Mechanical Engineering Department

Thermo-fluid System Design

(MEng 5313)

Lecture 3

Liquid Piping Systems

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- Liquid Piping Systems
- Minor Losses: Fittings and Valves in Liquid Piping Systems
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- Fluid Machines (Pumps) and Pump–Pipe Matching
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Liquid Piping Systems

- Piping systems are used to transport diverse liquids for a variety of different applications.
- These applications may range from water service for buildings to complex two-phase flow systems in industrial plants. The design of these systems requires consideration of several groups of specialties and accessories that will be needed for a functional system.
- This chapter will focus on fittings and accessories, pipe materials, fluid machines, and design considerations necessary for the successful design of practical piping systems for various applications.

Minor Losses: Fittings and Valves in Liquid Piping Systems

Fittings

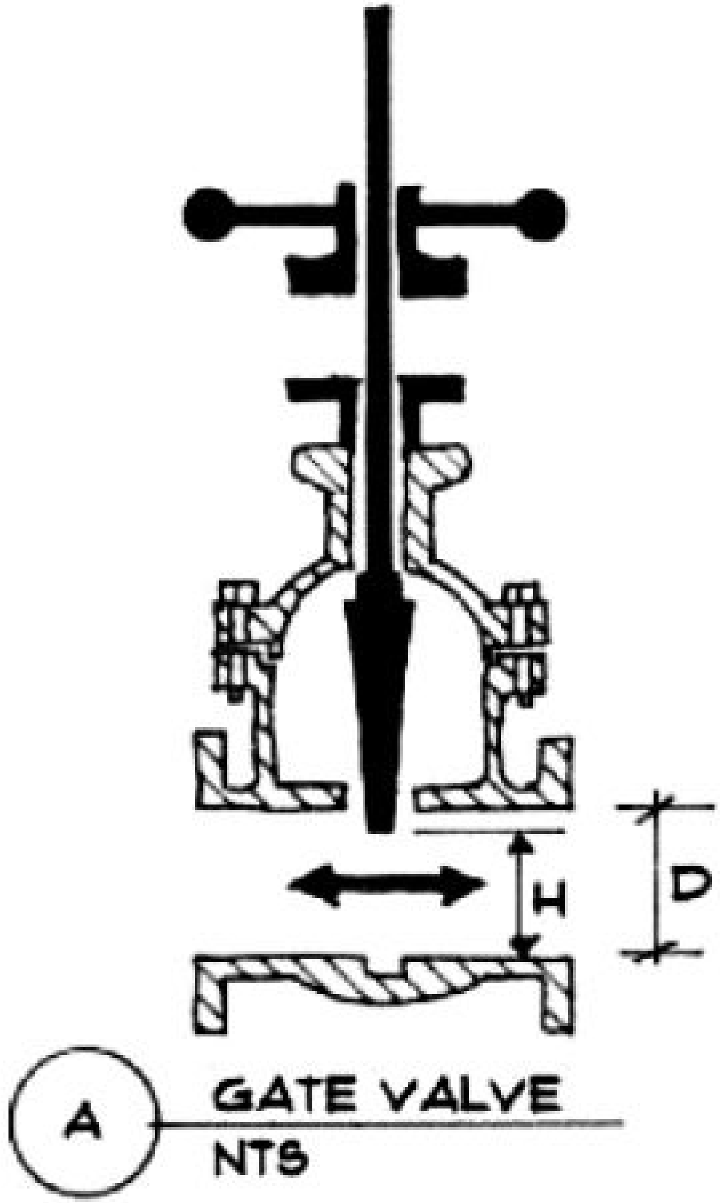
- Fittings are used to extend pipe lengths, expand the pipe network, or perform a selected function.
- Examples of fittings specific to liquid piping systems are plugs, unions, wyes, valves, tees, caps, ferrules, elbows, nipples, reducers, sleeves, couplings, adapters, fasteners, compression fittings, and bulkhead fittings, to name a few.
- All these add resistance to fluid flow. Tabulated K values (loss coefficients) are available for these and other fittings

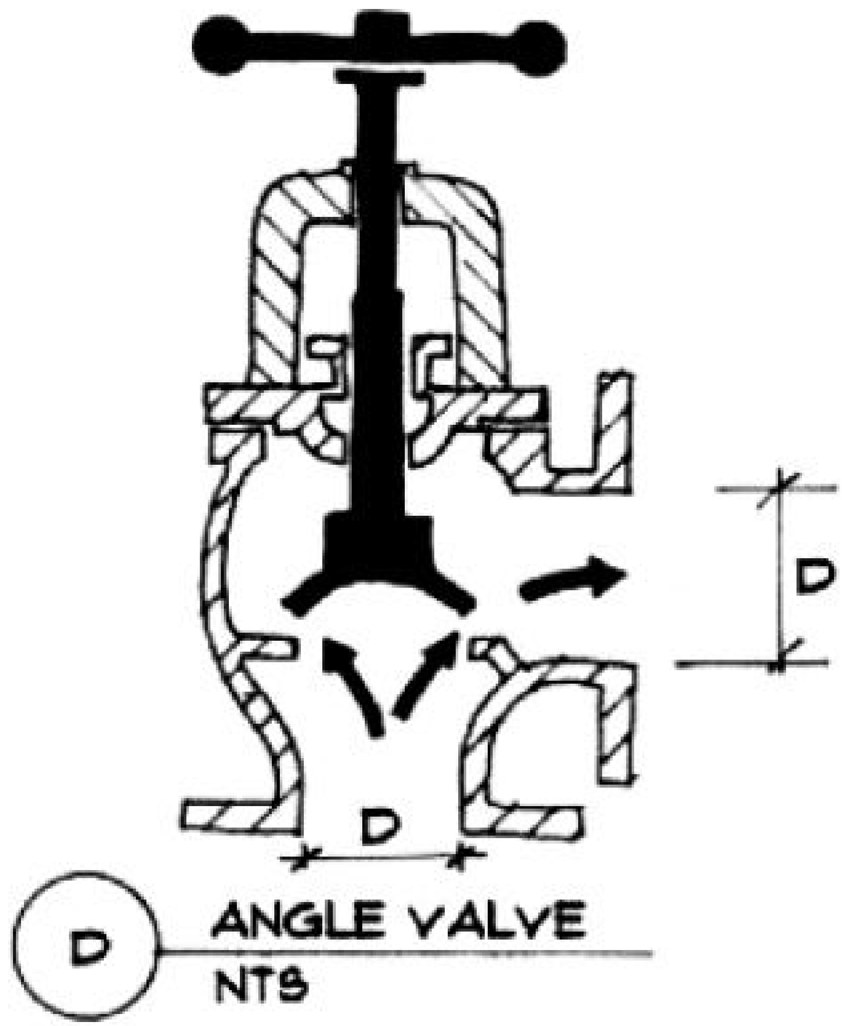
Minor Losses: Fittings and Valves in Liquid Piping Systems

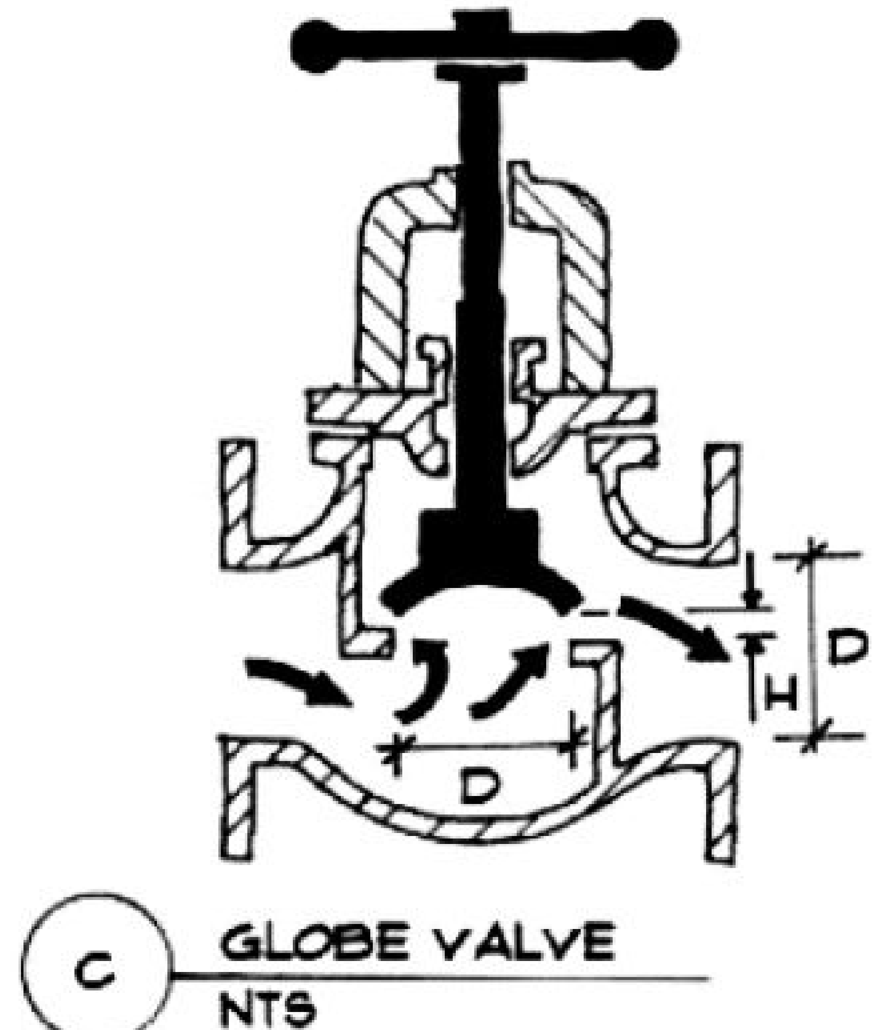
Valves

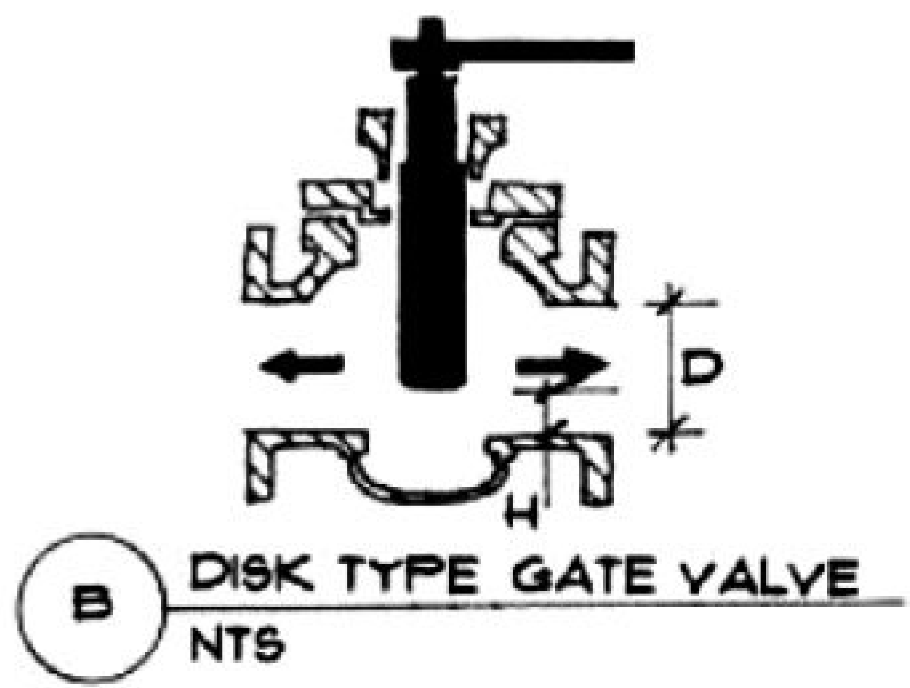
- Valves are used to control the flow rate of fluid in piping systems.
- For valves, lower K values occur when they are fully open; thus, frictional losses will be low. The K values will increase as the valve is closed. A similar trend will apply to L_{equiv} values.
- There are many types of valves . Some of these valves and their drawing symbols are shown in Figure below.
- Manufacturer's catalogs should be consulted to find other valves and/or pipe fittings and their K or L_{equiv} values

Minor Losses: Fittings and Valves in Liquid Piping Systems









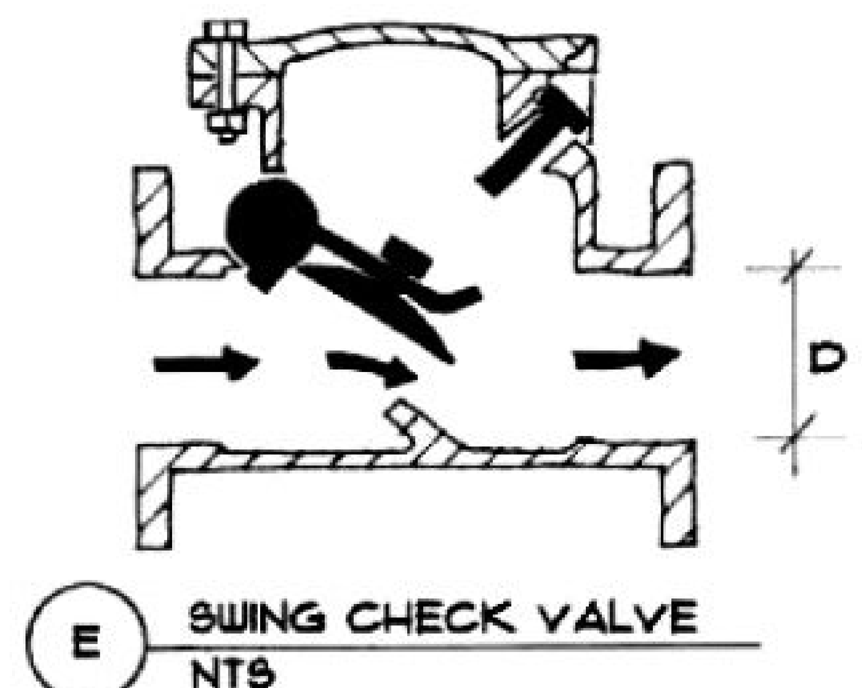


Figure. Some typical industrial valves

Sizing Liquid Piping Systems

General Design Considerations

- Sizing liquid piping follows a similar procedure to that outlined in Example 1 of Chapter 2. Consider the following additional points when sizing and designing piping systems.

a) *Pipe Materials*: Is the fluid corrosive? Are there particulates in the fluid (e.g., oil sand liquid slurries) that will erode the pipe? Is the fluid temperature high?

b) *Pipe Thickness*: Is higher pipe strength required for high stress (high pressure) applications?

c) *Plastic Piping*: Can plastic piping be used instead of metal pipes? Plastic pipes are lightweight, easy to join, and corrosion resistant.

d) *Flow Velocities:* Different services and applications have different pipe velocity requirements and ranges.

Sizing Liquid Piping Systems

Examples of Plastic Piping

a) *Polyvinyl chloride (PVC)*: For applications in pipes for building cold water, drains, and condensate piping. **Not recommended** for hot water piping.

b) *Chlorinated polyvinyl chloride (CPVC):* Similar to PVC. This plastic material is able to withstand temperatures of up to 140◦F.

c) *Reinforced thermosetting resin plastic (RTRP):* Recommended for hot water piping systems with temperatures on the order of 200◦F.

d) *Cross-linked polyethylene (PEX):* For applications in hydronic-radiant heating systems, domestic water piping, natural gas and offshore oil applications, chemical transportation, and transportation of sewage and slurries. Recently, it has become a viable alternative to PVC, CPVC, and copper tubing for use as residential water pipes (particularly in Canada).

e) *Acrylonitrile butadiene styrene (ABS):* Used in building plumbing systems as drain, vent, and sewage piping. It may also be used to transport potable (drinking) water, chemicals, or chilled water.

Sizing Liquid Piping Systems

Table 3.1 Typical average velocities for selected pipe flows[a]

Fluid	Application	Velocity (fps)	Velocity (m/s)
Steam	Superheated process steam	148–328	45–100
	Auxiliary heat steam	98–246	30–75
	Saturated and low-pressure steam	98–164	30–50
Water	Centrifugal pump suction lines	3–4.9 (must be < 4.9 fps)[b]	0.9–1.5 (must be <1.5 m/s)[b]
	Power plant feedwater	7.9–15	2.4–4.6
	General building service	3.9–10.2	1.2–3.1
	Potable water	Up to 6.9 (must be < 9.8 fps)[b]	Up to 2.1 (must be <3.0 m/s)[b]

[a] Adapted from the US Department of the Army, TM 5-810-15, Central Boiler Plants, August 1995.
[b] Adapted from *2005 Fundamentals*, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Atlanta, GA, 2005, pp. 36–11.

Sizing Liquid Piping Systems

Pipe Data for Building Water Systems

- As seen in Chapter 2, pipe and duct sizing can be laborious, requiring many iterations of complex correlation equations.
- In practice for building water systems, charts and tables are used to simplify the process of liquid pipe sizing. Of importance is pipe material selection, determination of pipe diameter, and installation specifications.
- While the focus is on building water systems, the procedures outlined here will apply directly to other types of piping systems such as wastewater/sludge systems, food processing systems, and chemical liquid systems.

Sizing Liquid Piping Systems

A. *Pipe Materials for Building Water Systems*: Type L copper tubing is widely used. Schedule 40 steel is used in steam heating systems. Schedule 80 steel, which is thicker than Schedule 40 steel, is used in high-pressure steam lines. PEX may also be used in cold or hot water systems in lieu of copper. It is **not recommended** for high-pressure steam lines. Table 3.2 shows some data for copper and steel pipes.

B. *Pipe Sizing Considerations—Determination of Pipe Diameters*: Pipes in closed-loop building systems should be designed to limit friction losses to **3 ft of water per 100 ft of pipe**. This will reduce the final pump size and cost.

- In addition, pipe velocities should be **less than 10 fps** or 3 m/s to reduce noise and pipe material erosion. Typically, volume flow rates in the pipes are known, which makes determination of the pipe diameter easier.

Sizing Liquid Piping Systems

Table 3.2 Pipe data for copper and steel

Material	Diameter (in.) Nominal	Inner	Outer	Weight per Linear Foot of Pipe and Water (lb)	Gallons of Water per Linear Foot
Copper					
Type L	3/8	0.430	0.500	0.26	0.008
Type L	1/2	0.545	0.625	0.39	0.012
Type L	3/4	0.785	0.875	0.67	0.025
Type L	1	1.025	1.125	1.01	0.043
Type L	1 1/4	1.265	1.375	1.43	0.065
Type L	1 1/2	1.505	1.625	1.91	0.093
Type L	2	1.985	2.125	3.09	0.161
Type L	2 1/2	2.465	2.625	4.55	0.248
Type L	3	2.945	3.125	6.29	0.354
Type L	3 1/2	3.425	3.625	8.29	0.479
Type L	4	3.905	4.125	10.58	0.622

Steel					
Schedule 40	1/4	0.364	0.540	0.475	0.005
Schedule 40	1/2	0.622	0.840	0.992	0.016
Schedule 40	3/4	0.824	1.050	1.372	0.028
Schedule 40	1	1.049	1.315	2.055	0.045
Schedule 40	1 1/4	1.380	1.660	2.929	0.077
Schedule 40	1 1/2	1.610	1.900	3.602	0.106
Schedule 40	2	2.067	2.375	5.114	0.174
Schedule 40	2 1/2	2.469	2.875	7.873	0.248
Schedule 40	3	3.068	3.500	10.781	0.383
Schedule 40	3 1/2	3.548	4.000	13.397	0.513
Schedule 40	4	4.026	4.500	16.316	0.660
Schedule 80	1/2	0.546	0.840	1.189	0.012
Schedule 80	3/4	0.742	1.050	1.686	0.026
Schedule 80	1	0.957	1.315	2.483	0.037
Schedule 80	1 1/4	1.278	1.660	3.551	0.067
Schedule 80	1 1/2	1.500	1.900	4.396	0.092
Schedule 80	2	1.939	2.375	6.302	0.154
Schedule 80	2 1/2	2.323	2.875	9.491	0.220
Schedule 80	3	2.900	3.500	13.122	0.344
Schedule 80	3 1/2	3.364	4.000	16.225	0.458
Schedule 80	4	3.826	4.500	19.953	0.597

Sizing Liquid Piping Systems

Practical Note 1. Higher Pipe Friction Losses and Velocities

- The design engineer should bear in mind that the aforementioned points are guidelines. Flexibility exists in order to optimize the performance of a given system design. For example, higher friction losses may be acceptable for shorter run of pipes or for cases where smaller pipe sizes are mandatory by the client or application. Higher pipe velocities may be required for fluid systems that transport solid sediments. This ensures that the sediments do not clog the pipes by sticking to the pipe wall.
- However, for small diameter pipes, erosion becomes a concern when large velocities are used. For example, in stainless steel tubes, erosion becomes a concern when velocities are larger than 15 fps (consult Table A.13 for additional data).
- All deviations from the established guidelines should be justified by the design engineer.

Sizing Liquid Piping Systems

Table A.13 Erosion limits: maximum design fluid velocities for water flow in small tubes

Low carbon steel	10 ft/s
Stainless steel	15 ft/s
Aluminum	6 ft/s
Copper	6 ft/s
90–10 Cupronickel	10 ft/s
70–30 Cupronickel	15 ft/s
Titanium	50 ft/s
For other liquids	$V_{\text{liq,max}} = V_{\text{water,max}} \left[\frac{\rho_{\text{water}}}{\rho_{\text{liq}}} \right]$
For gases and dry vapors (ft/s), where M = molecular weight	$V_{\text{gas,max}} = \sqrt{\frac{1800}{PM}}$

Source: Adapted from Wolverine Tube Inc., *Wolverine Tube Heat Transfer Data Book*, Wolverine Tube Inc., Huntsville, AL, 2009, p. 48.

Sizing Liquid Piping Systems

- After selection of the pipe friction loss and velocity, an appropriate chart may be used to determine the pipe diameter. One such chart for plastic piping systems is shown in Figure below.

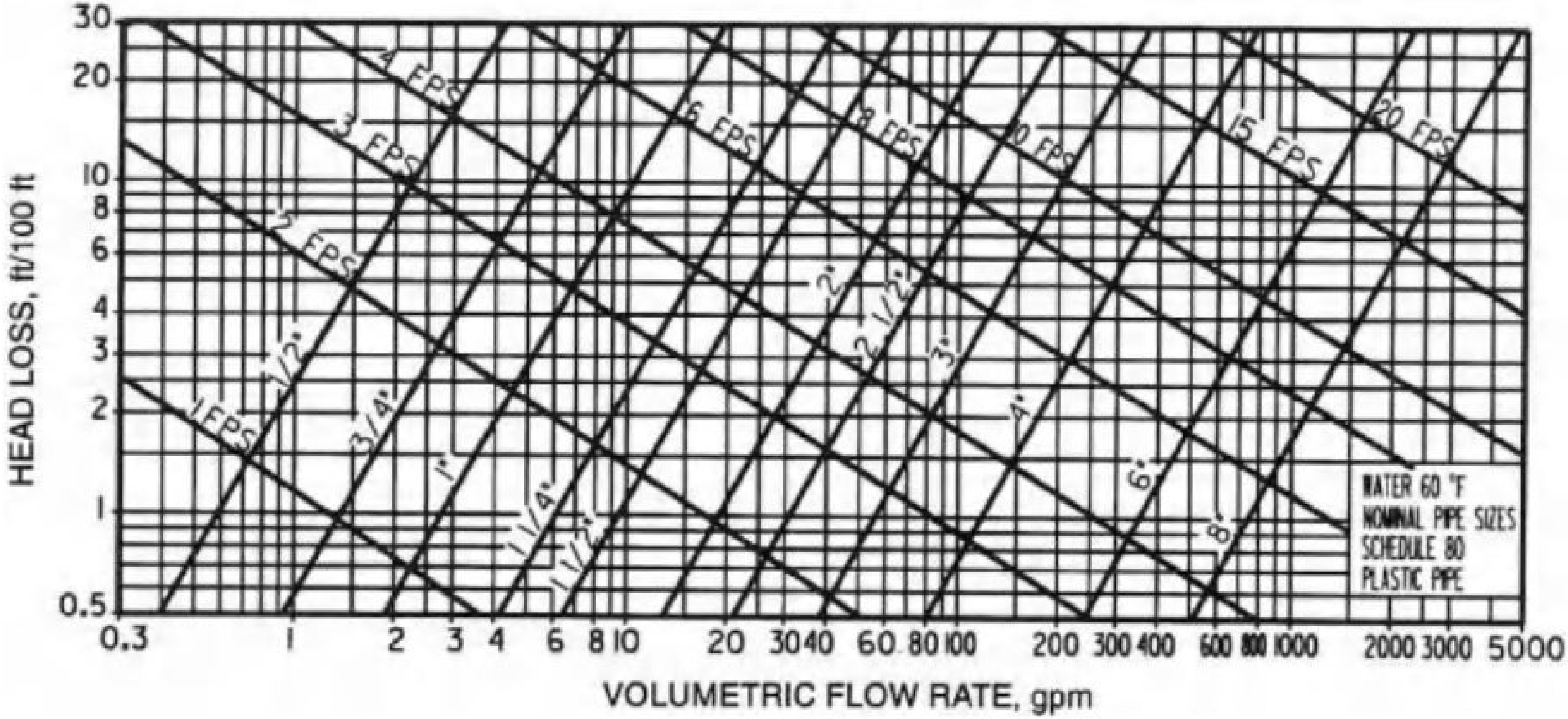


Figure. Plastic pipe (Schedule 80) friction loss chart

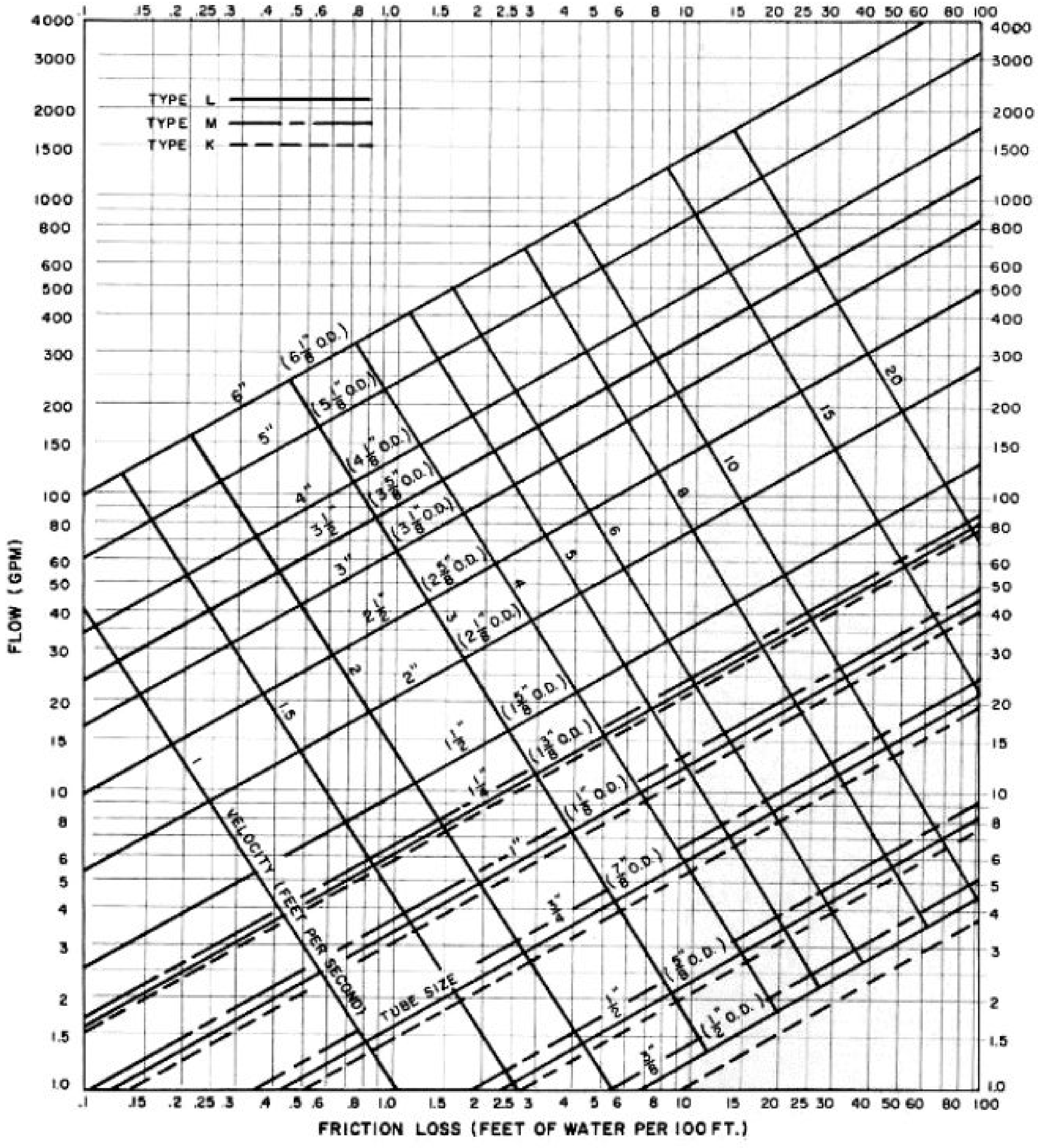


Figure A.3
Copper tubing friction loss (open and closed piping systems)

