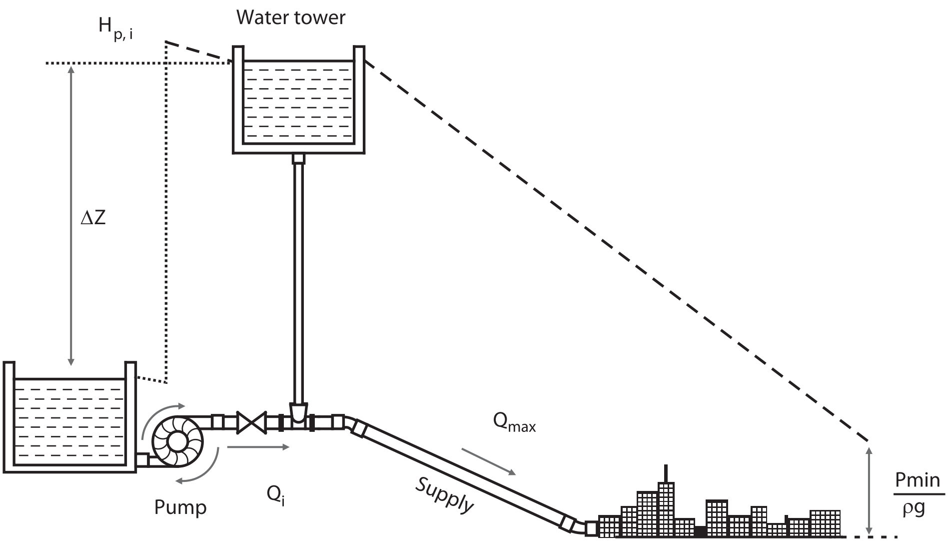


Figure 9.75: Pump operation in combination with water tower

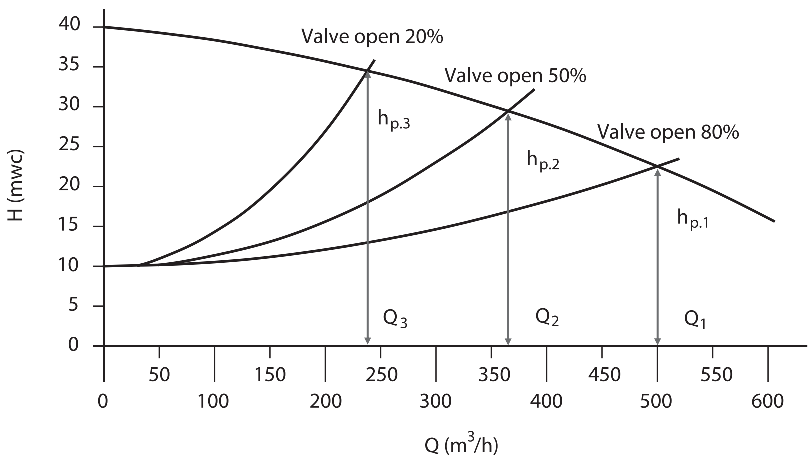


Figure 9.76: Effects of pump throttling on system characteristics

Pump-network-tank

This scheme is predominantly applied for distribution networks located in valleys. During the maximum supply conditions, both the pump and reservoir will cover part of the distribution area (Figure 9.77). If the only source of supply is close to the pumping station, that one will also be used to refill the volume of the tank. This is usually done overnight when the demand in the area is low (Figure 9.78).

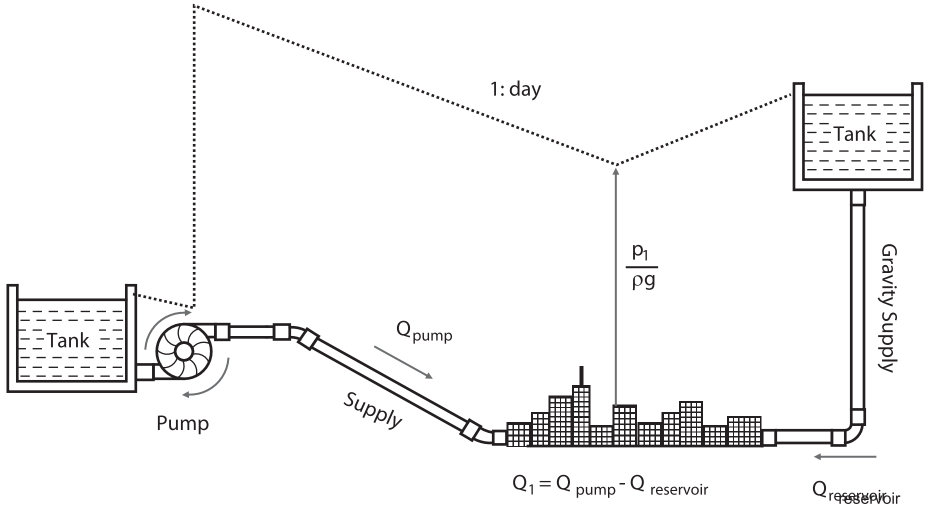


Figure 9.77: **Counter tank: daytime flows**

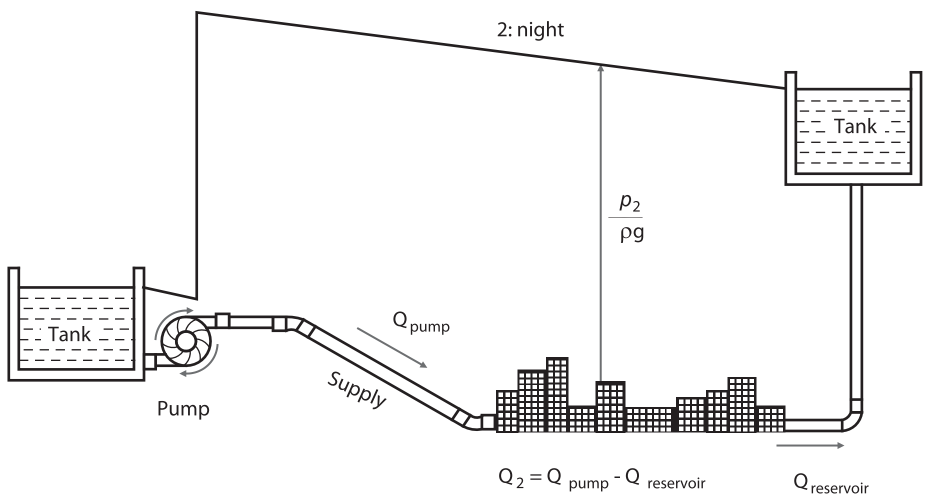


Figure 9.78: **Counter tank: night time flows**

Counter tank operating in this way functions as a kind of counter tank to the one at the source. Depending on its size and elevation, it can balance the demand variation in the system, partly or completely. In the second case, the pumping station operates at constant (average) capacity ($Q_{pump} = Q_{average}$).

9.30 Conclusion

In this chapter, the distribution of water has been discussed in details following different available standard codes in Water Supply Engineering. In the next chapter, the plumbing system of water supply will be discussed.

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