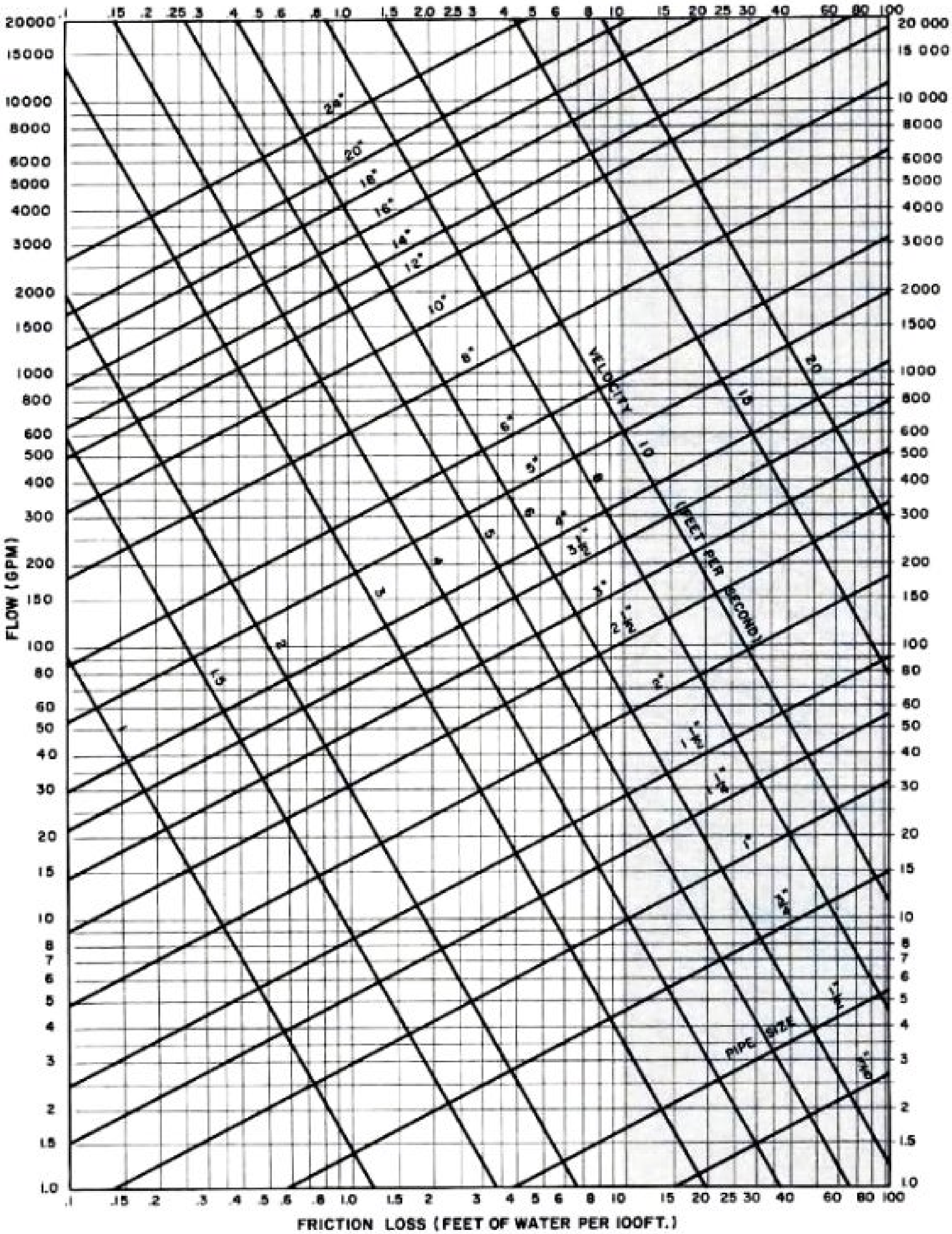


Figure A.4
Commercial steel pipe (Schedule 40) friction loss for open piping systems

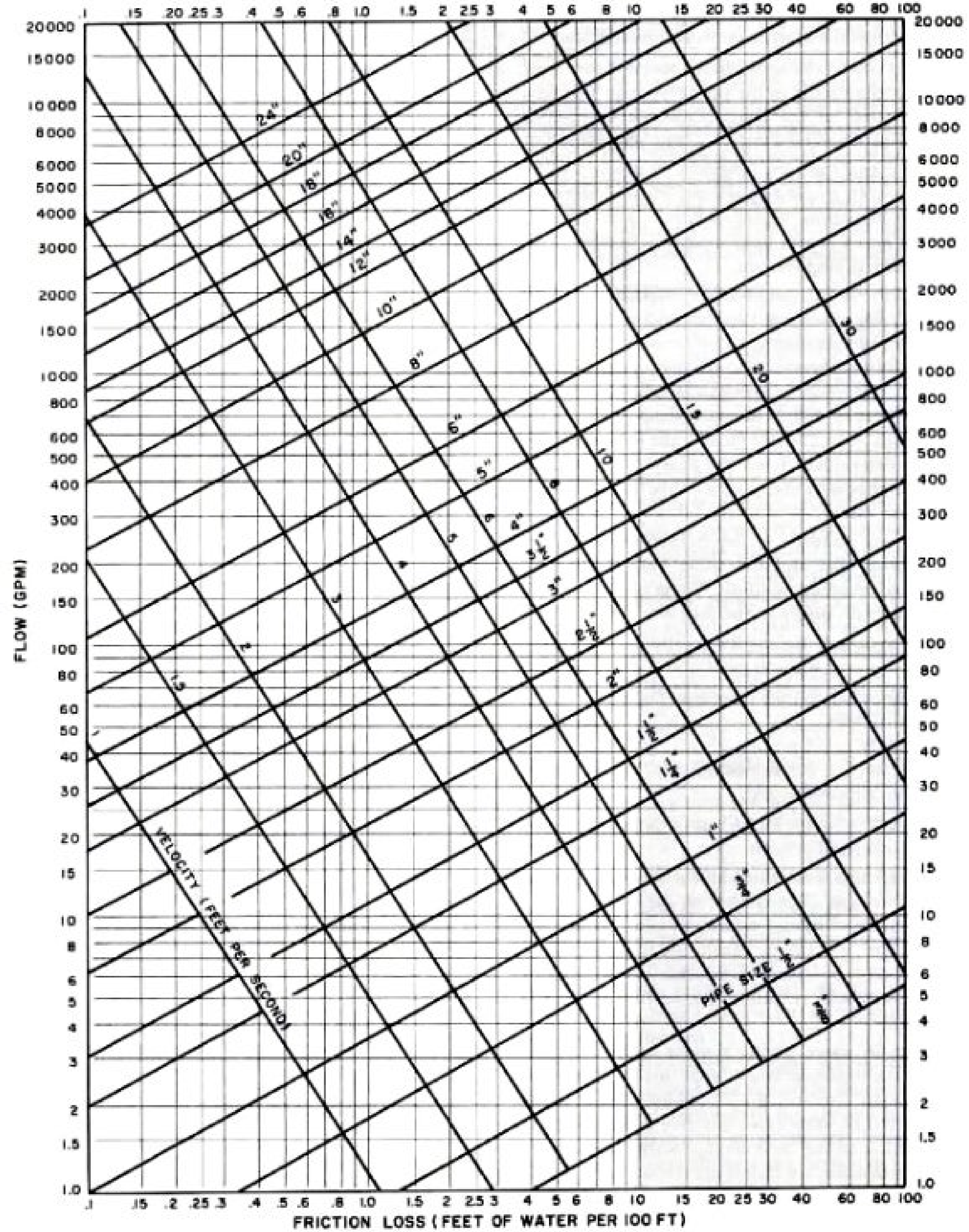


Figure A.4
Commercial steel pipe (Schedule 40) friction loss for closed piping systems

Pipe Installation

- Pipes are typically hung between the slab and the dropped (finished) ceiling and are supported by **pipe hangers**. The hangers are fastened around the pipe and their support rods are anchored to the ceiling. Figure below shows pipes supported on hangers. The weights per foot of piping filled with water are used to determine the spacing of the pipe hangers and the sizing of the support rods.

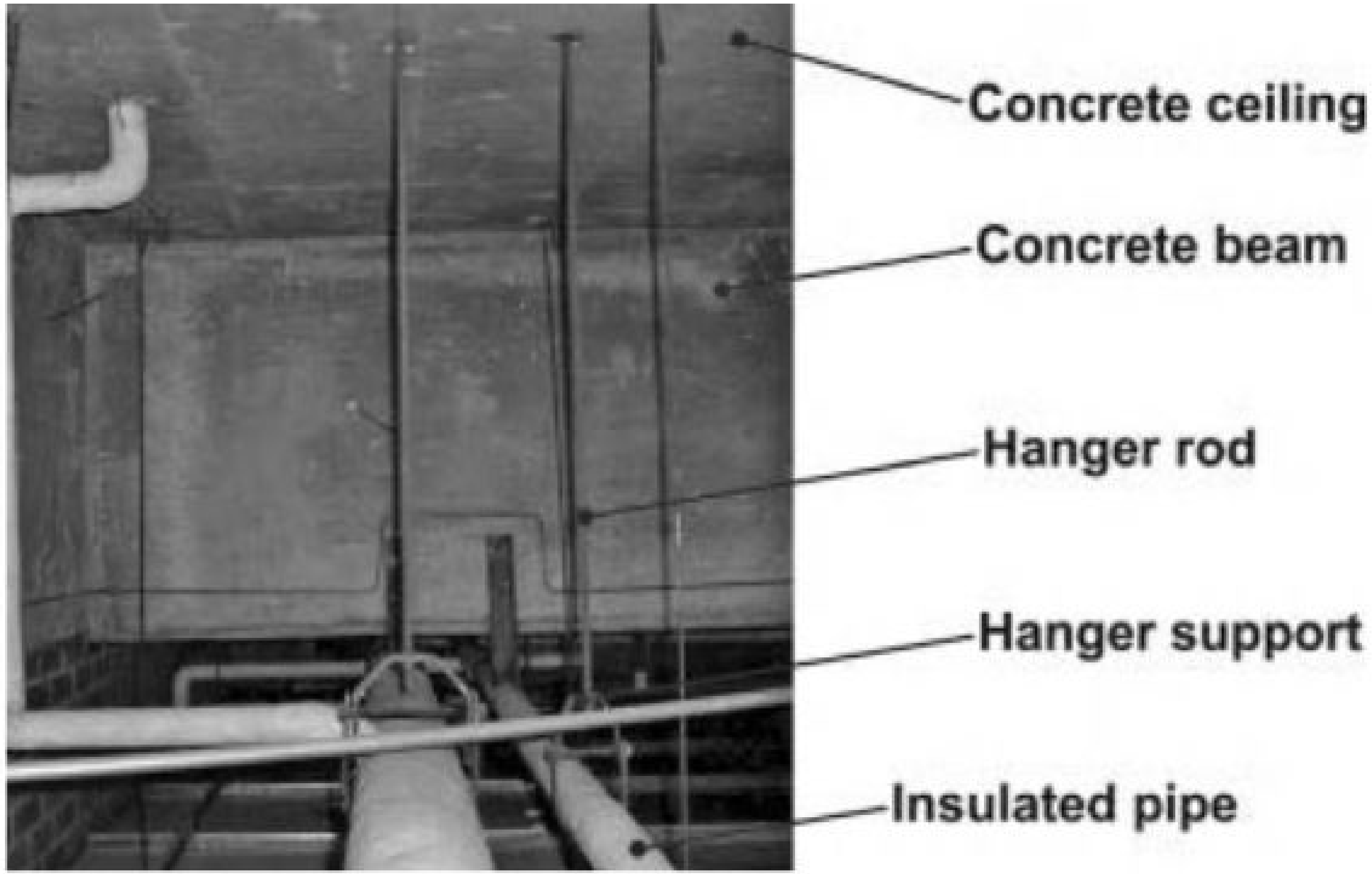


Table 3.3 Hanger spacing for straight stationary pipes and tubes [1]

Nominal Pipe Diameter (in.)	Steel Pipe (ft)		Copper Pipe (ft)	
	Water Service	Steam Service	Water Service	Steam Service
1/4	7	8	5	5
3/8	7	8	5	6
1/2	7	8	5	6
3/4	7	9	5	7
1	7	9	6	8
1 1/4	7	9	7	9
1 1/2	9	12	8	10
2	10	13	8	11
2 1/2	11	14	9	13
3	12	15	10	14
3 1/2	13	16	11	15
4	14	17	12	16
5	16	19	13	18
6	17	21	14	20
8	19	24	16	23
10	20	26	18	25
12	23	30	19	28
14	25	32		
16	27	35		
18	28	37		
20	30	39		
24	32	42		
30	33	44		

Table 3.4 Minimum hanger rod size for straight stationary pipes and tubes [1]

Nominal Pipe Diameter (in.)	Minimum Rod Size (in.)
$\frac{1}{4}$–2	$\frac{3}{8}$
$2\frac{1}{2}$–$3\frac{1}{2}$	$\frac{1}{2}$
4–5	$\frac{5}{8}$
6	$\frac{3}{4}$
8–12	$\frac{7}{8}$
14–18	1
20–24	$1\frac{1}{4}$

Practical Note 2. Piping System Supported by Brackets

- Not all pipes are installed by being hung on pipe hangers and supports. Pipes may be supported by mounting them on **brackets** that are attached to concrete walls. Shown in Figure below is a part of a piping system that is installed on stainless steel strut channels. A 1/4-hp (horsepower) in-line pump is also supported by the pipes and the channels.

Pipe Installation

- **Figure.** Pipes and an in-line pump mounted on brackets

Fluid Machines (Pumps) and Pump–Pipe Matching

Classifications and Terminology

- Pumps serve to move liquids. The pump adds energy to the fluid to keep it moving, to overcome head losses, and/or to build pressure on the fluid to overcome elevation head in the line.
- As the fluid passes through the pump and energy is added, the **discharge pressure** of the exiting fluid becomes greater than the pressure of the inlet fluid. There are many types of pumps available on the market.

Types of Pumps

a) *Gas pumps*: These pumps move gases. Examples of these types of pumps are fans and compressors.

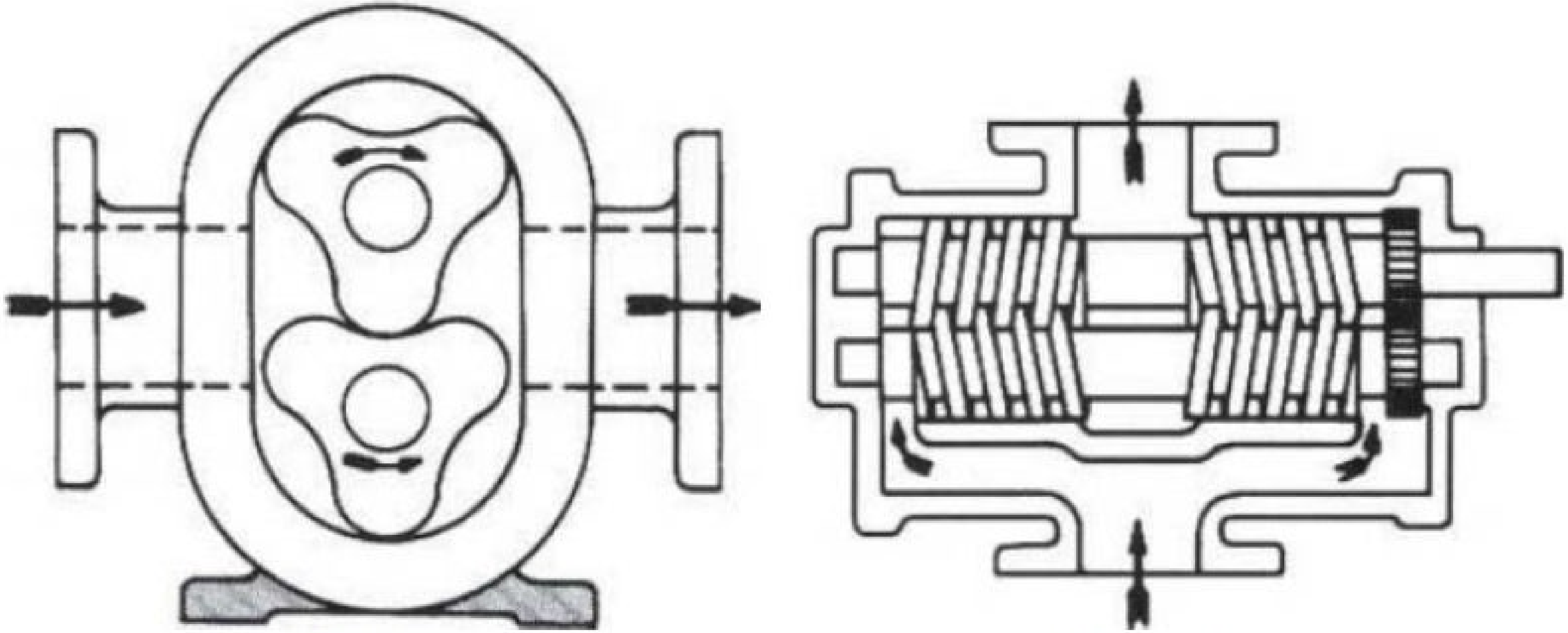
b) *Positive displacement pumps*: These pumps pressurize the fluid by contracting or changing their boundaries to force fluid to flow. Suction is achieved when the boundaries of the pump expand or the volume of the pump becomes larger.

Fluid Machines (Pumps) and Pump–Pipe Matching

- Examples of positive displacement pumps include the human heart, flexible-tube peristaltic pumps, and double-screw pumps. These pumps are capable of creating a significant vacuum pressure at their inlets, even when dry, and are able to lift fluids from long distances below the pump. These pumps are also called self-priming pumps.

c) ***Dynamic pumps***: Rotating blades are used to supply energy to the fluid. The blades impart momentum to the fluid. Fluid enters the eye of the impeller, is flung from the impeller blades into the scroll case (volute) of the pump where the fluid is pressurized. The fluid exits at high pressure.

Fluid Machines (Pumps) and Pump–Pipe Matching

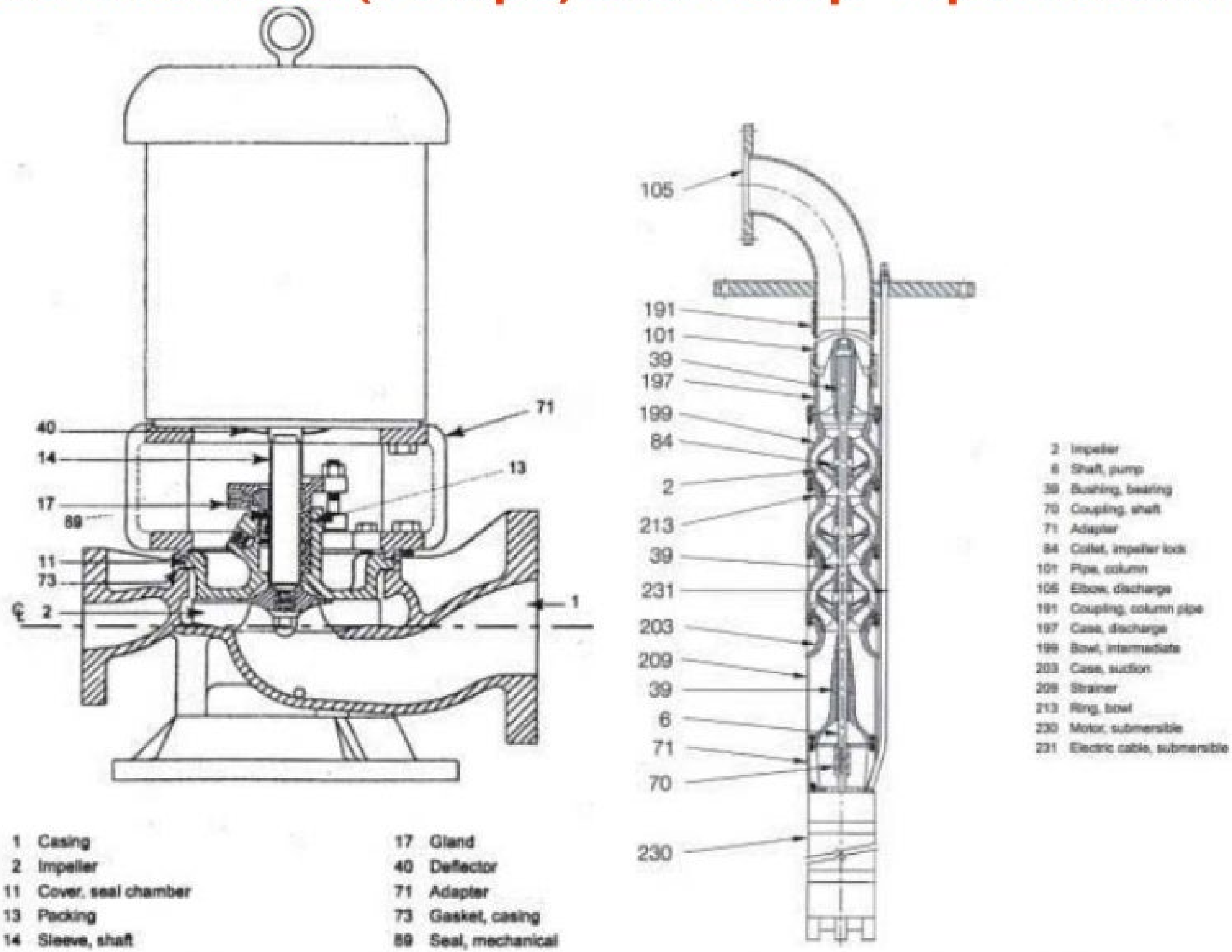


Three-lobe rotary pump
(*a*)

Two screw pump
(*b*)

Fig. Examples of positive displacement pumps

Fluid Machines (Pumps) and Pump–Pipe Matching



In-line centrifugal pump

(*c*)

Vertical multi-stage submersible pump

(*d*)

Fig. Example of a dynamic pump (centrifugal pump).

Pump Fundamentals

- There are several fundamental parameters that need to be mentioned before proceeding to pump sizing and selection.

a) *Pump Capacity*: Volume flow rate of fluid through the pump:

$$\dot{V} = \frac{\dot{m}}{\rho}$$

b) *Pump Net Head*: Used to increase the fluid energy (usually in units of length). An expression for the pump net head can be found by considering the energy equation. Thus, the pump specific work is

$$w_{\text{pump}} = \left(\frac{p_2}{\rho} + \frac{V_2^2}{2} + gz_2\right) - \left(\frac{p_1}{\rho} + \frac{V_1^2}{2} + gz_1\right) + h_{\text{lT}},$$

where points 1 and 2 are points in the pipe system chosen by the engineer.

Pump Fundamentals

- Dividing by g gives the pump net head (in units of length):

$$H_{\text{pump}} = \frac{w_{\text{pump}}}{g} = \left(\frac{p_2}{\rho g} + \frac{V_2^2}{2g} + z_2\right) - \left(\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1\right) + H_{\text{lT}}$$

- If a **control volume** were drawn around the pump only, the following could be assumed:
- $H_{\text{lT}} = 0$ (applies to pipe head loss, only); point 1 is the pump inlet; point 2 is the pump outlet; $z_2 = z_1$ (horizontally mounted pump); $D_2 = D_1$ (assuming suction and discharge pipe diameters are equal); $V_2 = V_1$ (no area changes yields equal velocities).
- Therefore

$$H_{\text{pump}} = \frac{p_2}{\rho g} - \frac{p_1}{\rho g} = \frac{p_2 - p_1}{\rho g} = \frac{p_{\text{outlet}} - p_{\text{inlet}}}{\rho g} = \frac{\Delta p_{\text{rise}}}{\rho g}$$

- The energy required to generate a pressure rise across the pump is directly related to the pump net head.

Pump Fundamentals

c) *Water Horsepower*: Power delivered directly to the fluid by the pump:

$$\dot{W}_{\text{water horsepower}} = \dot{m} g H_{\text{pump}} = \rho g \dot{V} H_{\text{pump}}$$

d) *Brake Horsepower* (bhp): External power supplied to the pump by a mechanical shaft or an electrical motor. For a rotating shaft that supplies the bhp:

$$\text{bhp} = \dot{W}_{\text{pump,shaft}} = \omega T_{\text{shaft}},$$

where ω is the rotational speed and T_{shaft} is the torque generated by the shaft.

e) *Pump efficiency*: Ratio of useful power to supplied power:

$$\eta_{\text{pump}} = \frac{\dot{W}_{\text{water horsepower}}}{\text{bhp}} = \frac{\rho g \dot{V} H_{\text{pump}}}{\omega T_{\text{shaft}}}.$$

Pump Performance and System Curves

- Curves of H_{pump}, η_{pump}, bhp as functions of $\dot{V}$ are called **pump performance curves**.
- Consider the H_{pump} versus $\dot{V}$ curve shown in the Figure for a centrifugal pump.
- As the pump net head increases, total resistance to flow increases and the volume flow rate decreases. At the **shut-off head**, the pump net head is a maximum ($H_{\text{pump}} = H_{\text{pump,max}}$) and the volume flow rate or pump capacity is zero ($\dot{V} = 0$). There is no fluid flow at the pump shut-off head.

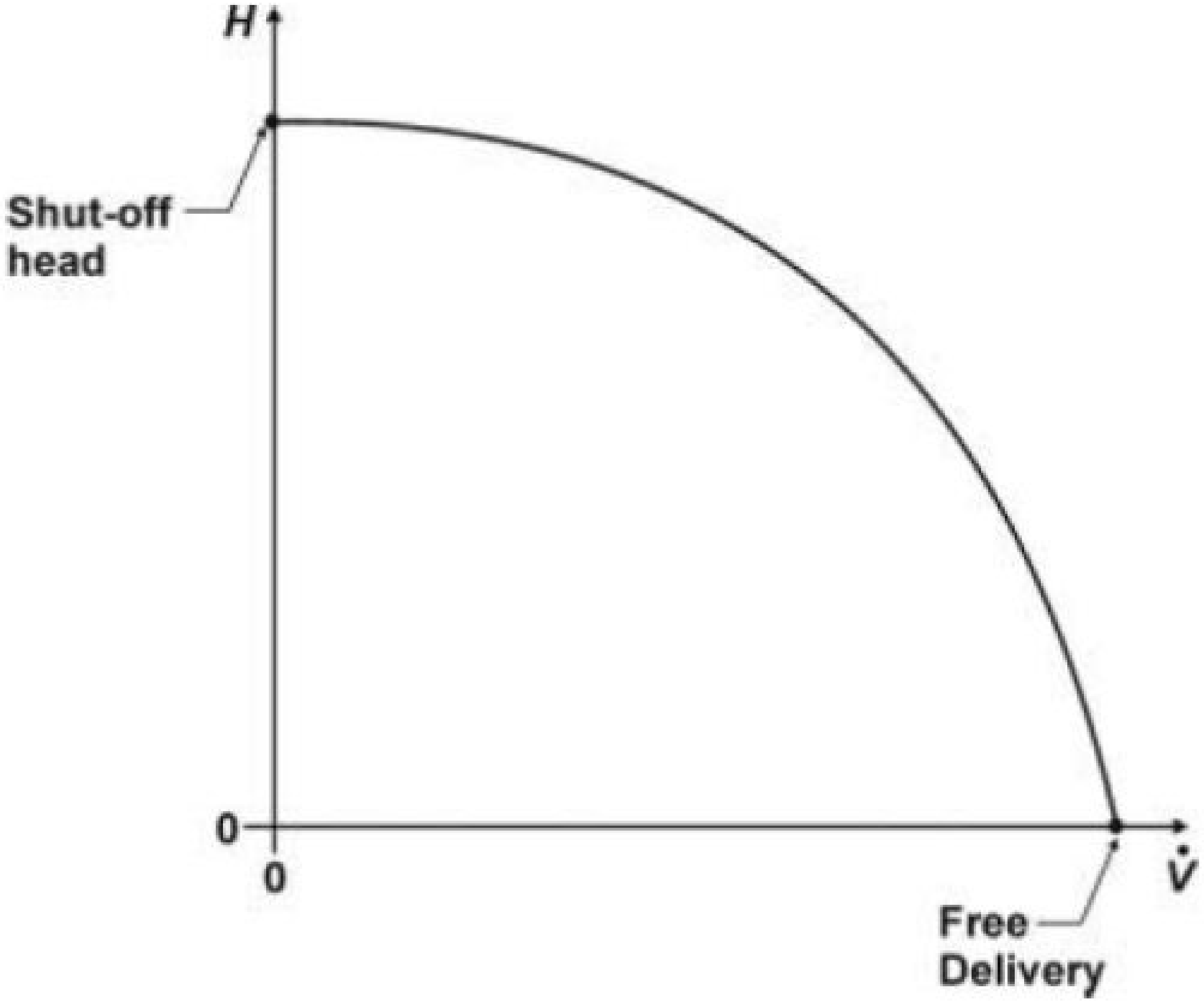


Fig. Schematic of a H_{pump} versus $\dot{V}$ curve for a centrifugal pump

Pump Performance and System Curves

- As the pump net head decreases, total resistance to flow decreases and the volume flow rate increases. At **free delivery**, the pump net head is zero ($H_{pump} = 0$) and the volume flow rate or pump capacity is a maximum ($\dot{V} = \dot{V}_{max}$). No energy is added to the fluid, and the fluid flows through the pump as though it were a pipe.
- Consider the η_{pump} versus $\dot{V}$ curve shown in Figure below. At the pump shut-off head, $\dot{V} = 0$. Therefore,

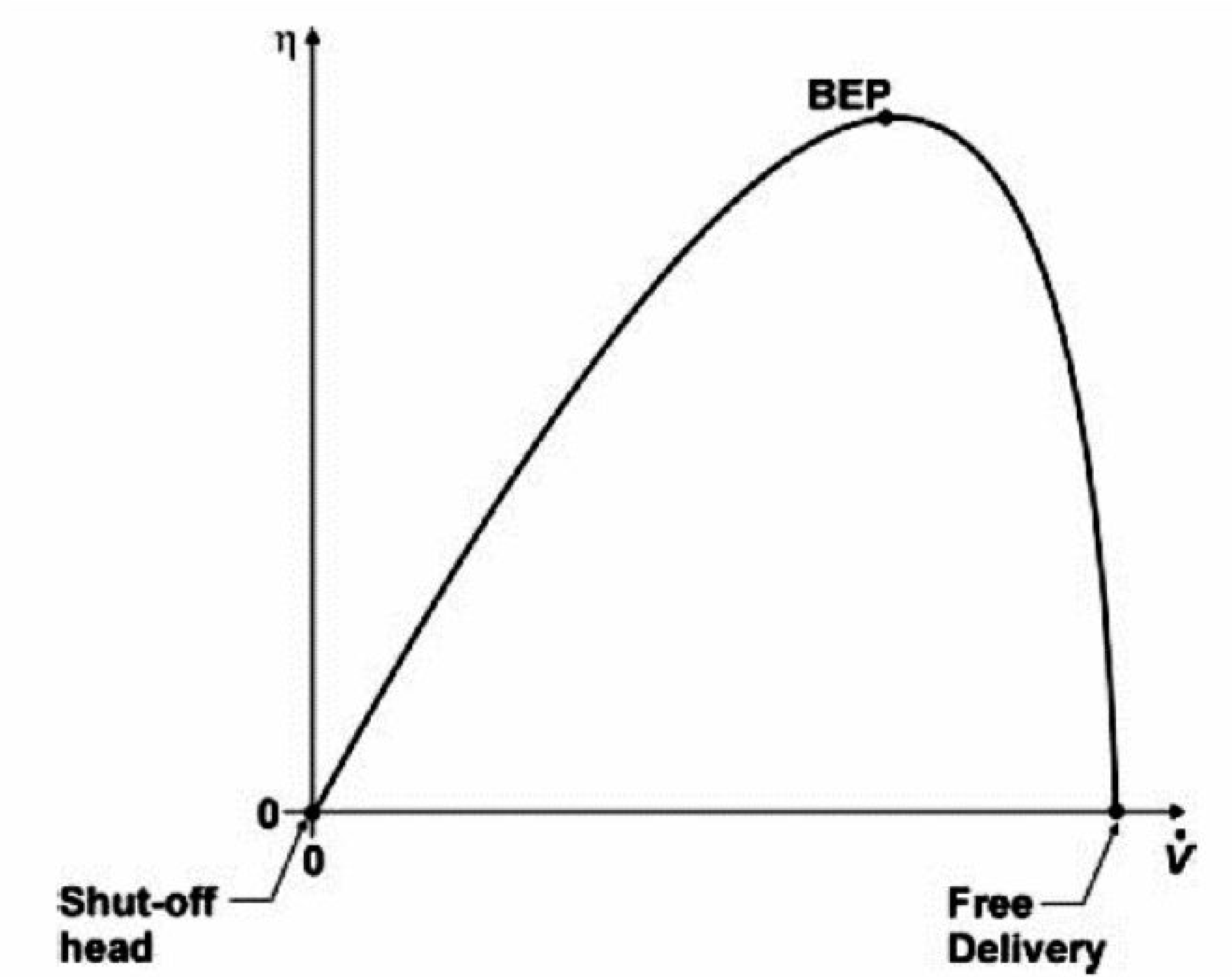
$$\eta_{pump} = \frac{\rho g \dot{V} H_{pump}}{\omega T_{shaft}} = \frac{\rho g\,(0)\,H_{pump}}{\omega T_{shaft}} = 0.$$

The pump efficiency is zero.
At free delivery, $H_{pump} = 0$. Therefore,

$$\eta_{pump} = \frac{\rho g \dot{V} H_{pump}}{\omega T_{shaft}} = \frac{\rho g \dot{V}\,(0)}{\omega T_{shaft}} = 0.$$

The pump efficiency is zero.

Pump Performance and System Curves



- Fig. Schematic of a η_{pump} versus $\dot{V}$ curve

Pump Performance and System Curves

- The **best efficiency point (BEP)** of the pump is the maximum pump efficiency ($\eta_{pump,max}$). Note that the performance curve will show the pump capacity that produces the BEP.
- When selecting a pump, the design engineer needs to determine the total amount of head that is required to overcome all the losses in the piping system, build up the pressures required by the design, overcome elevation differences, and increase the fluid velocity (as required) across selected points in the system.
- Thus, between two points, 1 and 2, the required head for the system is

$$H_{pump,required} = \left(\frac{p_2}{\rho g} + \frac{V_2^2}{2g} + z_2\right) - \left(\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1\right) + H_{lT}$$

- The pump performance curve shows the amount of head that is available ($H_{pump,available}$) and can be delivered by a specific pump.
- The pump head required must be calculated, and the performance curve must be checked to determine if the selected pump can provide the required head.

Pump Performance and System Curves

- Calculated values of the required pump head ($H_{\text{pump,required}}$) at different pump capacities (flow rates) can be plotted alongside the performance curve.
- These curves of $H_{\text{pump,required}}$ versus $\dot{V}$ are called **system curves**. The point of intersection between the pump performance curve and the system curve is the **operating point of the pipe system**.
- Therefore, at the operating point,

$$H_{\text{pump,available}} = H_{\text{pump,required}}$$

- This indicates that the selected pump can provide the total head required by the system.

Pump Performance and System Curves

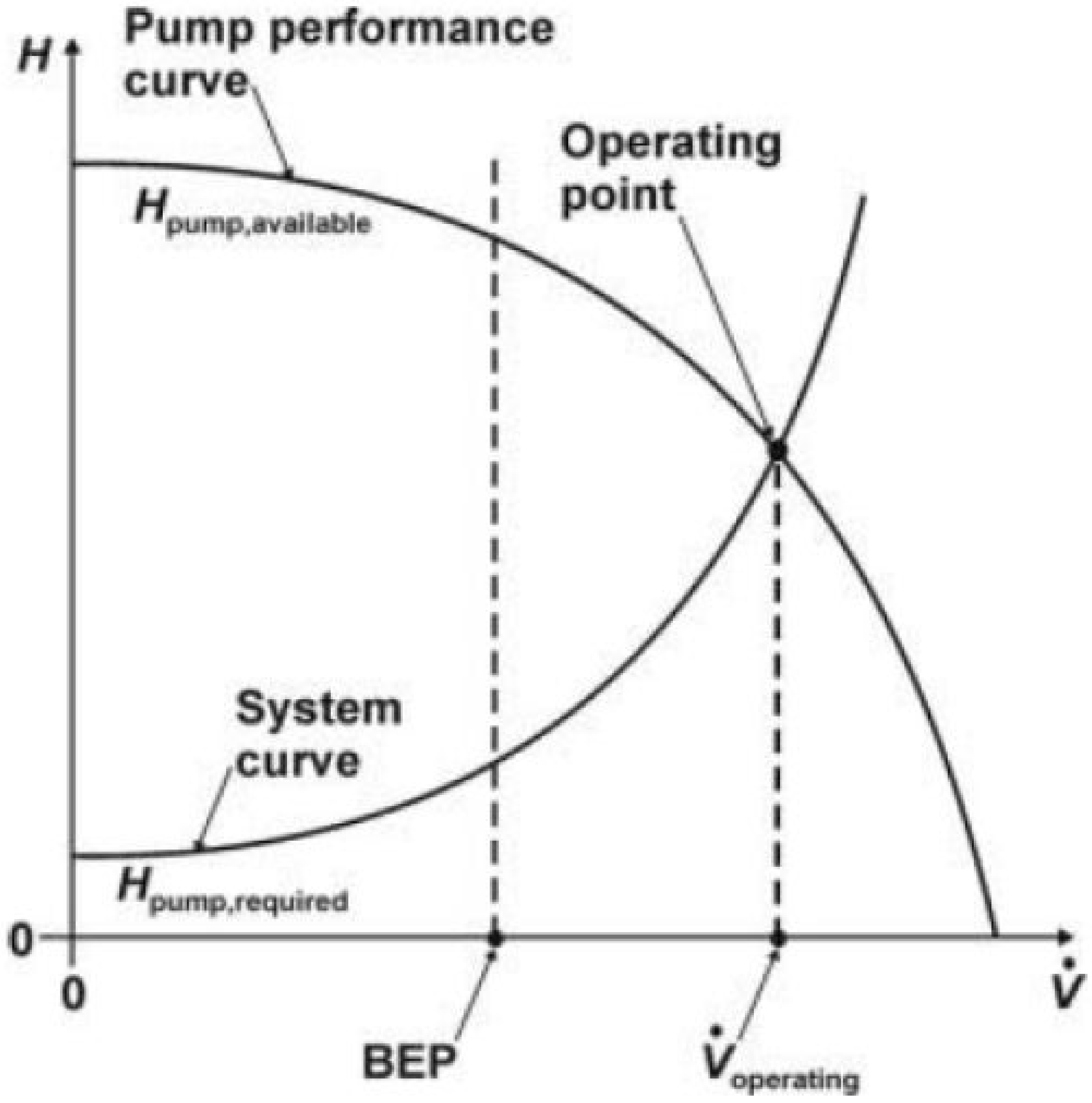


Fig. Schematic of a system curve intersecting a pump performance curve. This performance curve is for a pump of a fixed impeller diameter and rotational speed

Pump Performance and System Curves

Practical Note 3. Manufacturers' Pump Performance Curves

- The pump performance curves must be determined experimentally by the pump manufacturers. These are usually available in equipment catalogs online or in hard copy. The curves are usually (almost always) based on water as the working fluid.
- The system curve or the operating point must be determined by the design engineer by considering the pipe system flow rate and the total system pump head required ($H_{\text{pump,required}}$).

Pump Performance Curves for a Family of Pumps

- Manufacturers will typically (almost always) provide a group of pump performance curves for a group of pumps, all on one plot.
- Below in the Figure shown is a schematic of a group of performance curves for a family of "**geometrically similar**" pumps. Note the following points:

a) D_1, D_2, D_3 are impeller diameters of each pump.

b) The casing enclosure is the same for each pump to satisfy the requirement of geometric similarity.

c) Different pump efficiencies are shown. Note the shape of the efficiency curves. The BEP is as shown.

d) These curves will be determined experimentally by the manufacturer for a specific liquid. Typically, the liquid is water

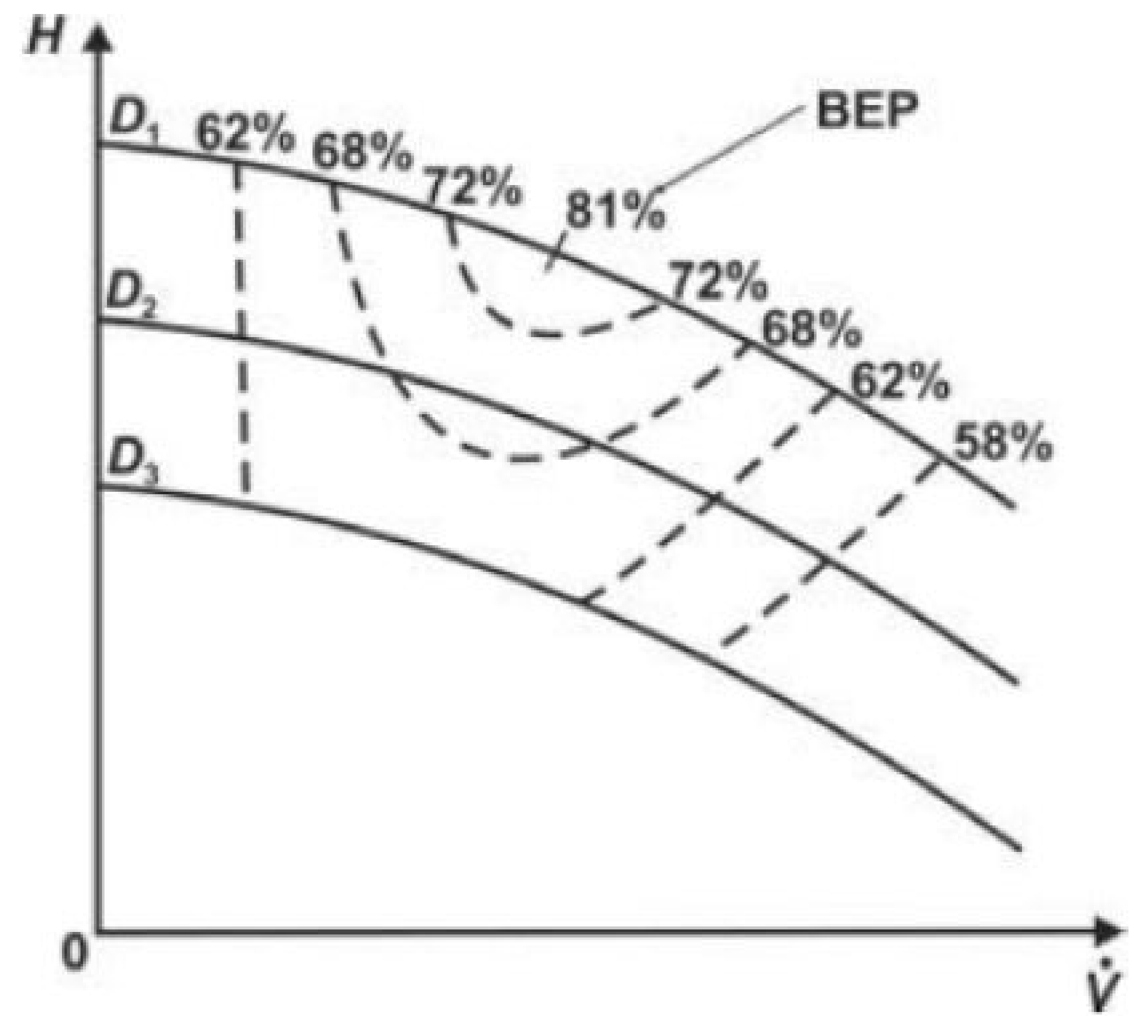
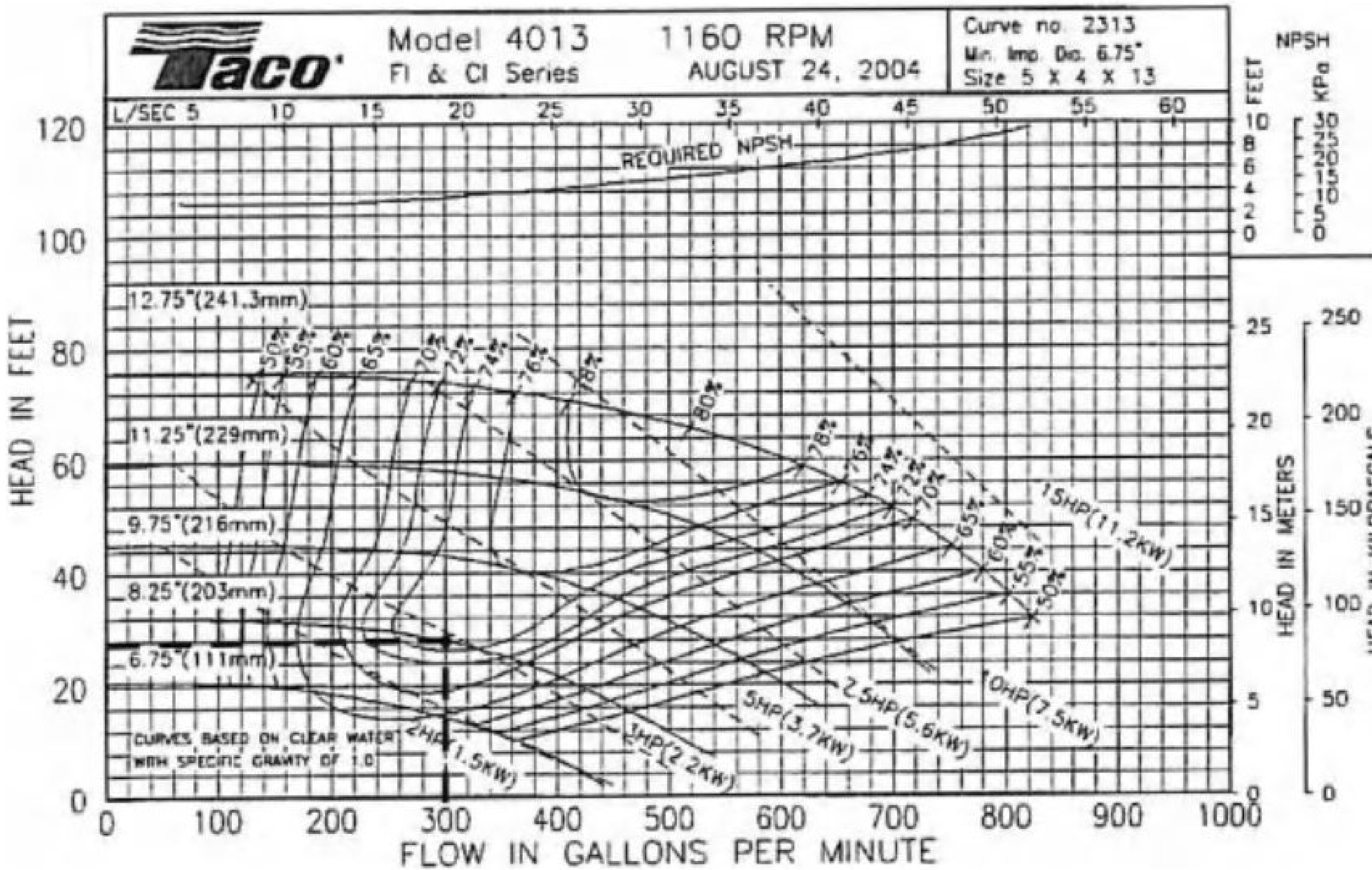


Fig. Performance curves for a family of geometrically similar pumps

A Manufacturer's Performance Plot for a Family of Centrifugal Pumps

- Figure below shows a real pump performance plot from a catalog provided by Taco Inc.



A Manufacturer's Performance Plot for a Family of Centrifugal Pumps

Note the following points:

a) *Model Information:* Model 4013, FI and CI Series centrifugal pumps.

b) *Pump Speed:* 1160 rpm.

c) *Casing Size:* 5 in. × 4 in. × 13 in. (5 in. suction diameter, 4 in. discharge diameter, 13 in. casing).

d) *Identify the Axes:* Pump head available on the *y*-axis and pump capacity on the *x*-axis.

e) Provided are performance curves for each impeller diameter.

f) The BEP is approximately 80% for this family of pumps.

g) Provided are curves for the bhp. The minimum pump power is 2 hp and the maximum pump power is 15 hp for this family of pumps.

A Manufacturer's Performance Plot for a Family of Centrifugal Pumps

- Consider the following scenario:
- Select an appropriate Taco pump to provide 300 gpm of fluid ($\dot{V}$) and overcome 28 ft of head ($H_{\text{pump,required}}$)
  - Select a pump with: $D_{\text{impeller}} = 8.25$ in.
  - $\eta_{\text{pump}} = 74\%$
  - bhp = 3 hp.

Practical Note 4. "To-the-point" Design

Choosing a 3-hp motor for the pump above would be recommended for "**to the point**" design. However, what would happen if the flow rate spiked to 325 gpm (an 8% increase in flow rate), which is possible in practice? **The motor would overload.** Hence, for this design, we would select a 5-hp motor, which would be **nonoverloading** for the entire pump curve

A Manufacturer's Performance Plot for a Family of Centrifugal Pumps

Practical Note 5. Oversizing Pumps

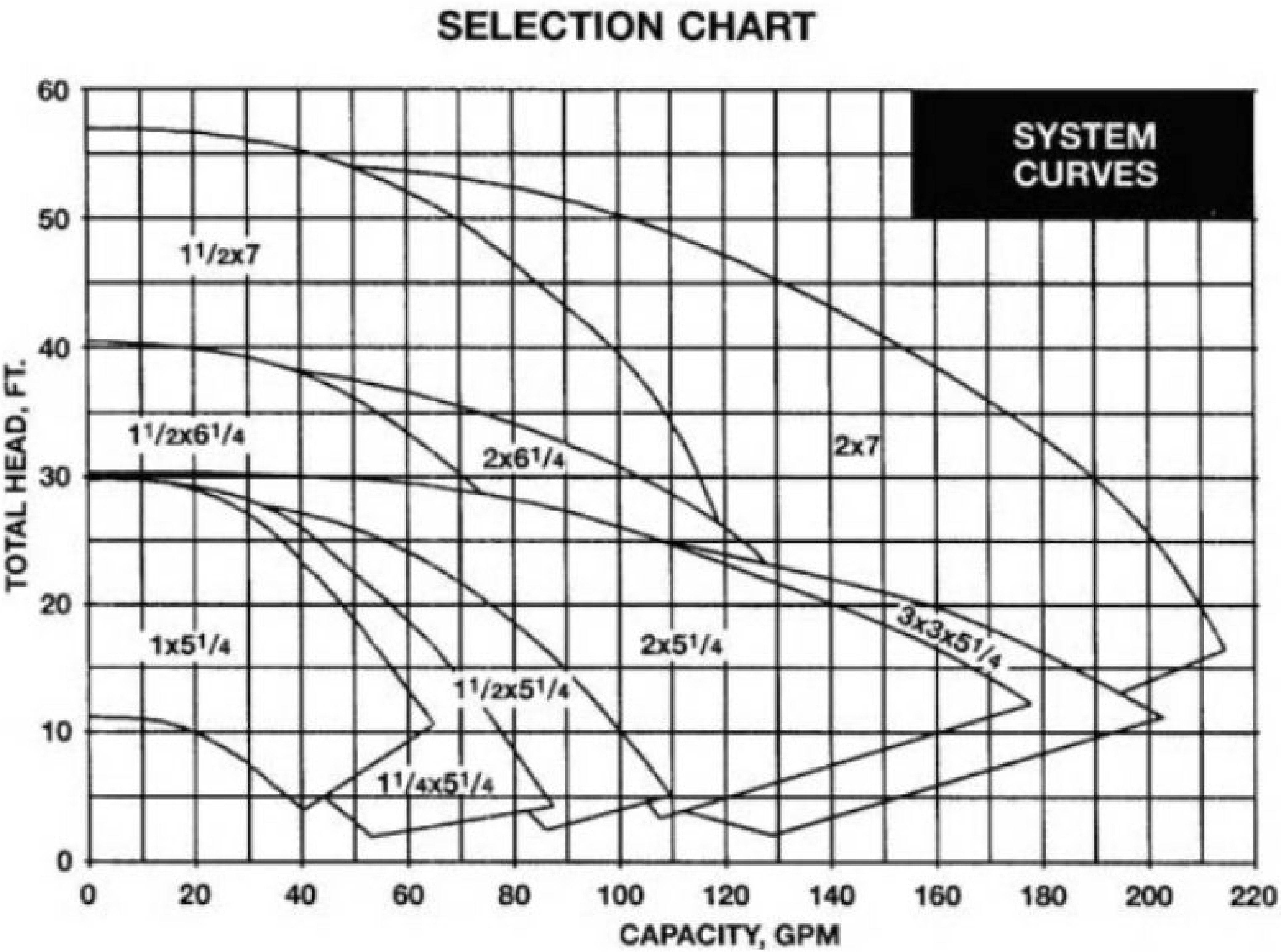
Do not oversize your pumps simply to be safe. After selecting the next larger motor size to ensure that the pump would be nonoverloading for the entire pump curve, do not proceed to oversize the pump further to be safe. Hence, do not choose a 10-hp motor, when a 5-hp motor is sufficient to provide nonoverloading. At 300 gpm, all that is needed is 3–5 hp. Choosing a 10-hp motor would simply waste energy, increase the installation cost, and require extra resources to support the larger pump.

Example. Pump Selection from Manufacturer's Performance Plots

- You are a design engineer charged with the responsibility of selecting an appropriate in-line mounted centrifugal pump for an application. The flow rate will be constant at 80 gpm, but the required head could vary between 20 and 40 ft. Select an appropriate Bell & Gossett Series 60 pump for this application.

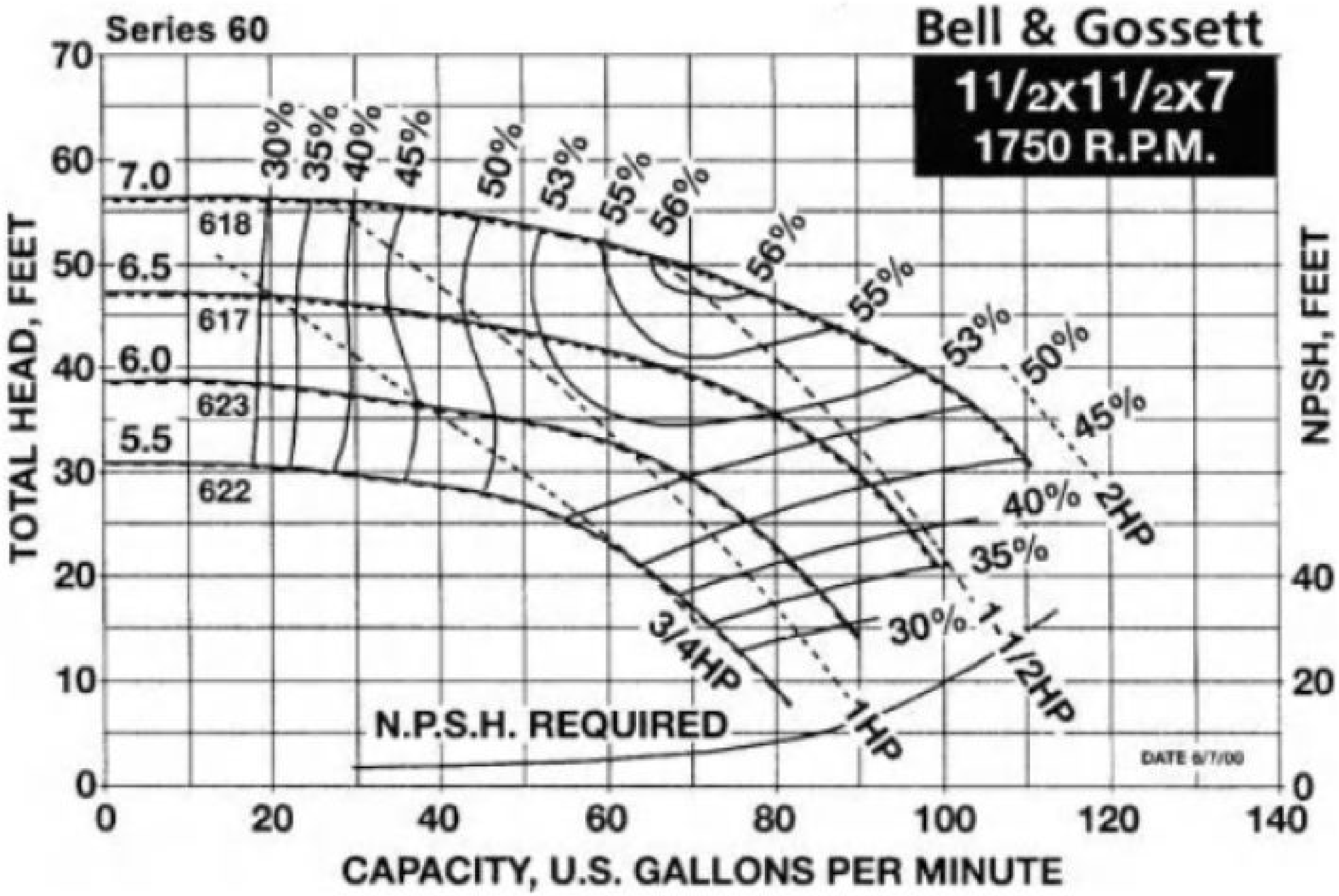
- **Solution.** A study of each performance plot from the large Bell & Gossett Series 60 centrifugal pump catalog to find an appropriate pump could be time consuming. A **master pump selection chart** is used to identify a family (or families) of pumps that will meet the requirements of the design. The master pump selection chart for the Bell & Gossett Series 60 centrifugal pumps is shown below.

Example. Pump Selection from Manufacturer's Performance Plots



Example. Pump Selection from Manufacturer's Performance Plots

- From the master pump selection chart, the family of pumps with a $1^1/_2$ in. × 7 in. casing is chosen. This family of pumps covers the range of interest for the head loss (20–40 ft). The pump impeller diameter, motor power, and efficiency are found from the performance plot for the $1^1/_2$ in. × 7 in. family of pumps. The plot is shown below.



Example. Pump Selection from Manufacturer's Performance Plots

- From the performance plot, and for $\dot{V}$ = 80 gpm and H_{pump} = 40 ft (to cover the entire range of interest):

$$D_{\text{impeller}} = 7.0 \text{ in.}$$
$$\eta_{\text{pump}} \approx 55\%$$
$$\text{bhp} \approx 2 \text{ hp.}$$

- For this family of pumps, the BEP is approximately 56%. The efficiency of the selected pump is close to the maximum efficiency.
- "To-the-point" design would require the selection of a $1^1/_2$-hp motor for this pump. To avoid motor overloading, a 2-hp motor was chosen instead.

Cavitation and Net Positive Suction Head

- Cavitation is the formation of vapour cavities in a liquid, small liquid-free zones ("bubbles" or "voids"), that are the consequence of forces acting upon the liquid.
- It usually occurs when the liquid is subjected to rapid changes of pressure that cause the formation of cavities in the liquid where the pressure is relatively low.
- When subjected to higher pressure, the voids implode and can generate an intense shock wave.
- Collapsing voids that implode near to a metal surface cause cyclic stress through repeated implosion. This results in surface fatigue of the metal causing a type of wear also called "Cavitation".

Cavitation and Net Positive Suction Head

- If left untreated, pump cavitation can cause:
  - Failure of pump housing
  - Destruction of impeller
  - Excessive vibration -leading to premature seal and bearing failure
  - Higher than necessary power consumption
  - Decreased flow and/or pressure

What can be done to prevent cavitation?

- Don’t force the pump to operate too far to the right or left of its BEP (best efficiency point) . Doing so will result in cavitation over time. If a pump is correctly sized and not starved, the pump will run at the intended speed while maintaining the BEP.
- Altitude has also a major effect on pump cavitation. When pumps operate at higher altitudes, special attention must be given to make sure that cavitation does not occur since liquids boil at much lower temperature. The boiling point of a liquid depends on the vapor pressure of that liquid matching the pressure of the gas above it.

Cavitation and Net Positive Suction Head

- Suction into a pump may occur as reduced pressure and high velocity is created to promote the movement of fluid into the pump.
- The inlet pressures on the suction side of the pump may be significantly lower than the discharge pressures, and may be lower than atmospheric pressure.
- In some cases in liquid pumps, the inlet pressure to the pump may become lower than the **vapor pressure** of the liquid at the operating temperature. When this occurs, the liquid will vaporize to form bubbles. Pressure drop across the pump inlet passage or losses in the pump impeller may also decrease the pump inlet pressure to values lower than the vapor pressure of the fluid.
- Hence, vapor bubbles will form in liquid pumps if

$$P_{\text{inlet}} < P_{\text{vapor,liquid}}$$

- In a liquid pump, these vapor-filled bubbles are called **cavitation bubbles**. In the high-pressure regions of the pump, these cavitation bubbles will collapse, resulting in damage to the pump and reduction in pump performance.

Cavitation and Net Positive Suction Head

- Other negative consequences of cavitation include

a) noise;

b) vibration;

c) pump efficiency reduction;

d) erosion of pump impeller, casing, and blades due to high-pressure explosion of the bubbles.

- The presence of vapor in the impeller can also result in a loss of pump pressure rise. In effect, for a vapor-filled impeller, a total loss of pump performance will occur.

- It is necessary to ensure that $P_{\text{inlet}} > P_{\text{vapor,liquid}}$ to avoid cavitation and pump performance loss. **NPSH (Net Positive Suction Head)** can be used to verify if this requirement will be met.

- Therefore, in units of length

$$\text{NPSH} = \left(\frac{p}{\rho g} + \frac{V^2}{2g}\right)_{\text{pump,inlet}} - \frac{P_{\text{vapor}}}{\rho g}.$$

Cavitation and Net Positive Suction Head

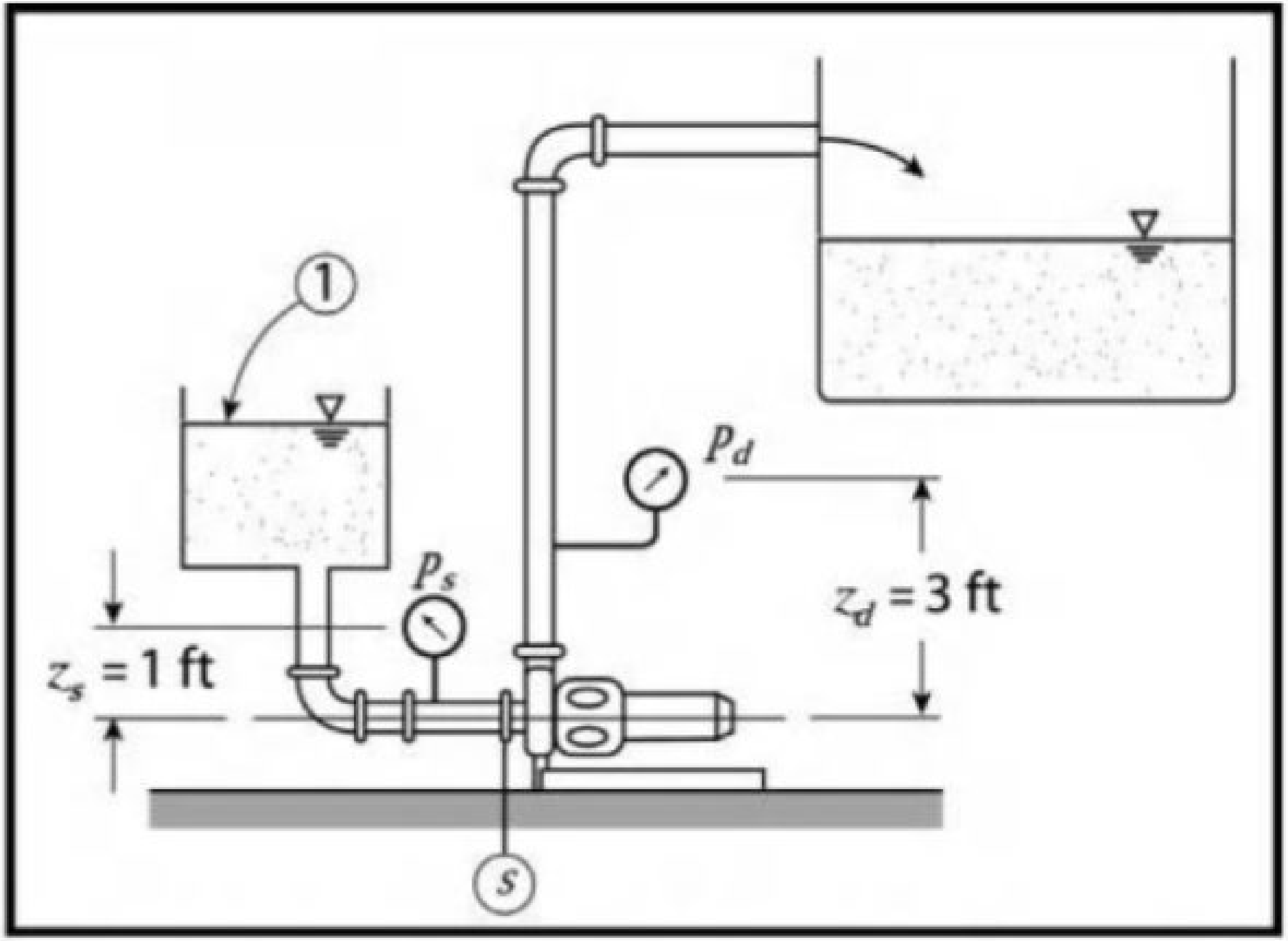
- Pump manufacturers will provide the $NPSH_{required}$ (NPSHR) on the performance plots for a family of pumps (NPSHR vs. $\dot{V}$), and is determined experimentally by the manufacturer. This NPSHR value must be compared with the calculated NPSH.
- Thus, to avoid cavitation,

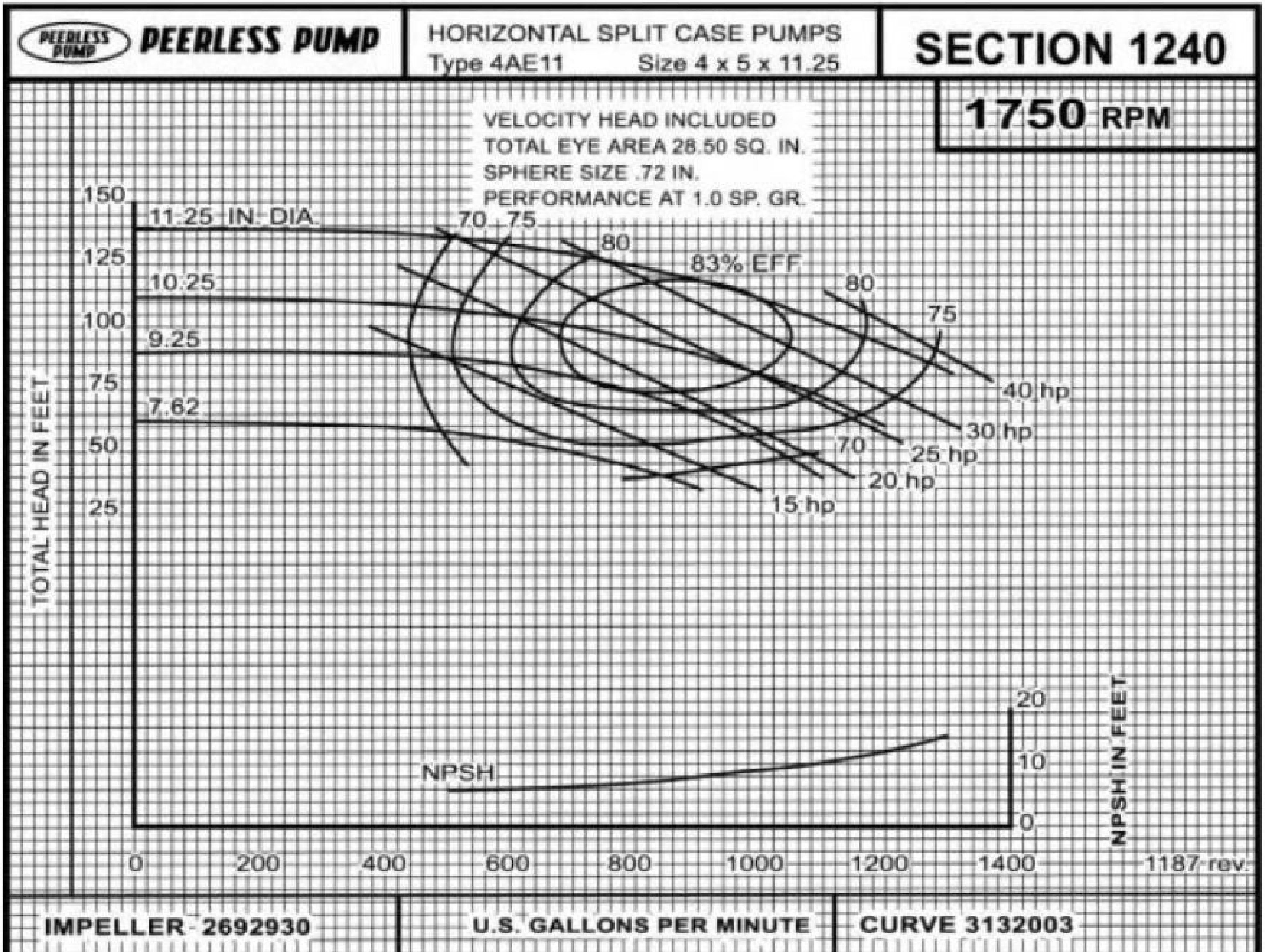
$$\text{NPSHR} < \text{NPSH}$$

- The NPSH must be calculated by the design engineer for comparison with the NPSHR to ensure that sufficient NPSH is available. It is common to see NPSH referred to as net positive suction head available (NPSHA). The NPSHA is always calculated

Example. Net Positive Suction Head

- A centrifugal *Peerless Pump* Type 4AE11 is tested at 1750 rpm using the flow system shown. The water level in the inlet reservoir is 3.5 ft above the pump centerline; the inlet line consists of 6 ft of 4-in. diameter Class 150 cast-iron pipe, a standard elbow, and a fully open gate valve. Calculate the NPSHA at the pump inlet. The volume flow rate is given as 1200 gpm of water at 60◦F. Will cavitation occur at this temperature?





Source: Peerless Pump Company (reprinted with permission)

Solution. The NPSHA is

$$\text{NPSHA} = \left(\frac{p_s}{\rho g} + \frac{V_s^2}{2g}\right)_{\text{pump,inlet}} - \frac{P_{\text{vapor}}}{\rho g}.$$

The energy equation will be used to find the pump inlet pressure, p_s:

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 = \frac{p_s}{\rho g} + \alpha_s \frac{V_s^2}{2g} + z_s + H_{\text{lT}}.$$

Point 1 is the free surface of the water in the reservoir. Thus, $V_1 \approx 0$ and $P_1 = P_{\text{atm}}$. Therefore,

$$\frac{p_s}{\rho g} = \frac{P_{\text{atm}}}{\rho g} - \alpha_s \frac{V_s^2}{2g} + (z_1 - z_s) - H_{\text{lT}}.$$

Assume that the pipe flow is turbulent. That is, $\alpha_s \approx 1$. The Reynolds number will need to be verified. Hence,

$$\frac{p_s}{\rho g} = \frac{P_{\text{atm}}}{\rho g} - \frac{V_s^2}{2g} + (z_1 - z_s) - H_{\text{lT}}.$$

Then,

$$\text{NPSHA} = \frac{P_{\text{atm}}}{\rho g} - \frac{V_s^2}{2g} + (z_1 - z_s) - H_{\text{lT}} + \frac{V_s^2}{2g} - \frac{P_{\text{vapor}}}{\rho g}$$

$$\text{NPSHA} = \frac{P_{\text{atm}} - P_{\text{vapor}}}{\rho g} + (z_1 - z_s) - H_{\text{lT}}.$$

Find the total head loss in the pipe. Note that since the diameter of the pipe does not change, the law of conservation of mass requires that the pipe average velocity be constant. Therefore,

$$H_{\text{lT}} = H_{\text{l}} + H_{\text{lm}} = \left(f\frac{L}{D} + \sum K\right)\frac{V_s^2}{2g}.$$

For the 4-in. nominal diameter pipe, it will be assumed that the system is assembled with flanged fittings. The loss coefficients are shown in Table A.14 (or other sources) and are shown below. Note that the values for the 4-in. diameter pipe were used:

90° regular elbow: $K_{\text{elbow}} = 0.3$
Open gate valve: $K_{\text{gate}} = 0.16$
Sharp-edged inlet: $K_{\text{inlet}} = 0.5$.

The Reynolds number and the relative roughness of the pipe are needed to determine the friction factor, f:

$$\mathrm{Re}_D = \frac{\rho V_s D}{\mu} = \frac{V_s D}{\nu} = \frac{4\dot{V}}{\nu \pi D}.$$

At 60°F, $\nu = 0.121 \times 10^{-4}\ \mathrm{ft}^2/\mathrm{s}$. The inner diameter of 4-in. nominal Class 150 cast-iron piping is 4.10 in. (see Table A.9):

$$\mathrm{Re}_D = \frac{4\,(1200\ \mathrm{gpm})}{\left(0.121 \times 10^{-4}\mathrm{ft}^2/\mathrm{s}\right)\pi\,(4.10\ \mathrm{in.})} \times \frac{35.315\ \mathrm{ft}^3/\mathrm{s}}{15850\ \mathrm{gpm}} \times \frac{12\ \mathrm{in.}}{1\ \mathrm{ft}} = 8.25 \times 10^5.$$

Since $\mathrm{Re}_D > 4000$, the flow is turbulent.

The average roughness of cast-iron piping is 0.00085 ft (Table A.1). Thus, the relative roughness is

$$\frac{\varepsilon}{D} = \frac{0.00085\ \mathrm{ft}}{4.10\ \mathrm{in.}} \times \frac{12\ \mathrm{in.}}{1\ \mathrm{ft}} = 0.00249.$$

From the Moody chart, the friction factor is

$$f \approx 0.025.$$

Therefore,

$$H_{\mathrm{IT}} = \left[(0.025)\,\frac{6\ \mathrm{ft}}{4.10\ \mathrm{in.}} \times \frac{12\ \mathrm{in.}}{1\ \mathrm{ft}} + (0.3 + 0.16 + 0.5)\right] \left[\frac{4\,(1200\ \mathrm{gpm})}{\pi\,(4.10\ \mathrm{in.})^2} \times \frac{35.315\ \mathrm{ft}^3/\mathrm{s}}{15850\ \mathrm{gpm}} \times \left(\frac{12\ \mathrm{in.}}{1\ \mathrm{ft}}\right)^2\right]^2 \frac{1}{2\left(32.2\ \mathrm{ft/s}^2\right)}$$

$H_{\mathrm{IT}} = 18.5$ ft.
At 60°F, $P_{\mathrm{vapor}} = 0.256$ psia. $P_{\mathrm{atm}} = 14.7$ psia.
Therefore,

$$\mathrm{NPSHA} = \frac{(14.7 - 0.256)\ \mathrm{lbf/in.}^2}{\left(62.36\ \mathrm{lb/ft}^3\right)\left(32.2\ \mathrm{ft/s}^2\right)} \times \frac{32.2\ \mathrm{lb\text{-}ft/s}^2}{1\ \mathrm{lbf}} \times \left(\frac{12\ \mathrm{in.}}{1\ \mathrm{ft}}\right)^2 + (3.5\ \mathrm{ft}) - 18.5\ \mathrm{ft}$$

NPSHA = 18.4 ft.
Cavitation will not occur if NPSHA > NPSHR. From the performance plots for this pump operating at 1200 gpm,
NPSHR ≈ 12 ft.
Cavitation will not occur at this temperature.

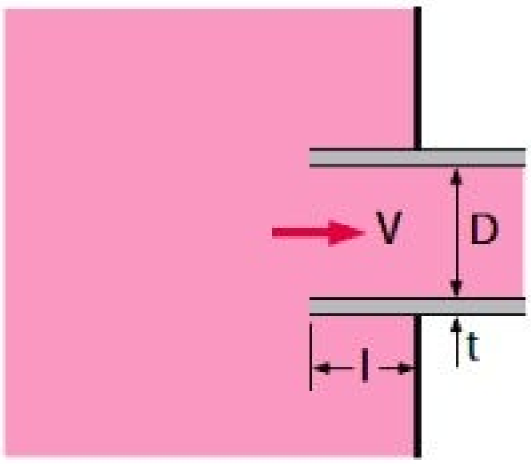
Table A.14 Loss coefficients for pipe fittings

	Screwed				Flanged				
Nominal Diameter (in.)	½	1	2	4	1	2	4	8	20
Valves (FO)									
Globe	14	8.2	6.9	5.7	13	8.5	6.0	5.8	5.5
Gate	0.30	0.24	0.16	0.11	0.80	0.35	0.16	0.07	0.03
Swing check	5.1	2.9	2.1	2.0	2.0	2.0	2.0	2.0	2.0
Angle	9.0	4.7	2.0	1.0	4.5	2.4	2.0	2.0	2.0
Ball valve[a]	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Gate valve[a]	¼C	½ C	¾ C						
	0.3	2.1	17						
Foot valve with strainer[a,b]	Poppet disk	Hinged disk							
	7	1.25							
Elbows									
45° regular	0.39	0.32	0.30	0.29					
45° long radius					0.21	0.20	0.19	0.16	0.14
90° regular	2.0	1.5	0.95	0.64	0.50	0.39	0.30	0.26	0.21
90° long radius	1.0	0.72	0.41	0.23	0.40	0.30	0.19	0.15	0.10
180° regular	2.0	1.5	0.95	0.64	0.41	0.35	0.30	0.25	0.20
180° long radius					0.40	0.30	0.21	0.15	0.10
Tees									
Line flow	0.90	0.90	0.90	0.90	0.24	0.19	0.14	0.10	0.07
Branch flow	2.4	1.8	1.4	1.1	1.0	0.80	0.64	0.58	0.41
Expansion[c]	*d/D*	*d/D*	*d/D*	*d/D*					
	0.2	0.4	0.6	0.8					
$K_{expansion}$	0.30	0.25	0.15	0.10					
Contraction[c]:	60° contraction angle								
	0.07								

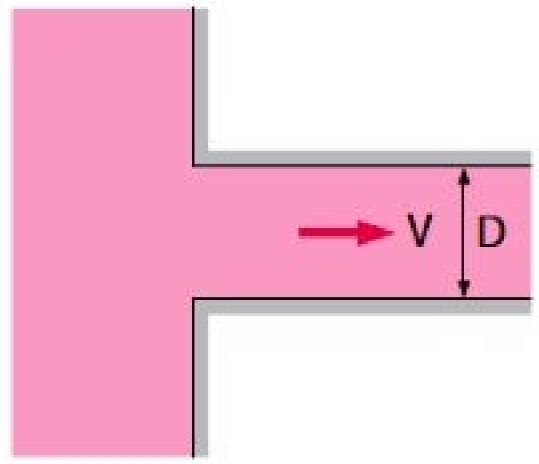
Loss coefficients K_L of various pipe components for turbulent flow (for use in the relation $h_L = K_L V^2/(2g)$, where V is the average velocity in the pipe that contains the component)*

Pipe Inlet

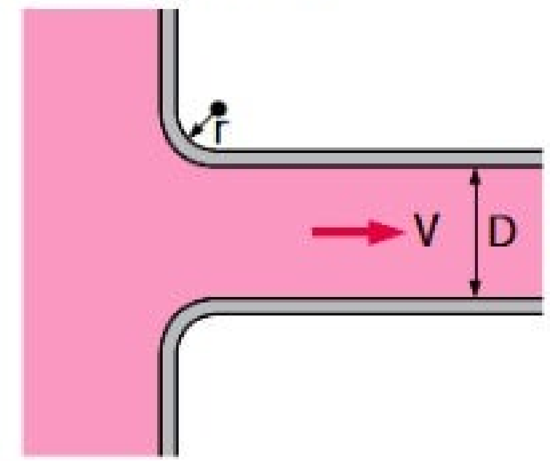
Reentrant: $K_L = 0.80$
($t \ll D$ and $l \approx 0.1D$)



Sharp-edged: $K_L = 0.50$

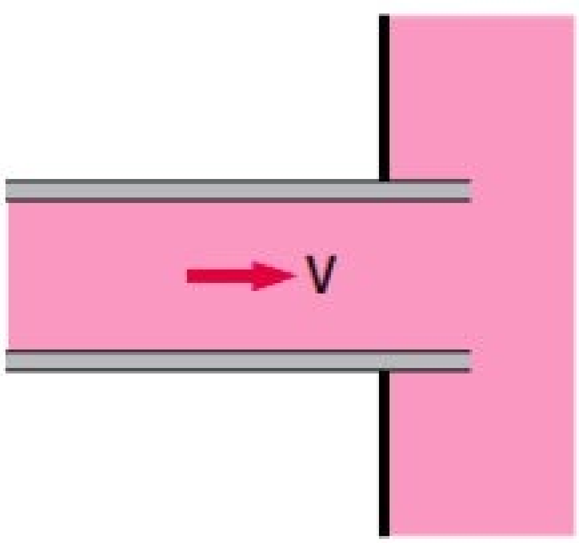


Well-rounded ($r/D > 0.2$): $K_L = 0.03$
Slightly rounded ($r/D = 0.1$): $K_L = 0.12$
(see Fig. 8–36)

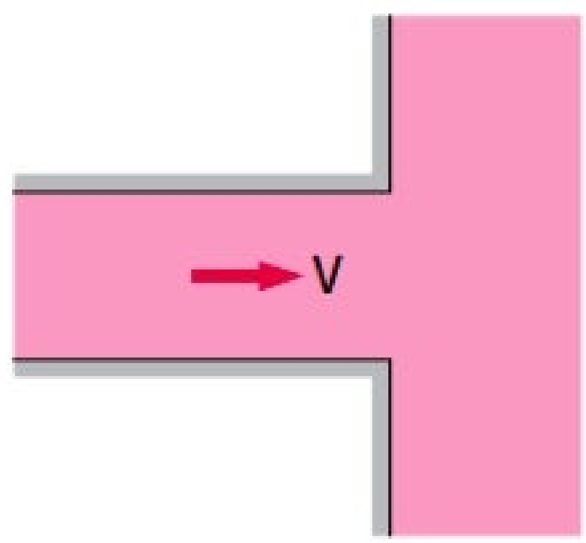


Pipe Exit

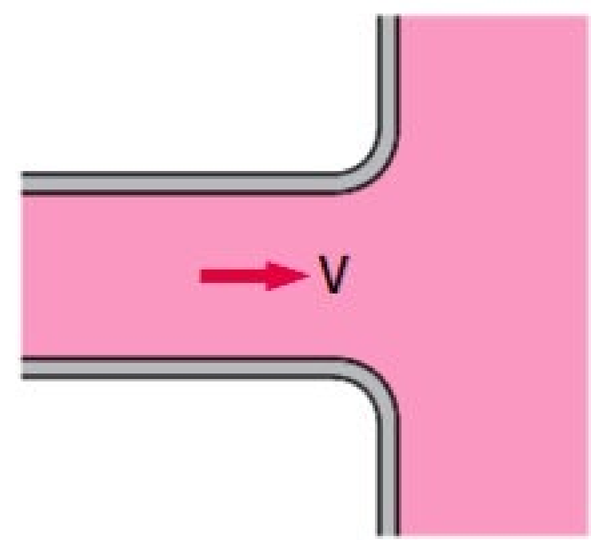
Reentrant: $K_L = \alpha$



Sharp-edged: $K_L = \alpha$



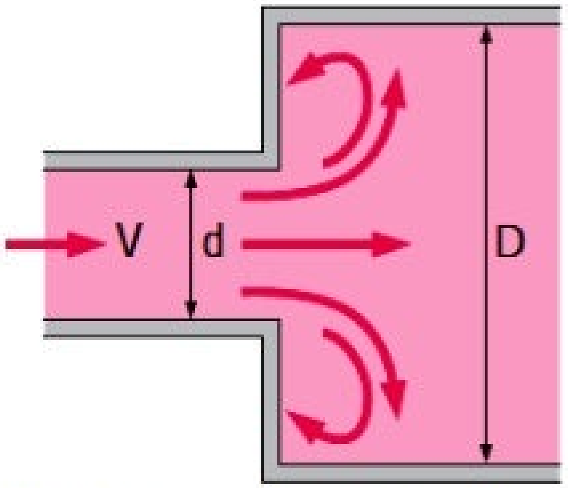
Rounded: $K_L = \alpha$



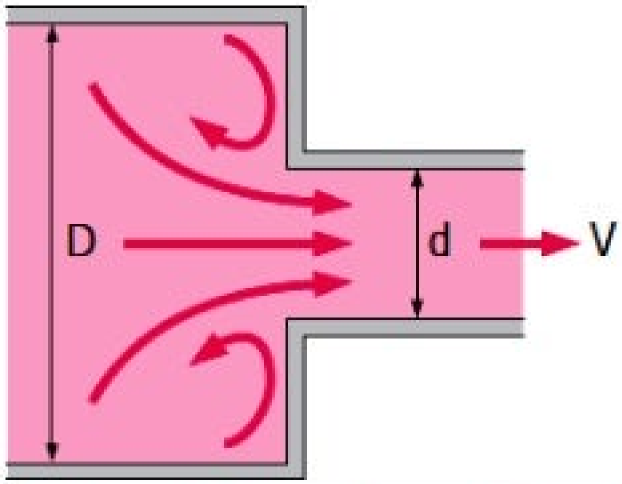
Note: The kinetic energy correction factor is $\alpha = 2$ for fully developed laminar flow, and $\alpha \approx 1$ for fully developed turbulent flow.

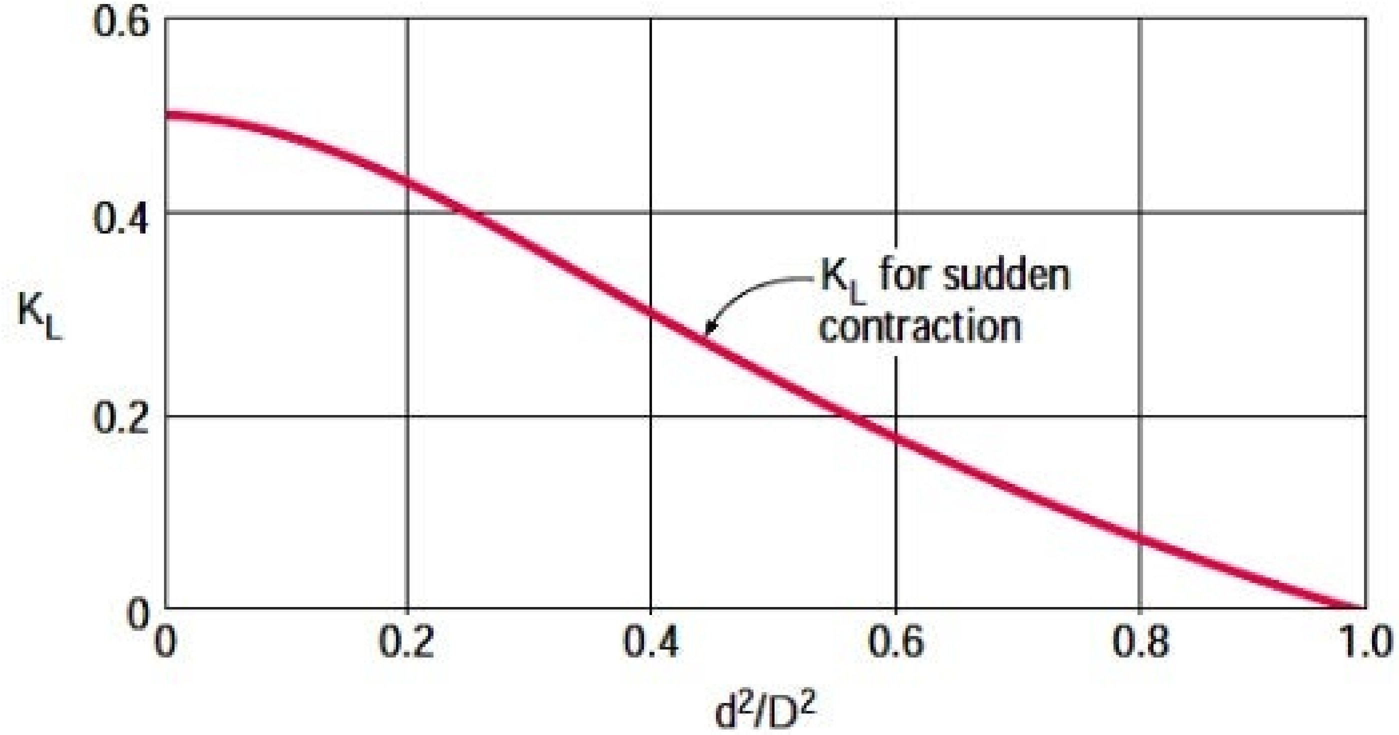
Sudden Expansion and Contraction (based on the velocity in the smaller-diameter pipe)

Sudden expansion: $K_L = \left(1 - \dfrac{d^2}{D^2}\right)^2$



Sudden contraction: See chart.





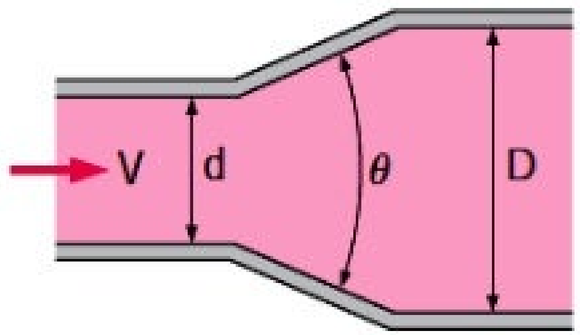
Gradual Expansion and Contraction (based on the velocity in the smaller-diameter pipe)

Expansion:

$K_L = 0.02$ for $\theta = 20°$

$K_L = 0.04$ for $\theta = 45°$

$K_L = 0.07$ for $\theta = 60°$



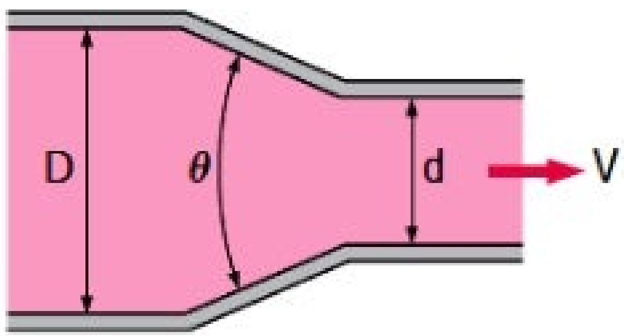
Contraction (for $\theta = 20°$):

$K_L = 0.30$ for $d/D = 0.2$

$K_L = 0.25$ for $d/D = 0.4$

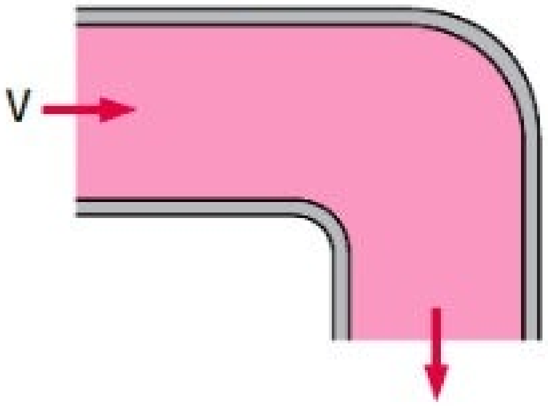
$K_L = 0.15$ for $d/D = 0.6$

$K_L = 0.10$ for $d/D = 0.8$

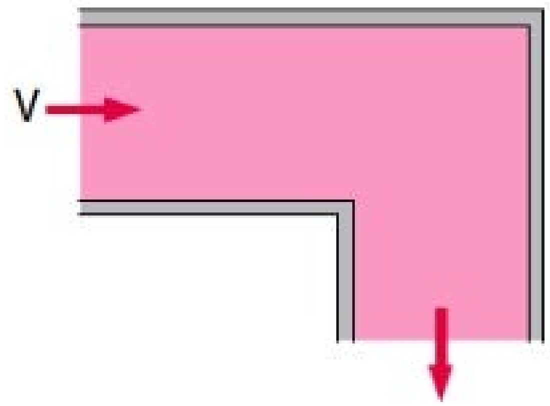


Bends and Branches

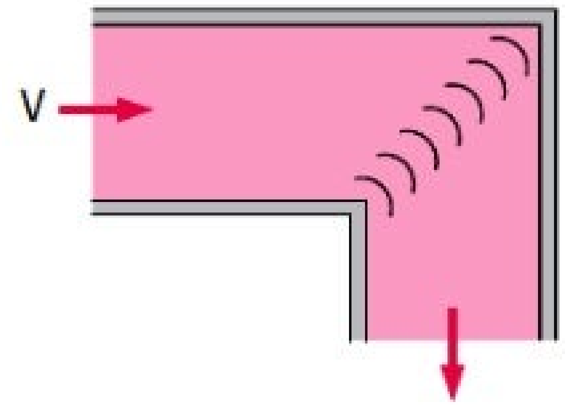
90° smooth bend:
Flanged: $K_L = 0.3$
Threaded: $K_L = 0.9$



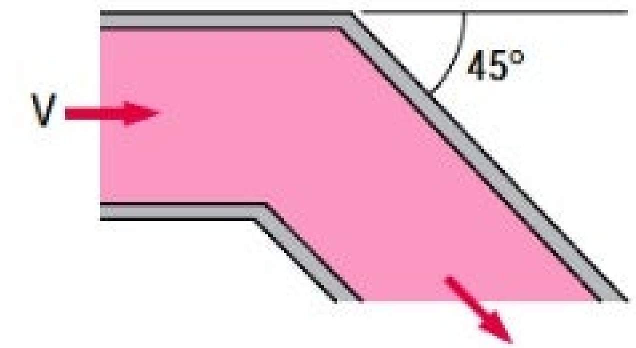
90° miter bend
(without vanes): $K_L = 1.1$



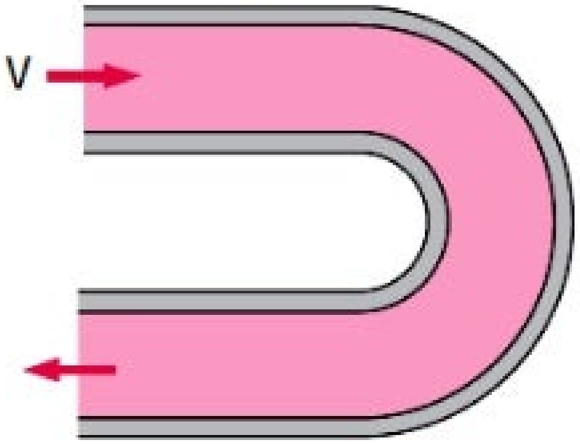
90° miter bend
(with vanes): $K_L = 0.2$



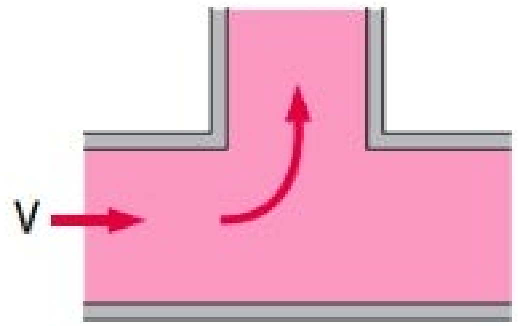
45° threaded elbow:
$K_L = 0.4$



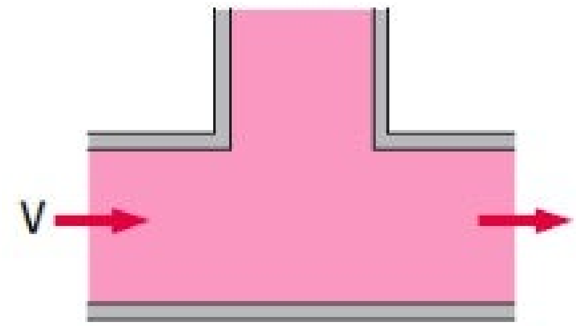
180° return bend:
Flanged: $K_L = 0.2$
Threaded: $K_L = 1.5$



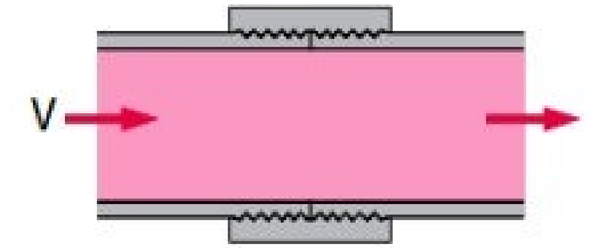
Tee (branch flow):
Flanged: $K_L = 1.0$
Threaded: $K_L = 2.0$



Tee (line flow):
Flanged: $K_L = 0.2$
Threaded: $K_L = 0.9$



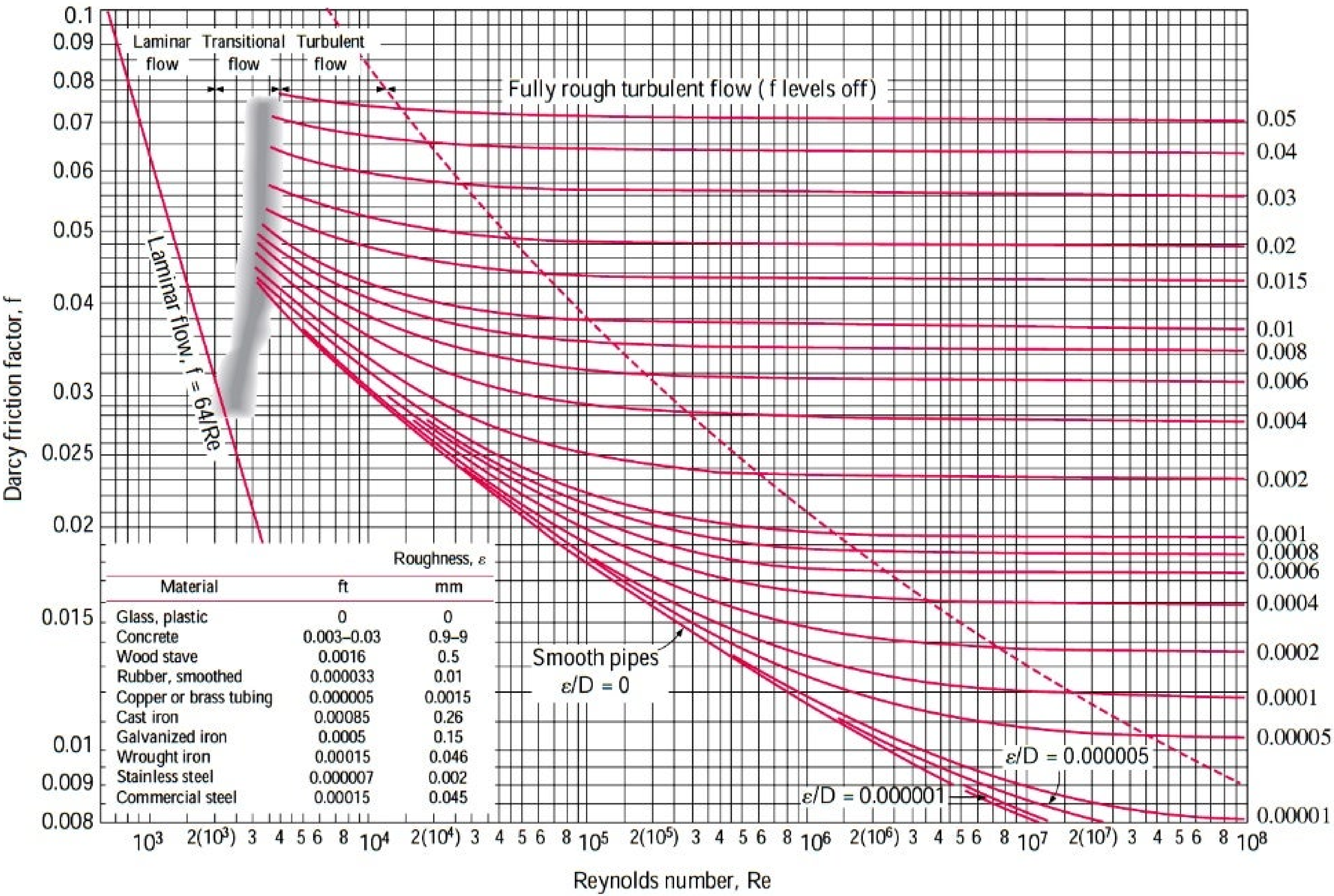
Threaded union:
$K_L = 0.08$



Valves

Globe valve, fully open: $K_L = 10$
Angle valve, fully open: $K_L = 5$
Ball valve, fully open: $K_L = 0.05$
Swing check valve: $K_L = 2$

Gate valve, fully open: $K_L = 0.2$
$\frac{1}{4}$ closed: $K_L = 0.3$
$\frac{1}{2}$ closed: $K_L = 2.1$
$\frac{3}{4}$ closed: $K_L = 17$



	Roughness, ε	
Material	ft	mm
Glass, plastic	0	0
Concrete	0.003–0.03	0.9–9
Wood stave	0.0016	0.5
Rubber, smoothed	0.000033	0.01
Copper or brass tubing	0.000005	0.0015
Cast iron	0.00085	0.26
Galvanized iron	0.0005	0.15
Wrought iron	0.00015	0.046
Stainless steel	0.000007	0.002
Commercial steel	0.00015	0.045

The Moody Chart

Pump Scaling Laws: Nondimensional Pump Parameters

- Dimensional analysis, the Buckingham Pi (π) theorem, and the method of repeating variables can be used to generate **nondimensional pump parameters** that include the pump head, pump capacity, and pump bhp.
- For a typical pump, the following functional forms will apply:

$$gH_{\text{pump}} = f\left(\dot{V}, D, \varepsilon, \omega, \rho, \mu\right)$$
$$\text{bhp} = f\left(\dot{V}, D, \varepsilon, \omega, \rho, \mu\right),$$

- where ε is the relative roughness and D is the diameter of the pump impeller. The resulting dimensionless π groups produce nondimensional pump parameters with special names. Following are the six nondimensional pump parameters of interest:

$$\text{(a)} \quad \frac{\rho\omega D^2}{\mu} = \text{Re} = \text{Reynold's number},$$

- where ωD is the characteristic velocity.

Pump Scaling Laws: Nondimensional Pump Parameters

(b) $\dfrac{\varepsilon}{D}$ = Relative roughness

(c) $\dfrac{gH_{\text{pump}}}{\omega^2 D^2} = C_H$ = Head coefficient

(d) $\dfrac{\dot{V}}{\omega D^3} = C_Q$ = Capacity coefficient

(e) $\dfrac{\text{bhp}}{\rho \omega^3 D^5} = C_P$ = Power coefficient

(f) $\dfrac{g \times \text{NPSHR}}{\omega^2 D^2} = C_{\text{NPSH}}$ = Suction head coefficient.

Application of the Nondimensional Pump Parameters—Affinity Laws

- Pumps that are geometrically similar may considered to be equivalent. **Pump equivalence** implies that the nondimensional pump parameters are equal for different pumps or for different states of the same pump.

Consider two pumps (pump A and pump B) or consider a pump at two different states of volume flow rate, speed, pump head, bhp, etc. The pump equivalence equations are:

$$\text{(a)} \quad \frac{\dot{V}_A}{\omega_A D_A^3} = \frac{\dot{V}_B}{\omega_B D_B^3} = C_{Q,A} = C_{Q,B}$$

$$\text{(b)} \quad \frac{\rho_A \omega_A D_A^2}{\mu_A} = \frac{\rho_B \omega_B D_B^2}{\mu_B} = \mathrm{Re}_A = \mathrm{Re}_B$$

$$\text{(c)} \quad \frac{\varepsilon_A}{D_A} = \frac{\varepsilon_B}{D_B}$$

$$\text{(d)} \quad \frac{g H_{\text{pump},A}}{\omega_A^2 D_A^2} = \frac{g H_{\text{pump},B}}{\omega_B^2 D_B^2} = C_{H,A} = C_{H,B}$$

$$\text{(e)} \quad \frac{\text{bhp}_A}{\rho_A \omega_A^3 D_A^5} = \frac{\text{bhp}_B}{\rho_B \omega_B^3 D_B^5} = C_{P,A} = C_{P,B}.$$

Application of the Nondimensional Pump Parameters—Affinity Laws

- The pump equivalence equations can be expressed as ratios in order to scale pumps of different sizes or determine the parameters at different states for the same pump.
- Thus, for different pumps,

$$\text{(a)} \quad \frac{\dot{V}_B}{\dot{V}_A} = \frac{\omega_B}{\omega_A}\left(\frac{D_B}{D_A}\right)^3$$

$$\text{(b)} \quad \frac{H_B}{H_A} = \left(\frac{\omega_B}{\omega_A}\right)^2\left(\frac{D_B}{D_A}\right)^2$$

$$\text{(c)} \quad \frac{\text{bhp}_B}{\text{bhp}_A} = \frac{\rho_B}{\rho_A}\left(\frac{\omega_B}{\omega_A}\right)^3\left(\frac{D_B}{D_A}\right)^5.$$

- The aforementioned equations are known as the **affinity laws**. The affinity laws also apply to a given pump moving the same liquid. For this case, $\rho_A = \rho_B$ and $D_A = D_B$. In general, $\omega = 2\pi N$, where N is the number of revolutions per minute of the pump impeller

Application of the Nondimensional Pump Parameters—Affinity Laws

- Hence, for the same pump that has experienced a speed change from N_A to N_B,

$$\text{(a)} \quad \frac{\dot{V}_B}{\dot{V}_A} = \frac{N_B}{N_A}$$

$$\text{(b)} \quad \frac{H_B}{H_A} = \left(\frac{N_B}{N_A}\right)^2$$

$$\text{(c)} \quad \frac{\text{bhp}_B}{\text{bhp}_A} = \left(\frac{N_B}{N_A}\right)^3.$$

Nondimensional Form of the Pump Efficiency

- Dimensional analysis can also be used to establish a nondimensional form of the pump efficiency. Consider the following analysis:

$$\eta_{\text{pump}} = \frac{\dot{W}_{\text{water horsepower}}}{\text{bhp}} = \frac{\rho g \dot{V} H_{\text{pump}}}{\text{bhp}}$$

Remember: $$\frac{gH_{\text{pump}}}{\omega^2 D^2} = C_{\text{H}}, \quad \frac{\dot{V}}{\omega D^3} = C_{\text{Q}}, \quad \frac{\text{bhp}}{\rho \omega^3 D^5} = C_{\text{P}}$$

Therefore,

$$\eta_{\text{pump}} = \frac{\rho \left(\omega D^3 C_{\text{Q}}\right)\left(\omega^2 D^2 C_{\text{H}}\right)}{\rho \omega^3 D^5 C_{\text{P}}} = \frac{C_{\text{Q}} C_{\text{H}}}{C_{\text{P}}}.$$

Example. Manipulating the Affinity Laws

- When operated at $N = 1170$ rpm, a centrifugal pump, with impeller diameter, $D = 8$ in., has shut-off head $H_o = 25.0$ ft of water. At the same operating speed, best efficiency occurs at $\dot{V} = 300$ gpm, where the head is $H = 21.9$ ft of water. Specify the discharge and head for the pump when it is operated at 1750 rpm at both the shut-off and BEPs.
- **Solution.** The pump remains the same, so the two flow conditions are geometrically similar.
- If no cavitation occurs, the flows will also be kinematically similar. Let condition 1 be at 1170 rpm and condition 2 be at 1750 rpm. For the flow rates (discharge),

$$C_{Q1} = C_{Q2}$$

$$\frac{\dot{V}_1}{\omega_1 D_1^3} = \frac{\dot{V}_2}{\omega_2 D_2^3}$$

Example. Manipulating the Affinity Laws

Note that $\omega = 2\pi N$. Thus,

$$\frac{\dot{V}_1}{2\pi N_1 D_1^3} = \frac{\dot{V}_2}{2\pi N_2 D_2^3}.$$

Since $D_1 = D_2$,

$$\frac{\dot{V}_1}{N_1} = \frac{\dot{V}_2}{N_2}$$

$$\dot{V}_2 = N_2 \frac{\dot{V}_1}{N_1}.$$

At the shut-off point,

$$\dot{V}_2 = 1750\,\text{rpm}\,\frac{0\,\text{gpm}}{1170\,\text{rpm}}$$

$$\dot{V}_2 = 0\,\text{gpm}.$$

Example. Manipulating the Affinity Laws

At the BEP:

$$\dot{V}_2 = 1750\ \text{rpm}\frac{300\ \text{gpm}}{1170\ \text{rpm}}$$

$$\dot{V}_2 = 449\ \text{gpm}.$$

For the head,

$$C_{H1} = C_{H2}$$

$$\frac{gH_1}{\omega_1^2 D_1^2} = \frac{gH_2}{\omega_2^2 D_2^2}.$$

Note that $\omega = 2\pi N$. Thus,

$$\frac{gH_1}{(2\pi N)_1^2\, D_1^2} = \frac{gH_2}{(2\pi N)_2^2\, D_2^2}.$$

Example. Manipulating the Affinity Laws

Since $D_1 = D_2$,

$$\frac{H_1}{N_1^2} = \frac{H_2}{N_2^2}$$

$$H_2 = H_1 \left(\frac{N_2}{N_1}\right)^2.$$

At the shut-off point,

$$H_2 = 25.0\text{ ft}\left(\frac{1750\text{ rpm}}{1170\text{ rpm}}\right)^2$$
$$H_2 = 55.9\text{ ft}.$$

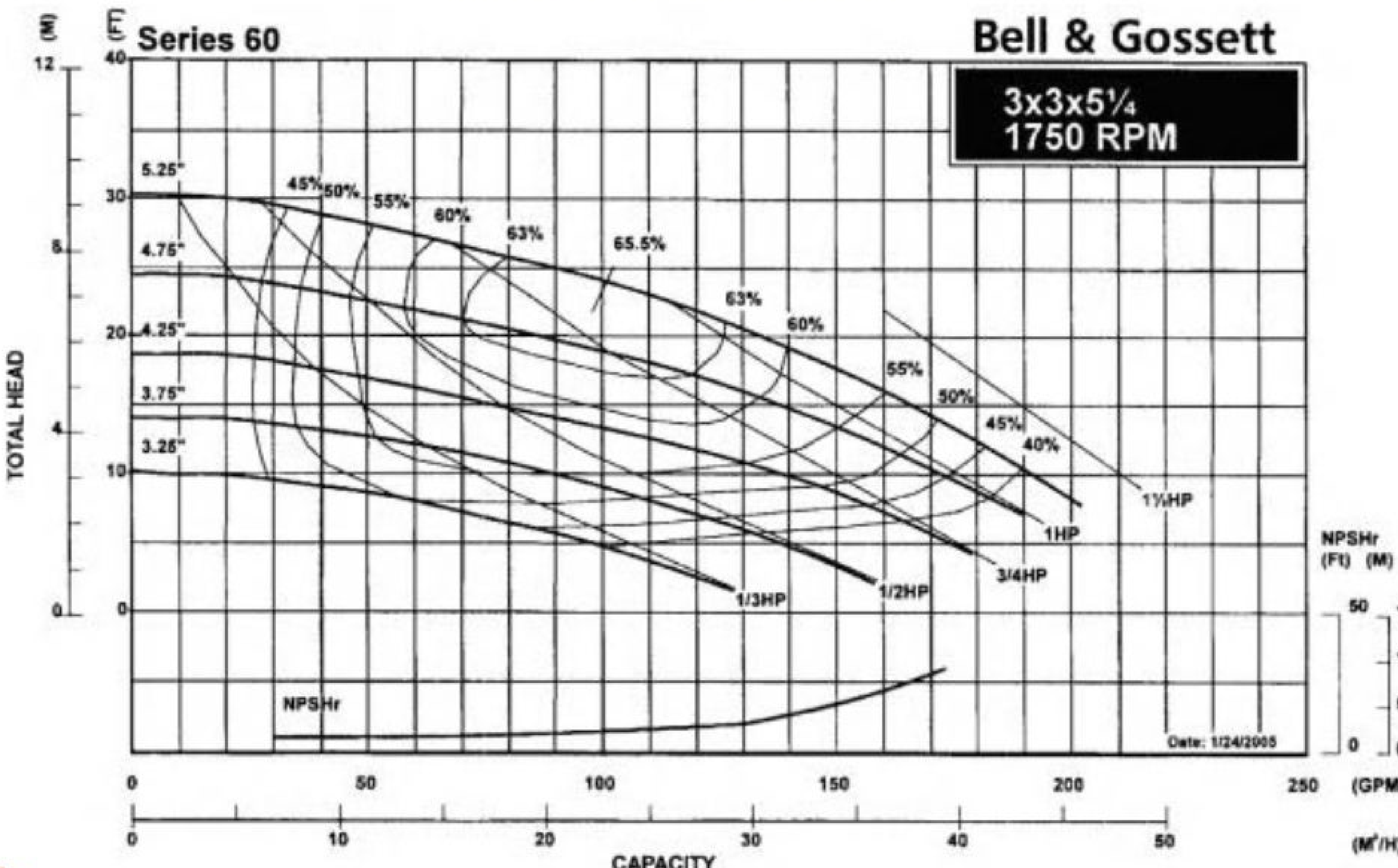
At the BEP,

$$H_2 = 21.9\text{ ft}\left(\frac{1750\text{ rpm}}{1170\text{ rpm}}\right)^2$$
$$H_2 = 49.0\text{ ft}.$$

Example. Selecting a Pump Using Equivalent Performance Plots

- Bell & Gossett manufacture 3 in. × 3 in. × $5^1/_4$ in. pumps that operate at 1750 rpm. Unfortunately, a design project requires a 1750 rpm pump that would provide 70 gpm of kerosene for a total head of 10 ft of kerosene, if it were to operate at 1150 rpm. Select an appropriate pump from the available Bell & Gossett performance curves. Prepare a pump schedule for the purposes of installation by the mechanical contractor.
- **Solution.** The performance plot for the Bell & Gossett Series 60 centrifugal pump 3 in. × 3 in. × $5^1/_4$ in. family of pumps is shown below.

Example. Selecting a Pump Using Equivalent Performance Plots



Example. Selecting a Pump Using Equivalent Performance Plots

- Only the performance curves for the 1750 rpm family of pumps are available. The plots are based on clean, pure water. The affinity laws and available data will be used to select a 1750 rpm pump that would give the same performance as an 1150 rpm pump.
- The affinity law for the volume flow rate is

$$\frac{\dot{V}_B}{\dot{V}_A} = \frac{\omega_B}{\omega_A}\left(\frac{D_B}{D_A}\right)^3.$$

The impeller diameter is constant. Hence,

$$\frac{\dot{V}_B}{\dot{V}_A} = \frac{N_B}{N_A}.$$

- Let state "A" correspond to the 1150 rpm speed. The flow rate required at this speed is 70 gpm.

Example. Selecting a Pump Using Equivalent Performance Plots

$$\dot{V}_B = \dot{V}_A \frac{N_B}{N_A}$$

$$\dot{V}_B = (70 \text{ gpm}) \frac{1750 \text{ rpm}}{1150 \text{ rpm}} = 107 \text{ gpm}.$$

- 107 gpm would be produced in the 1750 rpm pump.
- The pump head values in the Bell & Gossett pump performance plots are in terms of foot of water. The total head in terms of foot of kerosene must be converted to foot of water.
- Therefore,

$$H_A = 10 \text{ ft kerosene} \times SG_{\text{kerosene}} = 10 \text{ ft kerosene} \times \frac{51.2 \text{ lb/ft}^3}{62.4 \text{ lb/ft}^3} = 8.2 \text{ ft water}.$$

- The total pump head required for the 1150 rpm speed is 8.2 ft of water. At the 1750 rpm speed, the total pump head is found from the affinity law for the pump head:

Example. Selecting a Pump Using Equivalent Performance Plots

$$\frac{H_B}{H_A} = \left(\frac{\omega_B}{\omega_A}\right)^2 \left(\frac{D_B}{D_A}\right)^2.$$

For a constant impeller diameter,

$$H_B = H_A \left(\frac{N_B}{N_A}\right)^2$$

$$H_B = (8.2 \text{ ft water}) \left(\frac{1750 \text{ rpm}}{1150 \text{ rpm}}\right)^2 = 19 \text{ ft of water.}$$

- This head corresponds to the total pump head required for the pump operating at 1750 rpm.

The pump performance plot can be used to select a 1750 rpm pump to deliver 107 gpm of kerosene at 19 ft of water. According to the plot, a *4.85-in. diameter impeller* would be satisfactory. Note that the manufacturer could trim a 5.25 in. impeller down to 4.85 in. on a lathe. Since that option may be costly, or cannot be accomplished by the manufacturer, choose a 5.25-in. diameter impeller.

The bhp required to drive the 1750 rpm pump is 1 hp.

Therefore, the final choice is a pump with the following operating parameters:

3 in. × 3 in. × $5^1/_4$ in. pump casing
5.25 in. impeller diameter
1-hp motor
1750 rpm

The pump schedule is shown below.

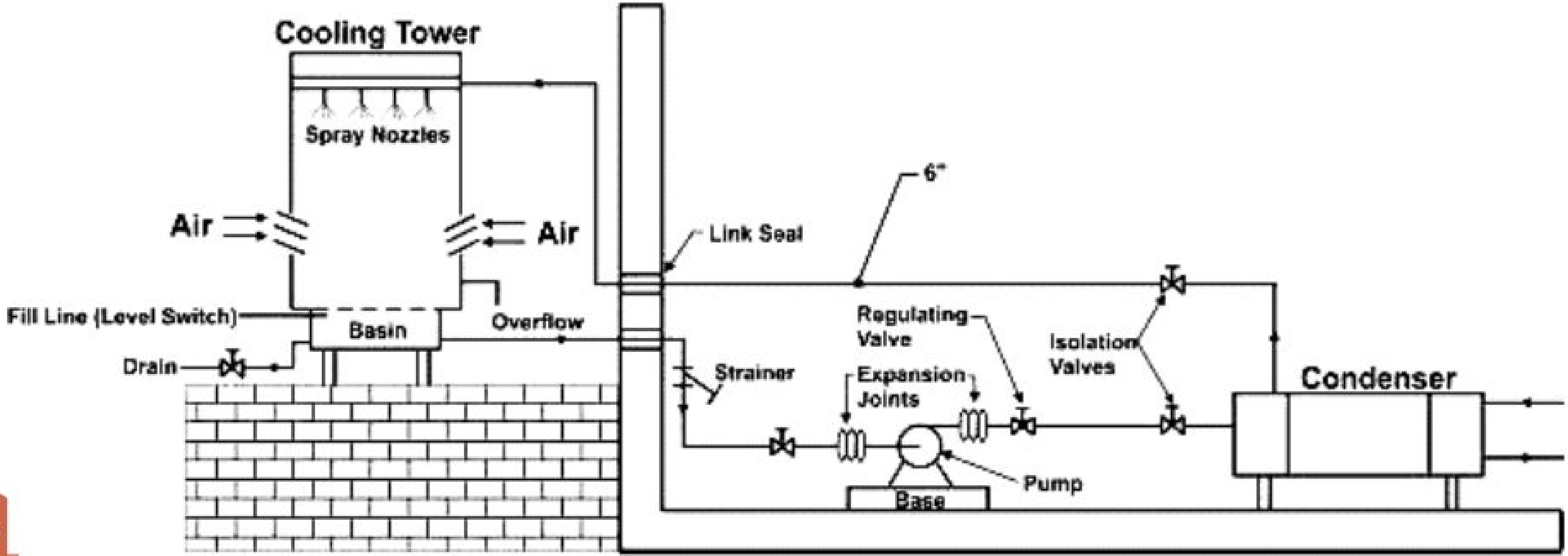
Pump Schedule

				Fluid			Electrical		
Tag	Manufacturer and Model Number	Type	Construction	Flow Rate (gpm)	Working Fluid	Head Loss (ft)	Motor Size (hp)	Motor Speed (rpm)	V/Ph/ Hz
P-1	Bell & Gossett Series 60, or Equal	Centrifugal, In-line Mounted	Iron 3 × 3 × $5^1/_4$ in. Casing, 5.25 in. ϕ	107	Kerosene	19	1	1750	208/3/ 60

Design of Piping Systems Complete with In-Line or Base-Mounted Pumps

Open-Loop Piping System

- In an **open-loop piping system**, some part of the circuit is open to the atmosphere.
- For example, a drain may be open to the atmosphere or the piping circuit may be open to the atmosphere as are typical of cooling towers. Figure below shows a typical open-loop condenser piping system for water.



Design of Piping Systems Complete with In-Line or Base-Mounted Pumps

Note the following regarding typical open-loop piping systems:

a) A strainer (filter in liquid piping systems) is required to protect the pump from large sediments and potential blockage.

b) Isolation valves may be installed around equipment in the piping system. These will allow for maintenance without the need for complete drainage of the piping system. These valves could be ball valves, globe valves, or gate valves.

c) Regulating valves are used to control the flow rate, especially on the discharge line from the pump. These valves are typically globe valves. This type of flow rate regulation is known as **discharge throttling**.

d) Expansion (flexible connectors) joints are required to protect the pipes from expansion and contraction forces due to pumping thermal stress. They also isolate the piping from pump vibrations.

Practical Note 6. Bypass Lines

- Not shown in the schematic are **bypass lines**. These are additional sections of piping that could be installed around equipment in the system to divert all or a part of the flow. The bypass line may include an auxiliary piece of equipment that is normally brought on-line during maintenance of the primary equipment. Isolation valves should be installed at the entrance and exit of the bypass line.

Practical Note 7. Regulation and Control of Flow Rate across a Pump

- Control of the flow rate across a pump can be accomplished by varying the pump head, speed, or both simultaneously. While there are many different methods of regulating the fluid flow rate, only three of the methods will be considered.
- **Discharge throttling** involves the use of a partially closed valve installed on the discharge line of the pump. In this case, the system curve intersects the pump performance curve (operating point) at a higher pump head and lower flow rate. However, this will produce lower pump efficiencies. This is the cheapest and most common method of flow control
- **Bypass regulation** occurs with the use of a bypass line. Diversion of a portion of the flow will result in a decrease in the amount of power required. For this reason, it may be preferred to discharge throttling.
- **Speed regulation** can be used to control the flow rate by varying the speed of the pump. Some options include variable-speed mechanical drives or variable frequency drives (VFD) on motors. By changing the pump speed, the power requirement varies, without adverse impact on the efficiency of the pump.

Practical Note 8. In-Line and Base-Mounted Pumps

- Pumps in piping systems can either be mounted in-line with the piping system and supported by the pipes and brackets. Or, the pumps could be **base mounted** on concrete pads.
- In the case of base-mounted pumps, **vibration isolators** or **vibration isolation pads** should be specified. Isolators are typically installed between the pump and the concrete pad. This will reduce the transmission of vibration and noise to the main building structure.
- If vibration isolators are not desired, then the design engineer should ensure that the pump's concrete foundation is **$1^1/_2$–3 times** the total weight of the pump and motor assembly to provide sufficient vibration isolation and noise control.

Example. Designing an Open-Loop Piping System

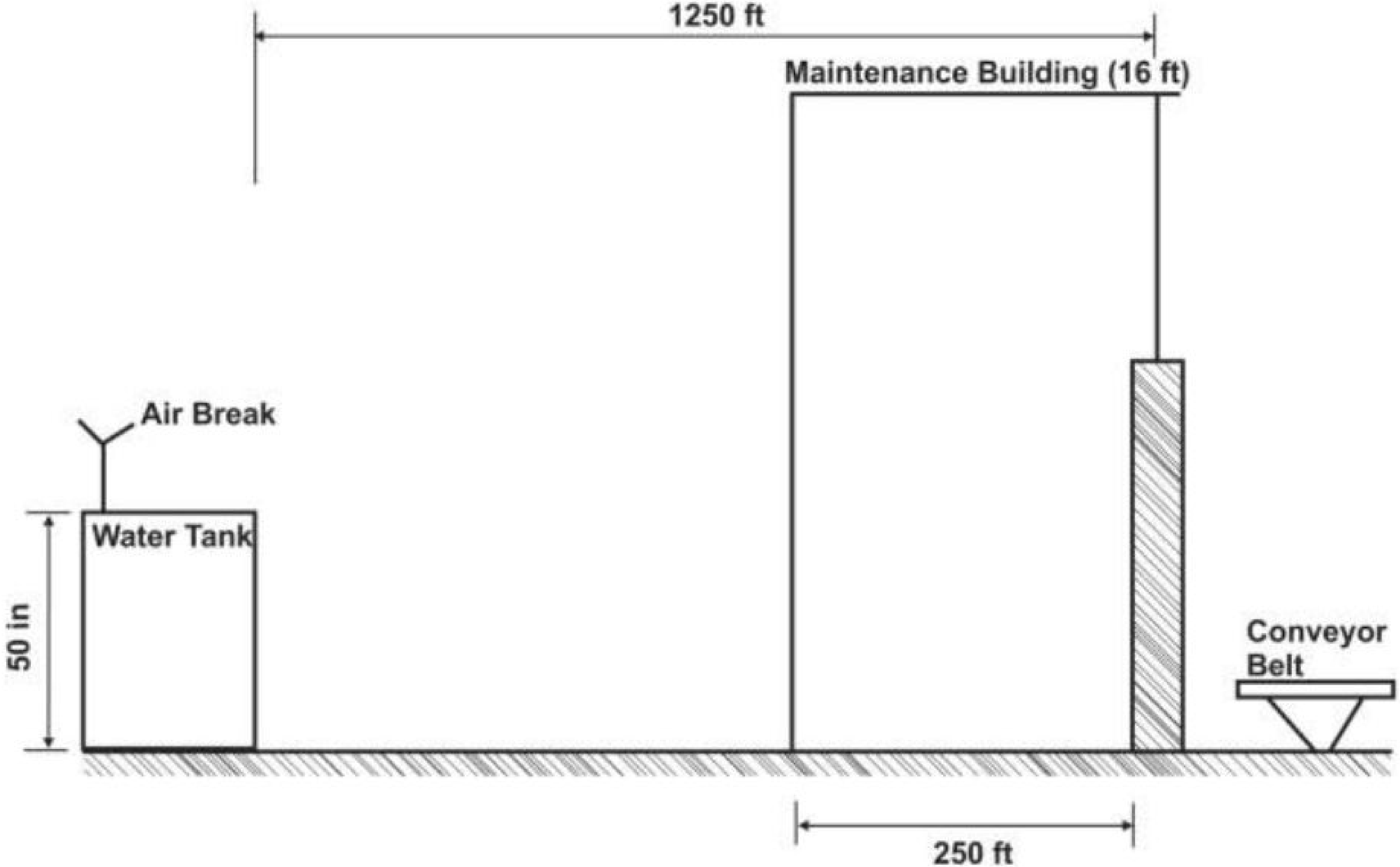
- Metal Mint, Inc. (located in Southern Mexico) has contracted the services of EBA Engineering Consultants, Ltd. to design a cooling system for their high-quality stainless steel bullion bars. The hot stainless steel bars will be transported slowly on a conveyor belt to allow for cooling. A design engineer at EBA Engineering Consultants, Ltd. has suggested the use of water in an evaporative cooling system. They wish to transport water from an existing 1200-gallon tank, which has a height of 50 in., to a large nozzle located 10 ft above grade (the ground). Once transported to the nozzle (K_L = 14), the water will be atomized to small droplets to produce shower-like streams of water to enhance evaporative cooling. Given that water is scarce in this area, the subsequent design of the conveyor cooling system will be determined by the piping system design. A sketch was submitted by the design engineer for consideration:

Example. Designing an Open-Loop Piping System

a) Based on the sketch provided by the design engineer, design an appropriate piping system, complete with all necessary equipment, to transport the water to the atomization nozzle. Metal Mint, Inc. (the client) has an open business agreement with Bell & Gossett—ITT Industries. Equipment schedules, specifications, and installation guidelines are not required by the client.

b) For the convenience of the client, it is recommended that a low-level switch be installed in the tank. Based on the piping system design and equipment selected, conduct an analysis to determine and/or justify the minimum height of water in the tank (i.e., minimum height of the low-level switch in the tank).

Example. Designing an Open-Loop Piping System



Possible Solution

Detailed Design

Objective

To determine the size of a pipe to move water. An appropriate pump will also be sized and selected, with selection from a Bell & Gossett pump performance curve.

Data Given or Known

(i) The water tank can hold 1200 gallons of liquid. The height of the tank is 50 in.
(ii) The distance from the tank to the wall adjacent to the conveyor belt is 1250 ft.
(iii) The maintenance building has a length of 250 ft and a height of 16 ft.
(iv) The water nozzle has to be located 10 ft above the ground.
(v) The K_L value of the nozzle is 14.
(vi) The tank is complete with an air break (vent) to the atmosphere.

Assumptions/Limitations/Constraints

(i) Let the pipe material be Schedule 40 steel. It appears as though a portion of the piping system will be outside the building. The thicker Schedule 40 steel will provide protection to the pipe, and will be more durable than copper piping. However, it should be noted that the steel may rust due to the presence of dissolved air in this open-loop piping. This could cause blockage of the nozzle.

(ii) Let the flow velocity in the pipe be lower than 10 fps, which is the erosion limit for small steel pipes. For the suction lines of centrifugal pumps (Bell & Gossett pumps), the velocity should not be more than 5 fps. Hence, any pipe velocity lower than 5 fps will be acceptable.

(iii) Limit the frictional losses in the straight run of pipe to 3 ft wg per 100 ft of pipe. This is based on industry standards/guidelines.

(iv) Let all pipe changes be gradual to reduce losses in the system.

(v) Install threaded fittings. This is common for smaller pipes.

(vi) Let the pipe be connected at the bottom of the water tank, approximately 6 in from the ground. The height of the fluid in the tank will provide some pressure head during pumping.

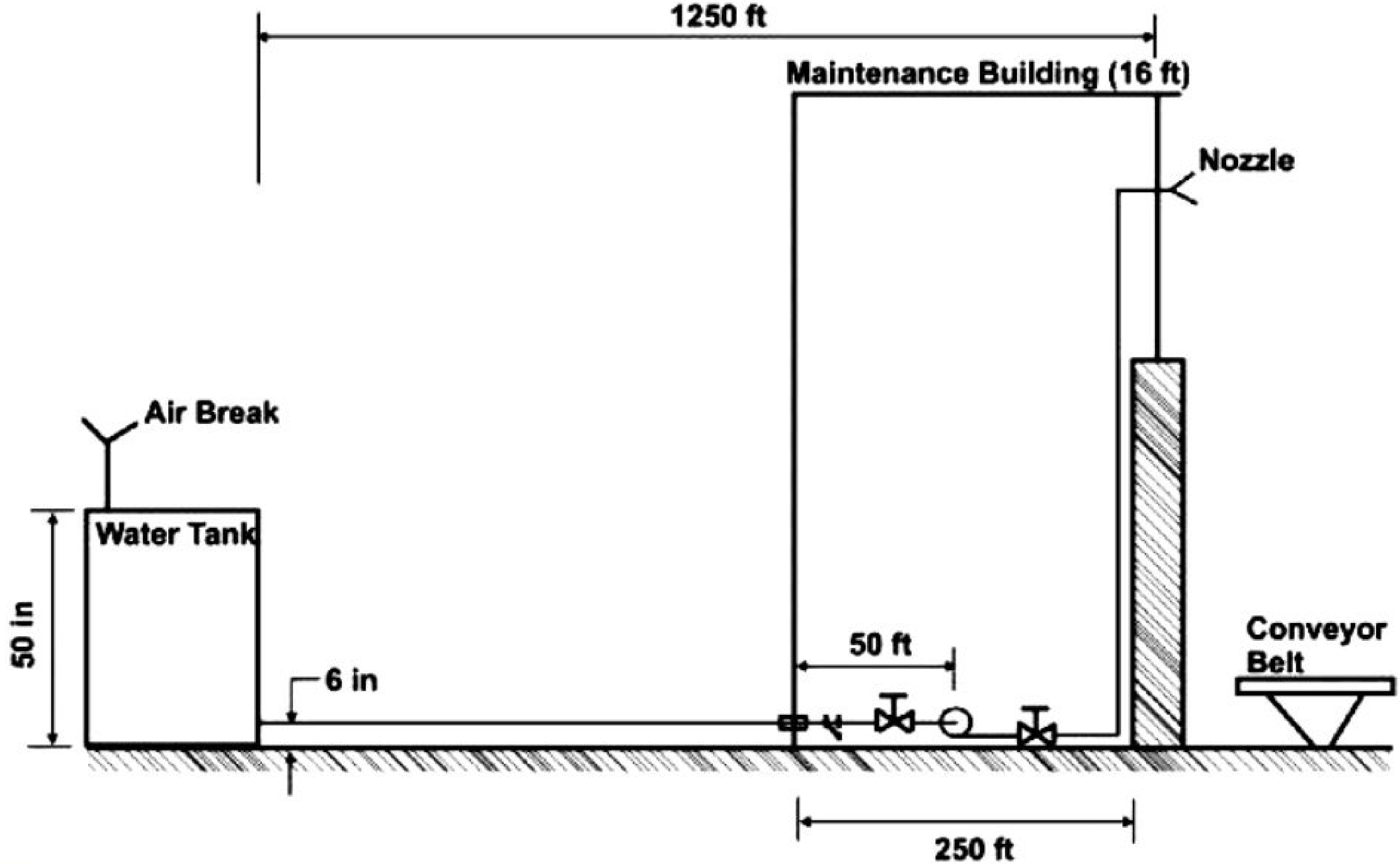
(vii) Isolation valves around the pump will be gate or globe valves. The globe valve should be installed on the pump discharge line to regulate flow, if required.

(viii) Assume that the pump is located in the maintenance building, 50 ft from the exterior wall closest to the water tank.

(ix) The water temperature is 70°F because the system is located in Southern Mexico.

Sketch

A sketch of the piping system, complete with valves and a pump, are shown.



Analysis

Pipe Sizing

The friction loss is fixed at 3 ft wg per 100 ft of pipe. The pipe velocity cannot exceed 5 fps. From the friction loss chart for Schedule 40 steel in open piping systems (Figure A.4),

Pipe diameter: 2 in. (nominal)
Pipe velocity: 3 fps
Flow rate: 30 gpm.

$$\text{Reynolds number}: \frac{\rho V D_{\text{inner}}}{\mu} = \frac{(62.30\ \text{lb/ft}^3)(3.0\ \text{ft/s})(2.067\ \text{in.})}{6.556 \times 10^{-4}\ \text{lb/ft/s}} \times \frac{1\ \text{ft}}{12\ \text{in.}} = 49105.$$

The flow is fully turbulent.

Pump Sizing

The pump head is required to size the pump. The energy equation is

$$H_{\text{pump}} = \left(\frac{p_2}{\rho g} + \frac{V_2^2}{2g} + z_2\right) - \left(\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1\right) + H_{\text{lT}}.$$

This is an open-loop piping system. Let point 1 be at the entrance of the pipe attached to the water tank and point 2 be at the exit of the nozzle. To select the largest possible pump (most conservative design), assume that the water level in the water tank is 6 in. above grade. Thus, $z_1 = 6$ in. The air break on the tank ensures that the free surface of the water reservoir remains at normal atmospheric pressure. Since the free surface is level with the pipe entrance, $p_1 = p_{\text{atm}}$. At the exit of the nozzle, $p_2 = p_{\text{atm}}$. Changes in the velocity will be assumed to be negligible since the pipe diameter is constant. So, $V_2 \approx V_1$.

Therefore,

$$H_{\text{pump}} = (z_2 - z_1) + H_{\text{lT}}.$$

The total head loss in the pipe is

$$H_{\text{lT}} = H_{\text{l}} + H_{\text{lm}} = H_{\text{l}} + \sum K_{\text{L}} \frac{V^2}{2g}.$$

For the 2 in. pipe, the loss coefficient values are:

For the pipe entrance: $K_{\text{L}} = 0.5$ (sharp-edged entrance)
For the strainer: $K_{\text{L}} = 1.5$
For the gate valve: $K_{\text{L}} = 0.16$
For the globe valve: $K_{\text{L}} = 6.9$
For the 90° elbows: $K_{\text{L}} = 0.95$ (regular elbows)
For the nozzle: $K_{\text{L}} = 14$.

The total length of piping is approximately (1250 + 10–0.5) ft = 1260 ft.
Thus,

$$H_{IT} = \frac{3.0 \text{ ft wg}}{100 \text{ ft}} \times 1260 \text{ ft} + [0.5 + 1.5 + 0.16 + 6.9 + 2\,(0.95) + 14]\,\frac{(3.0 \text{ ft/s})^2}{2(32.2 \text{ ft/s}^2)}$$

$$H_{IT} = 41 \text{ ft.}$$

Hence,

$$H_{pump} = (10 - 0.5)\,\text{ft} + 41 \text{ ft}$$

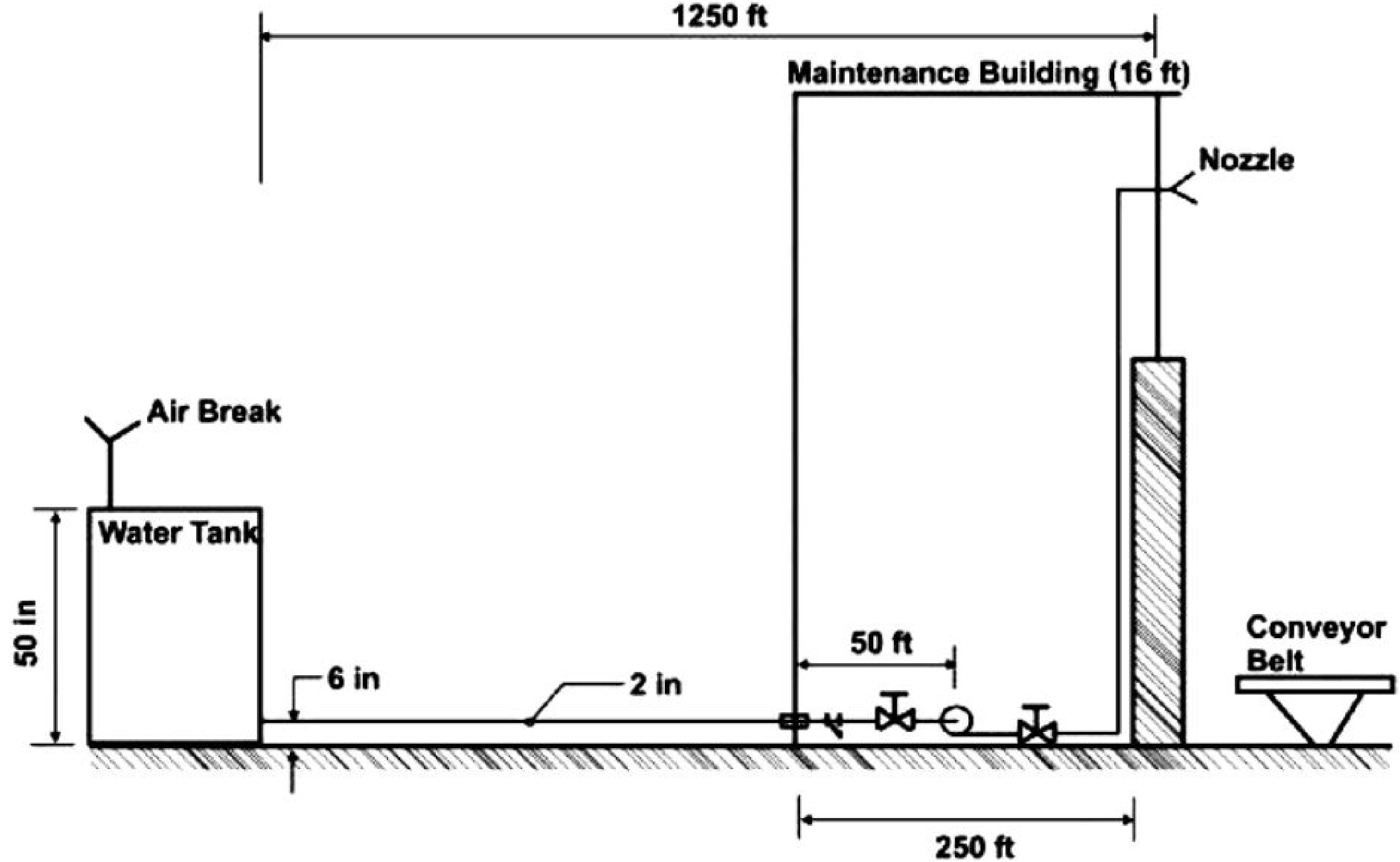
$$H_{pump} = 50.5 \text{ ft} \approx 51 \text{ ft.}$$

The flow rate is 30 gpm. From the performance plots for the Bell & Gossett Series 60 in-line mounted centrifugal pumps, select the following pump:

$1\frac{1}{2} \times 1\frac{1}{2} \times 7$ in. casing
7.0 in. impeller diameter
$1\frac{1}{2}$-hp motor
1750 rpm

Or, choose Bell & Gossett Series 60 pump **Model 618T**, which is in stock.

Drawings


1250 ft
Maintenance Building (16 ft)
Nozzle
Air Break
Water Tank
50 in
6 in
2 in
50 ft
Conveyor Belt
250 ft

Conclusion

The piping system and pump has been sized and designed. The following points should be noted:

(i) A gate and globe valve are used as isolation valves. The globe valve on the discharge line of the pump may be used to control the flow rate, if needed. Another option could be to use a VFD to control the pump via the motor.

(ii) If the water tank is not available as a packaged unit, complete with a valve on the discharge line, a gate valve may be required at the tank discharge. In any case, this would add a small additional loss to system.

(iii) Though negligible, there will be changes in the velocity between points 1 and 2, since the diameter of the atomized fluid jets at the nozzle will be smaller than the pipe diameter.

(iv) The pump was slightly oversized to avoid overloading the motor.

(v) It should be noted that a 2 in. × $1\frac{1}{2}$ in. reducing fitting will be required on the pump suction and discharge for installation of the pump.

(vi) The pump suction velocity should not be much larger than 5 fps. In this case, the velocity will be about 3 fps.

(vii) As requested by the client, no equipment schedules, specifications, or installation guidelines were provided.

A comparison of the NPSHR and the NPSHA will be used to determine the minimum height of water in the tank, below which cavitation will occur and pump performance will be adversely affected.

The NPSHA is

$$\text{NPSHA} = \left(\frac{p_s}{\rho g} + \frac{V_s^2}{2g}\right)_{\text{pump,inlet}} - \frac{P_{\text{vapor}}}{\rho g}.$$

The energy equation will be used to find the pump inlet pressure, p_s:

$$\frac{P_1}{\rho g} + \alpha_1 \frac{V_1^2}{2g} + z_1 = \frac{p_s}{\rho g} + \alpha_s \frac{V_s^2}{2g} + z_s + H_{\text{lT,suction}}.$$

Point 1 is the free surface of the water in the tank. Thus, $V_1 \approx 0$ and $P_1 = P_{\text{atm}}$.
The pipe flow is turbulent. Hence, $\alpha_s \approx 1$.
Therefore,

$$\frac{p_s}{\rho g} = \frac{P_{\text{atm}}}{\rho g} - \frac{V_s^2}{2g} + (z_1 - z_s) - H_{\text{lT,suction}}.$$

Then,

$$\text{NPSHA} = \frac{P_{\text{atm}}}{\rho g} - \frac{V_s^2}{2g} + (z_1 - z_s) - H_{\text{lT,suction}} + \frac{V_s^2}{2g} - \frac{P_{\text{vapor}}}{\rho g}$$

$$\text{NPSHA} = \frac{P_{\text{atm}} - P_{\text{vapor}}}{\rho g} + (z_1 - z_s) - H_{\text{lT,suction}}.$$

Note that $H_{lT,suction}$ is the total head loss in the suction line of the pipe, only.

z_1 Is the height of the free surface of the water in the tank. The minimum height will be that which gives NPSHA = NPSHR.

So,

$$z_1 = \text{NPSHR} - \frac{P_{atm} - P_{vapor}}{\rho g} + z_s + H_{lT}.$$

From the performance plot of the $1\frac{1}{2} \times 1\frac{1}{2} \times 7$ in. casing pump, and at 30 gpm, the NPSHR is approximately 3 ft. At 70°F, $P_{vapor} = 0.363$ psia. $P_{atm} = 14.7$ psia. Based on the pump casing size, assume that the centerline of the pump is 1 ft above grade. Therefore, $z_s = 1$ ft.

The head loss in the suction line of the pipe is

$$H_{lT} = H_{l,suction} + \sum K_L \frac{V^2}{2g}.$$

The total length of suction piping is approximately 1050 ft.

Hence,

$$H_{lT} = \frac{3.0 \text{ ft wg}}{100 \text{ ft}} \times 1050 \text{ ft} + [0.5 + 1.5 + 0.16] \frac{(3.0 \text{ ft/s})^2}{2\left(32.2 \text{ ft/s}^2\right)}$$

$$H_{lT} = 32 \text{ ft}$$

and

$$z_1 = 3\text{ ft} - \frac{(14.7 - 0.363)\text{ lbf/in.}^2}{\left(62.3\text{ lb/ft}^3\right)\left(32.2\text{ ft/s}^2\right)} \times \frac{32.2\text{ lb/ft/s}^2}{1\text{ lbf}} \times \left(\frac{12\text{ in.}}{1\text{ ft}}\right)^2 + 1\text{ ft} + 32\text{ ft}$$

$$z_1 \approx 2.86\text{ ft} = 2\text{ ft}\,10\text{ in.}$$

According to this calculation, cavitation will only occur if the free surface of the fluid is less than **2 ft 10 in.** above grade. Therefore, the designer may choose any appropriate height above 2 ft 10 in. to mount the low-level switch. If this is not possible, and the low-level switch must be installed close to the base of the tank, the client may consider mounting the tank on a support that is at least 2 ft 10 in. high. This will generate the same static pressure head required to avoid cavitation. If mounting the tank is not an option, the client is advised to consider another tank with height greater than 50 in.

Closed-Loop Piping System

- In a **closed-loop piping system**, there are no points open to the ambient. Figure below shows schematic drawings of two-pipe and four-pipe closed-loop piping systems.
- Note the following regarding typical closed-loop piping systems:

a) Filters are optional for this system. They may be included in the design if it is deemed necessary to protect equipment.

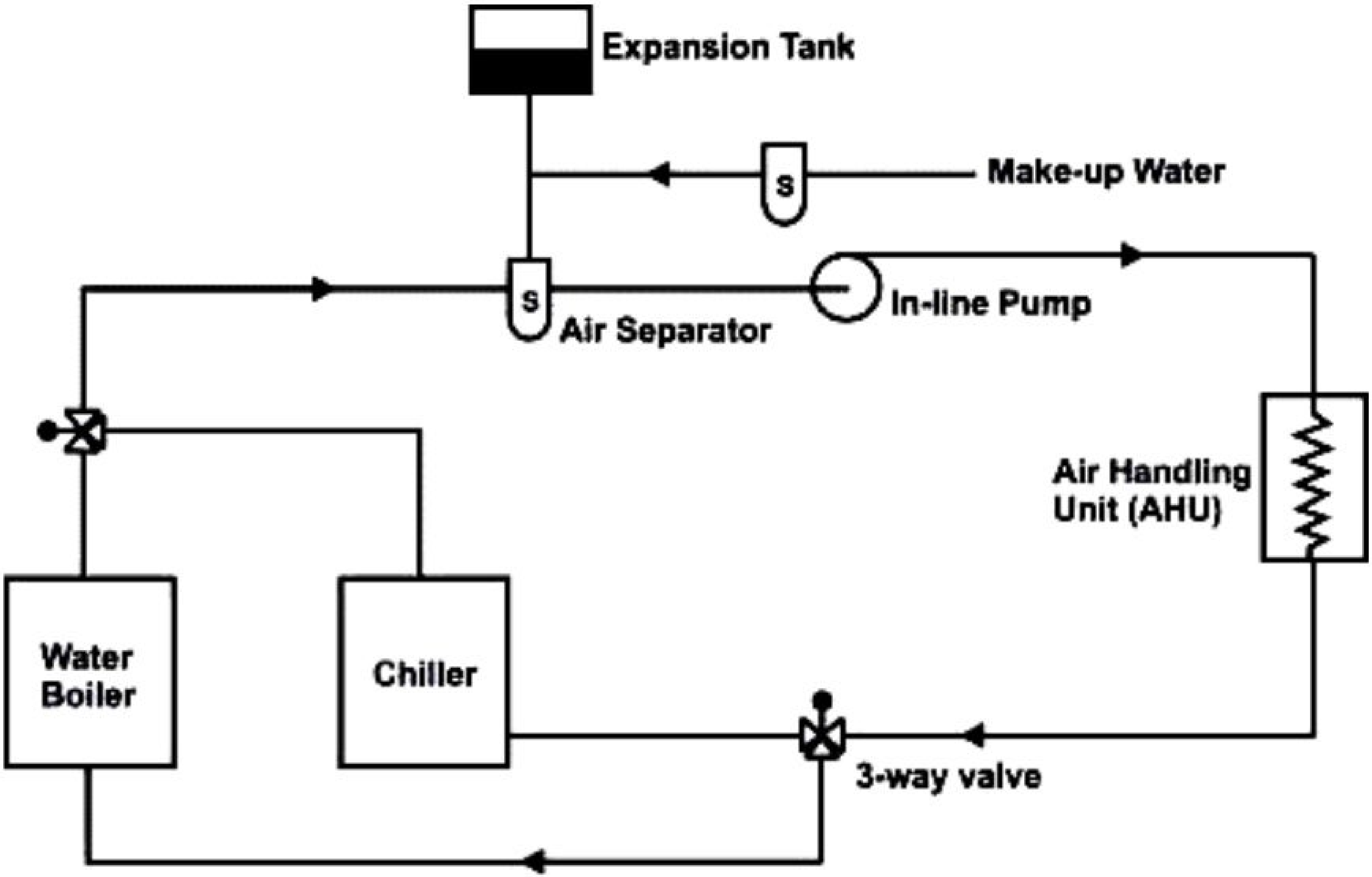
b) An expansion tank is required for closed-loop piping systems. There should be only one expansion tank in each loop of a piping system. Expansion tanks:

   i. Protect the system from damage due to volume changes induced by temperature variations.

   ii. Create a point of constant pressure in the system.

   iii. Provide a collecting space for entrapped air. Entrapped air is undesirable in the water systems since it makes the flow noisy and produces unwanted pipe vibrations—water hammer.

Closed-Loop Piping System



2-pipe system

Closed-Loop Piping System

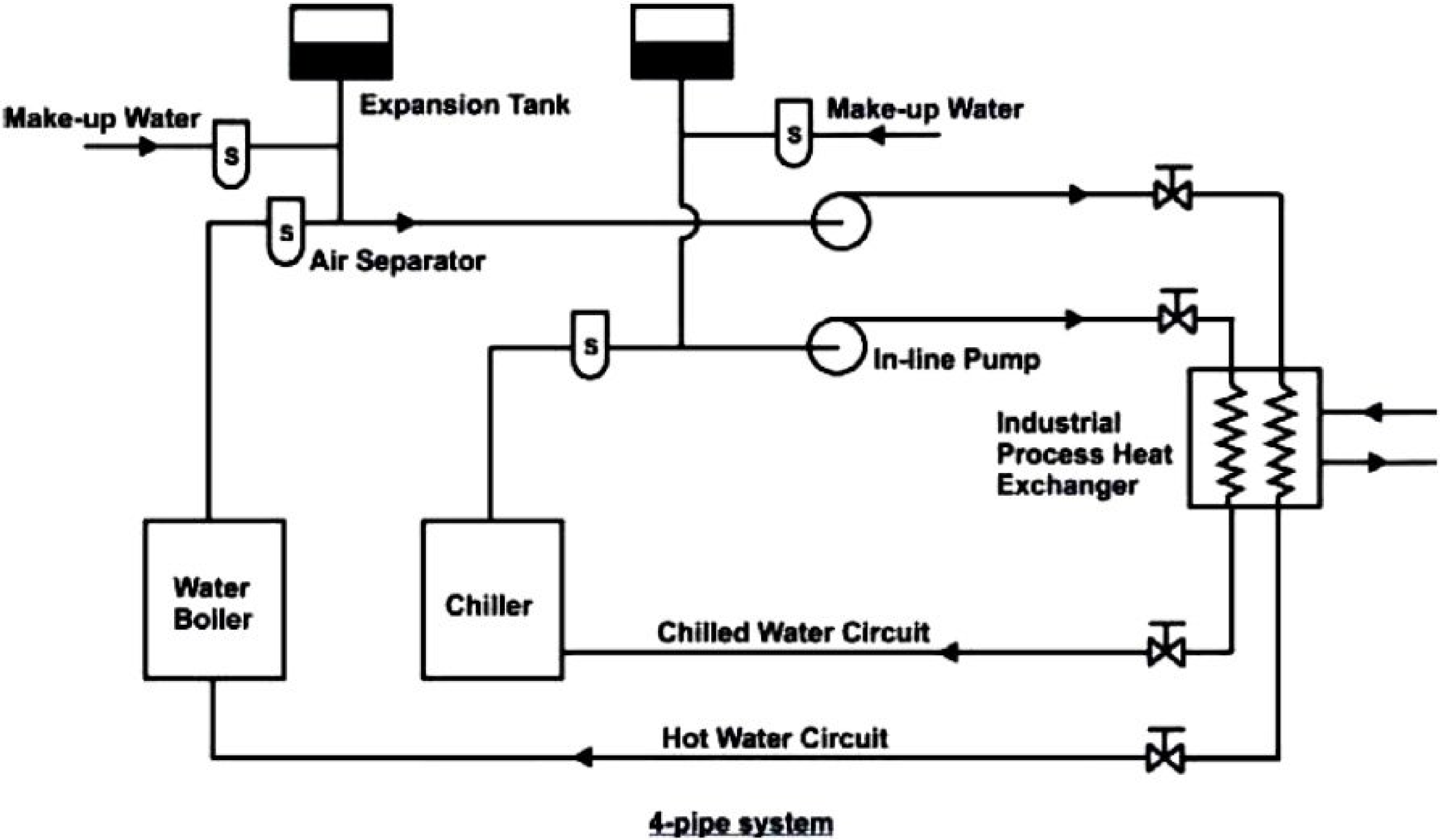


Figure. Diagrams of closed-loop piping systems

Closed-Loop Piping System

c) Air separators for air elimination from the closed-loop piping system are required. Air will enter the closed-loop piping system when **makeup water** from external sources is introduced. Devices such as a vortex air separator create a high-speed fluid vortex in its center. This low-pressure region enhances the release of air bubbles from the liquid, which is released through an automatic vent or other device.

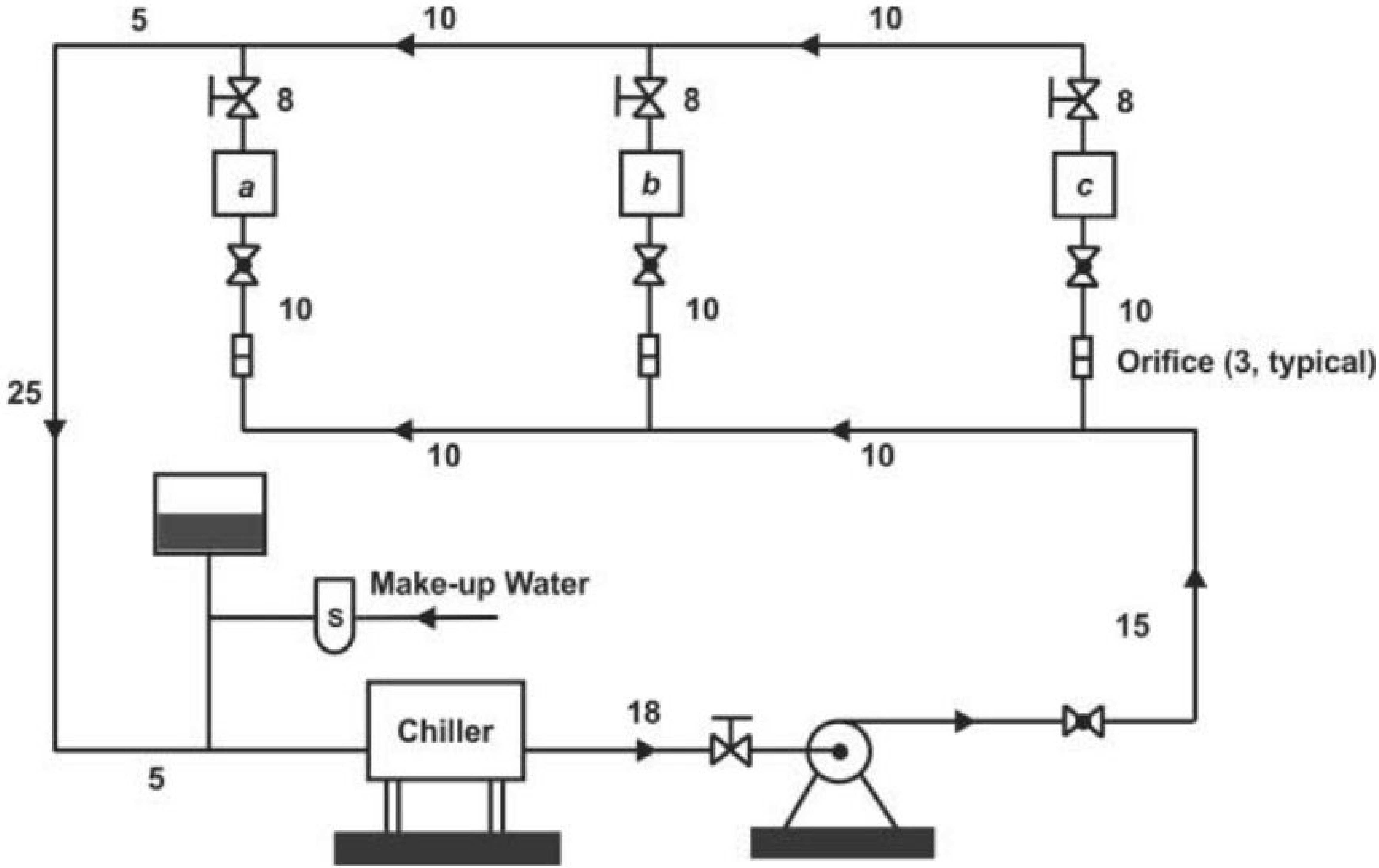
d) Pressure regulators, isolation, control, and/or three-way valves, flow meters, and thermocouples may also be required.

Practical Note 9. Flanged or Screwed Pipe Fittings?

Flanged pipe is generally specified for above ground service for air, water, sewage, oil, and other fluids where **rigid, restrained joints** are needed. It may be used for larger, heavier pipes and fluids. It is also widely used in industrial piping systems, water treatment plants, sewage treatment plants, and for other interior piping. American Water Works Association (AWWA) standards restrict the use of flanged joints underground due to the rigidity of the joints.

Example. Designing a Closed-Loop Piping System

- A consulting engineering firm received the following sketch of a water piping system from the Ministry of Defense in Ottawa.



Example. Designing a Closed-Loop Piping System

- For security reasons, units *a*, *b*, and *c* are unspecified. However, a chart is provided (see above) that gives details on the flow rates and head losses across the units. The head loss across each orifice is 6 ft. All pipe lengths are in foot. Complete the design of this system.
- ***Hint***: A piping and pump schedule must be provided.

Unit	Flow Rate (gpm)	Head Loss (ft)
a	30	15
b	40	12
c	50	10
Chiller	120	20

Possible Solution

Definition

Size the piping and specify the pumping requirements for a partially designed piping system provided by the Ministry of Defense.

Preliminary Specifications and Constraints

(i) The working fluid is cold water due to the presence of a chiller in the system.
(ii) Pipe velocity should be less than 10 fps for general building service. The choice of pipe material may limit the velocity further.
(iii) This is a two-pipe system with parallel piping, valves, orifices, undefined units, a chiller, and a pump.

Detailed Design

Objective

To size the pipes in the system and to size and select an appropriate pump. The piping material must be selected.

Data Given or Known

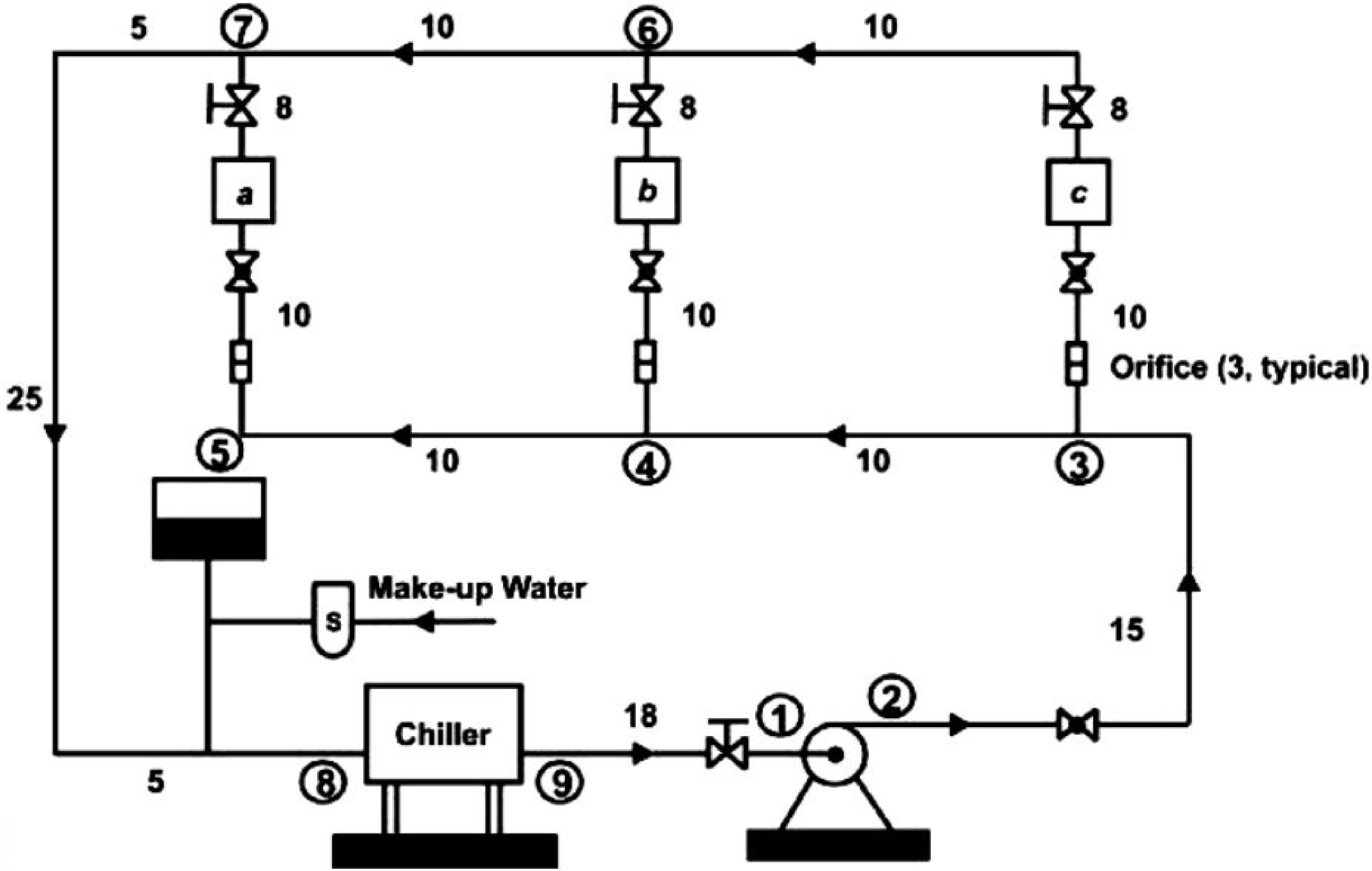
(i) Length of the pipe sections.
(ii) Total system flow rate is 120 gpm (from the chiller).
(iii) Head losses and flow rates for all the equipment are given (including the undefined units).
(iv) The orifice head loss is 6 ft.
(v) Preliminary layout of the piping system was provided by the Ministry of Defense.

Assumptions/Limitations/Constraints

(i) The Ministry did not provide information regarding the location of the system. Therefore, care will be taken to ensure that the system operates quietly.
(ii) Let the flow velocity be about 4 fps. This is acceptable for general building service or potable water. In addition, this velocity does not exceed the erosion limits of any general pipe material.
(iii) Limit pipe frictional losses to 3 ft of water per 100 ft of pipe.
(iv) Pipe changes should be gradual to reduce losses.
(v) All bends will be 90° flanged bends to facilitate maintenance and reduce losses. For smaller pipes, screwed/threaded bends will be used. This is more common in industry.
(vi) Branch and line flow tees are flanged. For smaller pipes, screwed/threaded tees will be used. This is more common in industry.
(vii) Negligible elevation head. Assume that all components are on the same level.
(viii) Assume that the piping material is **Schedule 40 steel**. This will provide durability and flexibility over type L copper pipes.

Sketch

The pipe sections are labeled and shown on the layout provided by the Ministry.



Analysis

Bear the following points in mind for this design problem:

(i) Pipe sizing for this system can be done quickly by using the appropriate friction loss charts for Schedule 40 pipe (Figure A.4). However, if the required sizes, flow rates, velocities, etc. are not included on the published charts or if the design is based on a specialized pipe material, then the designer should use Colebrook's equation (or other correlation equation) to find the friction factor (f) and iterate to find the pipe diameters.

(ii) Assign larger head losses per 100 ft of pipe to shorter pipe sections. The designer should attempt to have a constant velocity throughout the system (approximately).

Pipe Sections

The pipe sections in this design are:

Sections 2-3, 3-4, 4-5, 3-6, 4-6, 5-7, 6-7, 7-8, 9-1 (see sketch).

Consider Section 4-5. It has the shortest pipe length (10 ft) and the lowest flow rate (120–50–40 gpm = 30 gpm). Use a frictional loss of 3 ft of water per 100 ft of pipe as a guide to size the pipe.

With the friction loss and the flow rate, the friction loss chart for commercial steel pipe (Schedule 40) in a closed-loop piping system is consulted to find suitable pipe data (Figure A.4):

Nominal pipe size: 2 in.
Water velocity: 2.7 fps
Lost head: 1.6 ft per 100 ft of pipe
Major head loss: $\frac{1.6 \text{ ft}}{100 \text{ ft}} \times 10 \text{ ft} = 0.16 \text{ ft}$

The pipe system flow velocity should be close to 3 fps. This information will be used to size pipes in the other sections. See the Pipe Data table for the pipe sizes.

Now that the pipe diameters are known, the minor losses for each section can be estimated. See the Minor Losses table for information.

Minor Losses

The loss coefficients for the bends, fittings, and area changes for the piping in Section 4.5 (2 in. diameter) are given below from Table A.14:

Gate valve: $K_{\text{gate}} = 0.16$
Ball valve: $K_{\text{ball}} = 0.05$
Flanged 90° regular bends: $K_{90^\circ\ \text{bend}} = 0.39$
Branch tee: $K_{\text{branch tee}} = 0.80$
Line tee: $K_{\text{line tee}} = 0.19$
Pipe contraction: $K_{\text{contraction}} = 0.07$ for a contraction angle of 60°
Pipe expansion: $K_{\text{expansion}} = 0.30$ for $d/D = 0.2$

$$K_{\text{expansion}} = 0.25 \text{ for } d/D = 0.4$$
$$K_{\text{expansion}} = 0.15 \text{ for } d/D = 0.6$$
$$K_{\text{expansion}} = 0.10 \text{ for } d/D = 0.8$$

Minor Loss:

1 flanged 90° bend: $K_{90^\circ \text{ bend}} = 0.39$
1 line tee: $K_{\text{line tee}} = 0.19$
1 pipe contraction from a 3 in. diameter in Section 3–4 to a 2 in. diameter in Section 4-5:
$K_{\text{contraction}} = 0.07$

Therefore,

$$H_{\text{lm}} = \sum K_L \frac{V_{\text{ave}}^2}{2g} = (0.39 + 0.19 + 0.07)\left[\frac{(2.8\ \text{ft/s})^2}{2\left(32.2\ \text{ft/s}^2\right)}\right] = 0.079\ \text{ft}.$$

The total head loss in the Section 4-5 is

$$H_{\text{lT}} = H_{\text{l}} + H_{\text{lm}} = 0.16\ \text{ft} + 0.079\ \text{ft} = 0.24\ \text{ft}.$$

A similar procedure is followed for the other pipe sections. The results are shown in the tables below.

Minor Losses								
Pipe Section No.	Gate Valves	Ball Valves	90° Regular Bends	Tees-Branch	Tees-Line	Gradual Expansion (K_L = vary)	Gradual Contraction (K_L = 0.07)	Total Minor Loss (ft)
2-3		0.05 (one)	0.3 (three)	0.64 (one)				0.32
3-4					0.19 (one)		0.07 (one)	0.04
4-5			0.39 (one)		0.19 (one)		0.07 (one)	0.08
3-6	0.35 (one)	0.05 (one)	0.39 (one)	0.80 (one)		$K_L = 0.10$		0.27
4-6	0.35 (one)	0.05 (one)		0.80 (two)				0.23
5-7	0.35 (one)	0.05 (one)		0.80 (one)				0.15
6-7					0.19 (one)	$K_L = 0.10$		0.07
7-8			0.30 (two)		0.14 (two)			0.18
9-1	0.30 (one)		0.30 (one)					0.12

Pipe Data							
Pipe Section No.	Pipe Length (ft)	Flow Rate (gpm)	Lost Head (ft/100 ft)	Fluid Velocity (ft/s)	Nominal Size (in.)	Minor Losses (ft)	Total Head Loss (ft)
2-3	15	120	1.4	3.6	$3\frac{1}{2}$	0.32	0.53
3-4	10	70	1.3	3	3	0.04	0.17
4-5	10	30	1.6	2.8	2	0.08	0.24
3-6	28	50	1.8	3.2	$2\frac{1}{2}$	0.27	0.77
4-6	18	40	1.3	2.7	$2\frac{1}{2}$	0.23	0.46
5-7	18	30	1.6	2.8	2	0.15	0.44
6-7	10	90	1.9	3.8	3	0.07	0.26
7-8	35	120	1.4	3.6	$3\frac{1}{2}$	0.18	0.67
9-1	18	120	1.4	3.6	$3\frac{1}{2}$	0.12	0.37

Next, consider the lost head for each of the three parallel circuits between points 3 and 7. Let: circuit *a* includes sections 3-4, 4-5, 5-7, the orifice, and unit *a*.

Circuit *b* includes sections 3-4, 4-6, 6-7, the orifice, and unit *b*.
Circuit *c* includes sections 3-6, 6-7, the orifice, and unit *c*.

The circuit head losses are

$$H_a = H_{34} + H_{45} + H_{57} + H_{orifice} + H_{unit\text{-}a} = 0.17\text{ ft} + 0.24\text{ ft} + 0.44\text{ ft} + 6\text{ ft} + 15\text{ ft} = 21.9\text{ ft}$$
$$H_b = H_{34} + H_{46} + H_{67} + H_{orifice} + H_{unit\text{-}b} = 0.17\text{ ft} + 0.46\text{ ft} + 0.26\text{ ft} + 6\text{ ft} + 12\text{ ft} = 18.9\text{ ft}$$
$$H_c = H_{36} + H_{67} + H_{orifice} + H_{unit\text{-}c} = 0.77\text{ ft} + 0.26\text{ ft} + 6\text{ ft} + 10\text{ ft} = 17.0\text{ ft}.$$

The system will be balanced by installing balancing valves in circuits *b* and *c* to increase the head to 21.9 ft (for circuit *a*). The head loss for circuit *a* will be used to find the pump head (h_{pump}).

For this closed-loop system, let the starting and ending point be point 1. Hence, the pump head is

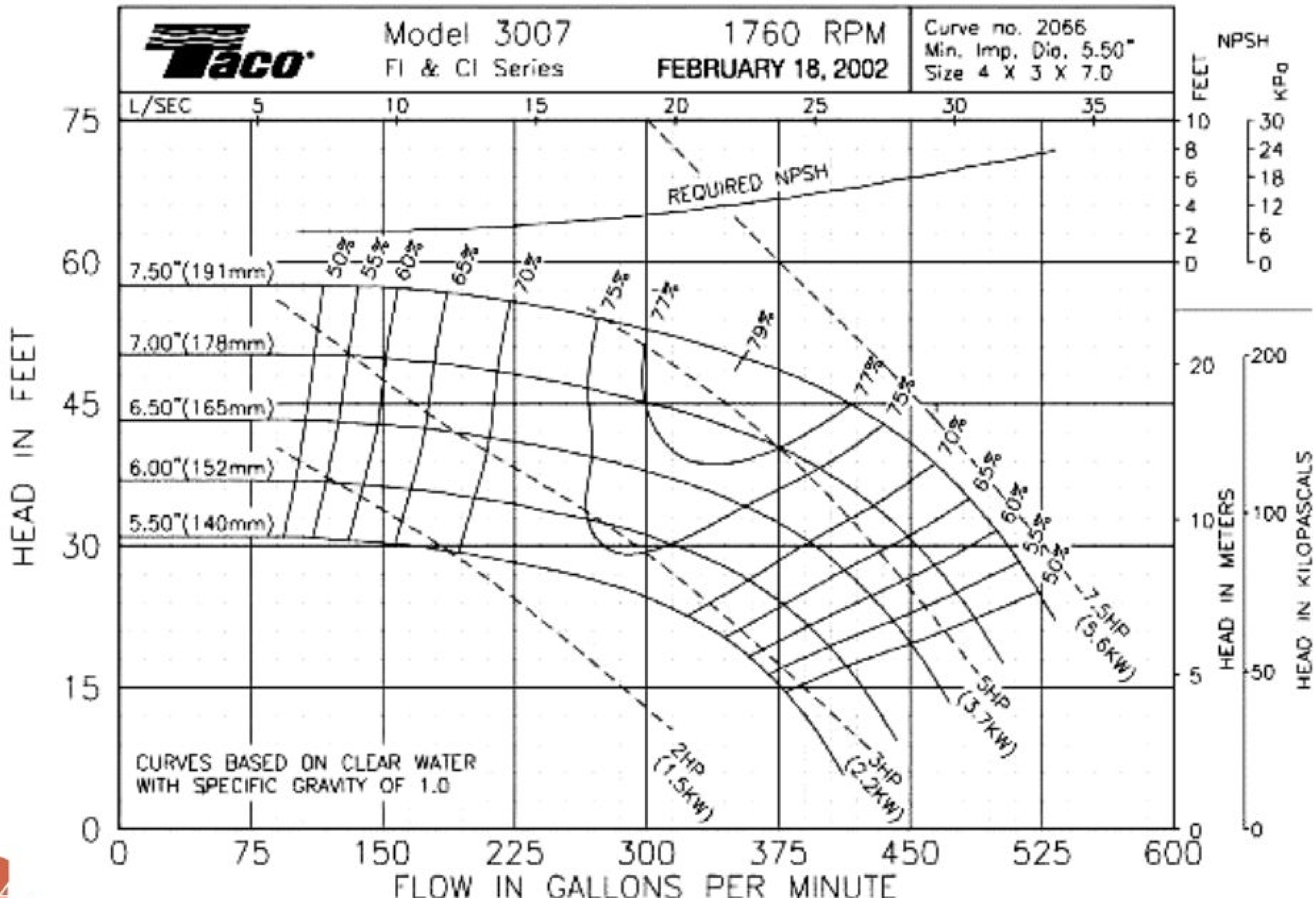
$$h_{\text{pump}} = \left(\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1\right) - \left(\frac{p_1}{\rho g} + \frac{V_1^2}{2g} + z_1\right) + H_{\text{lT}} = H_{\text{lT}}$$

$$h_{\text{pump}} = H_{23} + H_{34} + H_{45} + H_{57} + H_{78} + H_{91} + H_{\text{orifice}} + H_{\text{unit-a}} + H_{\text{chiller}}$$
$$h_{\text{pump}} = 0.53\ \text{ft} + 0.17\ \text{ft} + 0.24\ \text{ft} + 0.44\ \text{ft} + 0.67\ \text{ft} + 0.37\ \text{ft} + 6\ \text{ft} + 15\ \text{ft} + 20\ \text{ft}$$
$$h_{\text{pump}} = 43.4\ \text{ft} \sim 45\ \text{ft of water.}$$

For this system, a pump that is rated to produce 120 gpm at 45 ft of head is required. Use manufacturer's charts to select an appropriate pump. The drawing from the Ministry shows a base-mounted pump. In addition, the flow rate and pump head are large. A base-mounted pump will provide additional support. Choose a Taco FI/CI Series pump. A 4 in. × 3 in. × 7 in. casing pump is selected. From the performance plot for this family of pumps, the final choice is

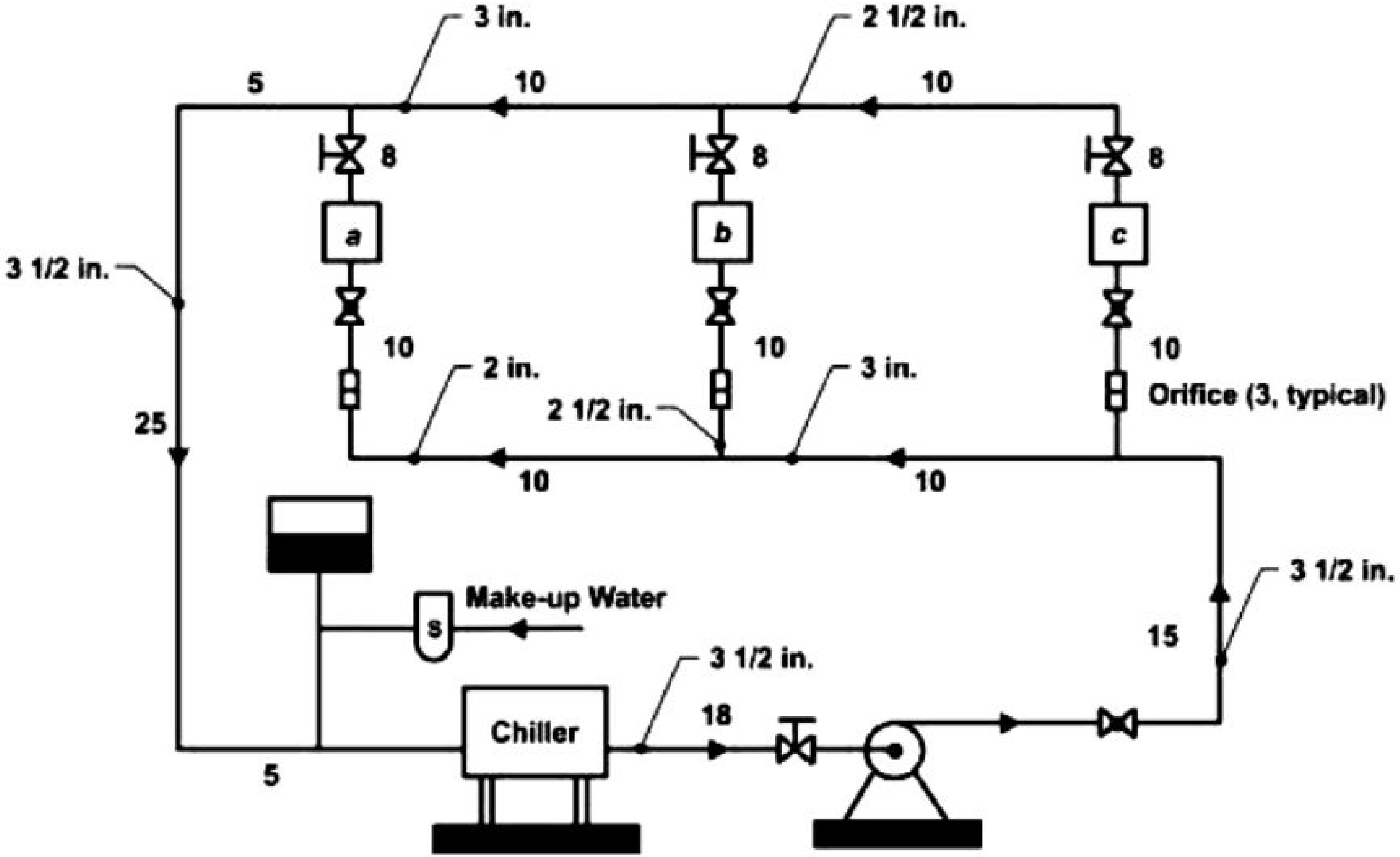
4 in × 3 in × 7 in. casing
7 in. impeller diameter
3-hp motor
1760 rpm speed.

- The pump performance curve is shown below



Drawings

The drawing below shows the pipe sizes required.



Conclusions

The piping system has been designed. All the pipe sizes are known, and the piping material has been selected. A pump has been specified and selected. It should be noted that for the pipe sizes in this design, threaded fittings could have been used. In that case, the losses would have increased. It should also be noted that $4 \times 3^1/_2$ in. and $3^1/_2 \times 3$ in. reducing fittings will be required on the pump suction and discharge, respectively, for installation of the pump.

The Pipe Data and Pump Schedule are shown below.

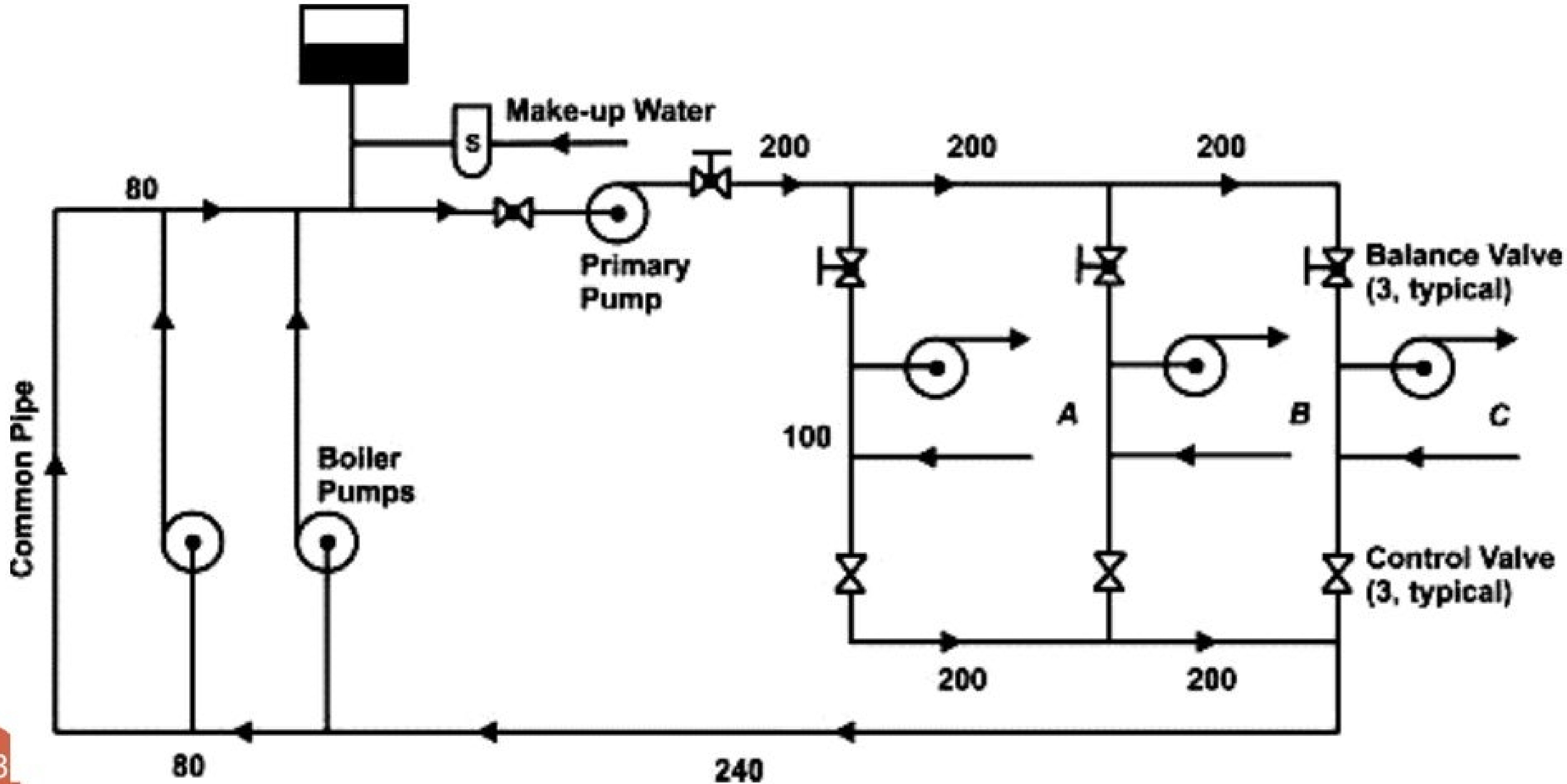
Pipe Data

Pipe Section No.	Pipe Length (ft)	Flow Rate (gpm)	Lost Head (ft/100 ft)	Fluid Velocity (ft/s)	Nominal Size (in.)	Minor Losses (ft)	Total Head Loss (ft)
2-3	15	120	1.4	3.6	$3^1/_2$	0.32	0.53
3-4	10	70	1.3	3	3	0.04	0.17
4-5	10	30	1.6	2.8	2	0.08	0.24
3-6	28	50	1.8	3.2	$2^1/_2$	0.27	0.77
4-6	18	40	1.3	2.7	$2^1/_2$	0.23	0.46
5-7	18	30	1.6	2.8	2	0.15	0.44
6-7	10	90	1.9	3.8	3	0.07	0.26
7-8	35	120	1.4	3.6	$3^1/_2$	0.18	0.67
9-1	18	120	1.4	3.6	$3^1/_2$	0.12	0.37

Pump Schedule									
				Fluid			Electrical		
Tag	Manufacturer and Model Number	Type	Construction	Flow Rate (gpm)	Working Fluid	Head Loss (ft)	Motor Size (hp)	Motor Speed (rpm)	V/Ph/Hz
P-1	Taco, FI/CI Series, Model 3007, or equal	Centrifugal, base-mounted	Cast iron $4 \times 3 \times 7$ in. Casing, 7 in.ϕ	120	Water	45	3	1760	208/3/60

Problems

- MDM Consulting Engineers, Inc. has prepared the following piping schematic to supply water to a chemical plant. The lengths shown are in foot. Piping is Schedule 40, commercial steel. Complete the design of the primary circuit piping system.



Problem

Circuit	Flow Rate (gpm)	Control Valve Head Loss (ft)
A	60	40
B	70	50
C	70	50

End of Lecture 3

Next Lecture

Lecture 4: Fundamentals of Heat Exchanger Design