TURBOMACHINES (FLUID MACHINERY)

Objectives of the chapter
- Give a brief introduction about tubomachines
- Explain pump testing and selection
- Explain fan testing and selection

A fluid machine is a device that either performs work on, or extracts work (or power) from a fluid.

Examples: **<u>Pump</u>**, turbine, compressor, **<u>fan</u>**

Fluid machines may be classified as

1) Positive displacement

2) Dynamic flow machines

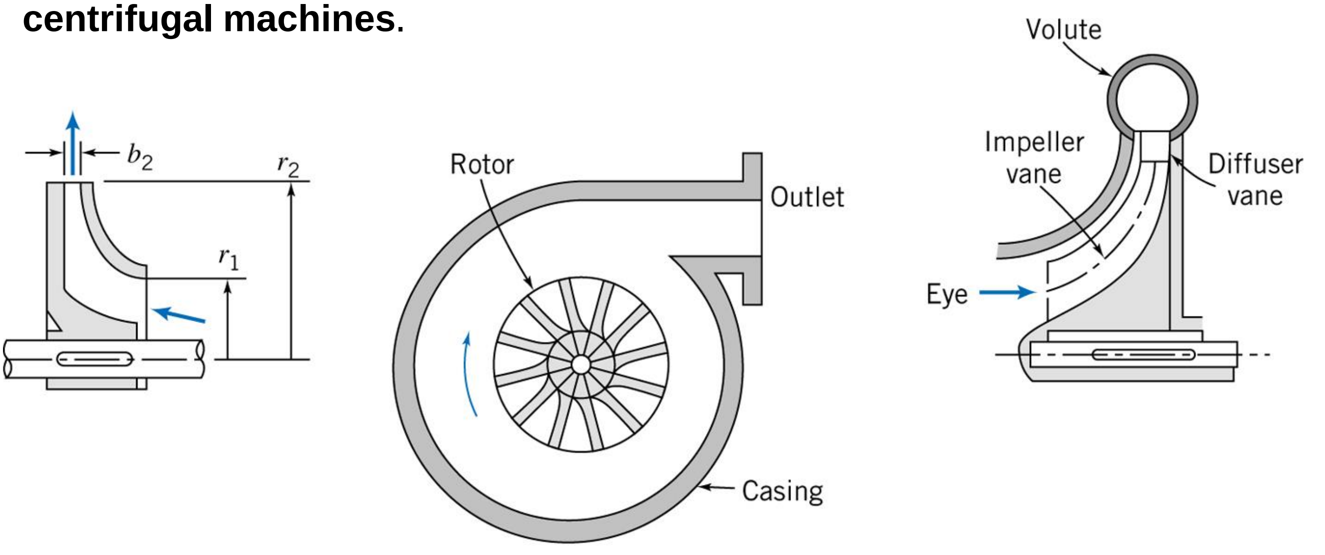
In a **positive displacement machine**, energy transfer is accomplished by volume changes that occur due to the movement of a boundary.

- reciprocating pump
- reciprocating compressor
- reciprocating steam engine

A **dynamic fluid machine** which is called **turbomachine** uses a moving (rotating) rotor, carrying a set of blades or vanes, to transfer work to or from a moving stream of fluid. If the **work done on the fluid** by the rotor, the machine is called a pump or compressor. If **the fluid delivers work to the rotor**, the machine is called turbine.

Depending on the **motion of the fluid with respect to axis of the rotor**, the turbomachines are classified as

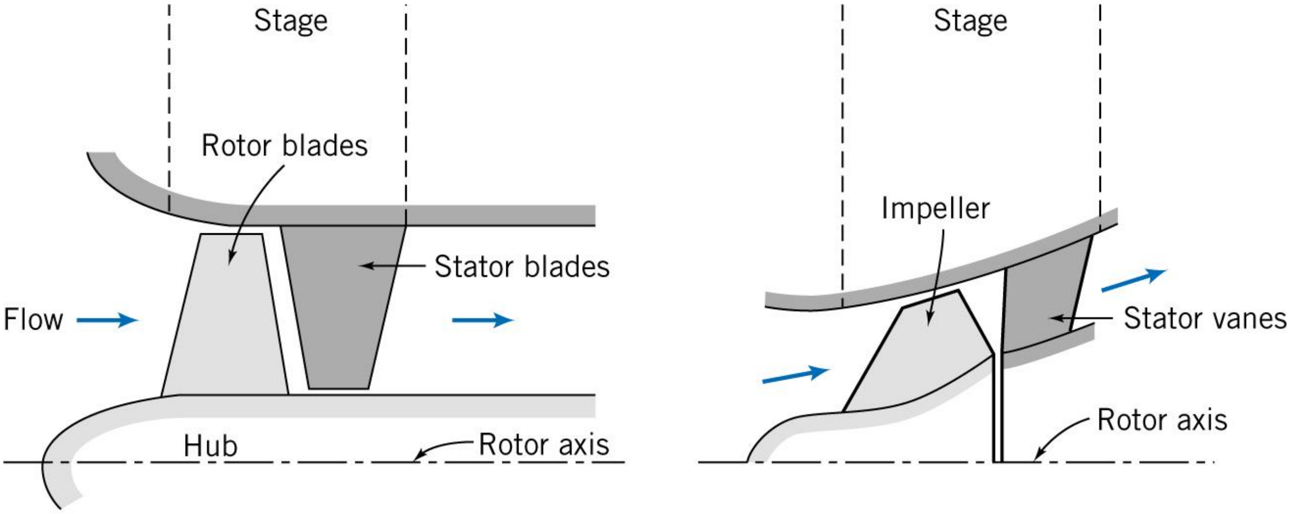
1) **Radial flow machines**: In radial flow machine, fluid moves primarily radialy from rotor inlet to rotor outlet, although the fluid may be moving in the axial direction at the machine inlet and outlet. Such machines are also called **centrifugal machines**.



(*a*) Centrifugal pump (*b*) Centrifugal blower (*c*) Centrifugal compressor

2) **Axial flow machines**: In an axial flow machine, the fluid follows a path which is nearly parallel to the axis of the rotor.

3) **Mixed flow machines**: In mixed flow machines, the fluid has both axial and radial velocity components as passes through the rotor.



(*a*) Axial-flow compressor stage (*b*) Mixed-flow pump stage

PUMPS AND PIPING SYSTEMS

Pumps are devices used to move liquid through a pipeline.

Objective of the chapter
- Examine the types of the pumps
- Provide guidelines useful in selecting the type of the pump for a particular application.
- Discuss pump testing methods
- Discuss cavitation and how it is avoided
- Design practices

Types of Pumps: There are two types of pumps.

1) Dynamic pumps
2) Positive displacement pumps

Dynamic pumps usually have a rotating component that imparts energy to the fluid in the form of high velocity, high pressure, or high temperature.

Positive displacement pumps have fixed volume chambers that take in and discharge the fluid.

Dynamic Pumps

1) **Axial pump (propeller pump or turbine pump)**
This type of pumps are used for short vertical pumping distance (low-lift) applications.

2) Radial flow pump (Centrifugal pump)

3) Mixed flow pump

Multistage pumps
In some pump designs, the discharge of one impeller immediately enters another. The discharge from the first or lowest impeller casing enters the second and so forth. The impeller casings are bolted together and can consist of any number of desired stages.

Positive Displacement Pump

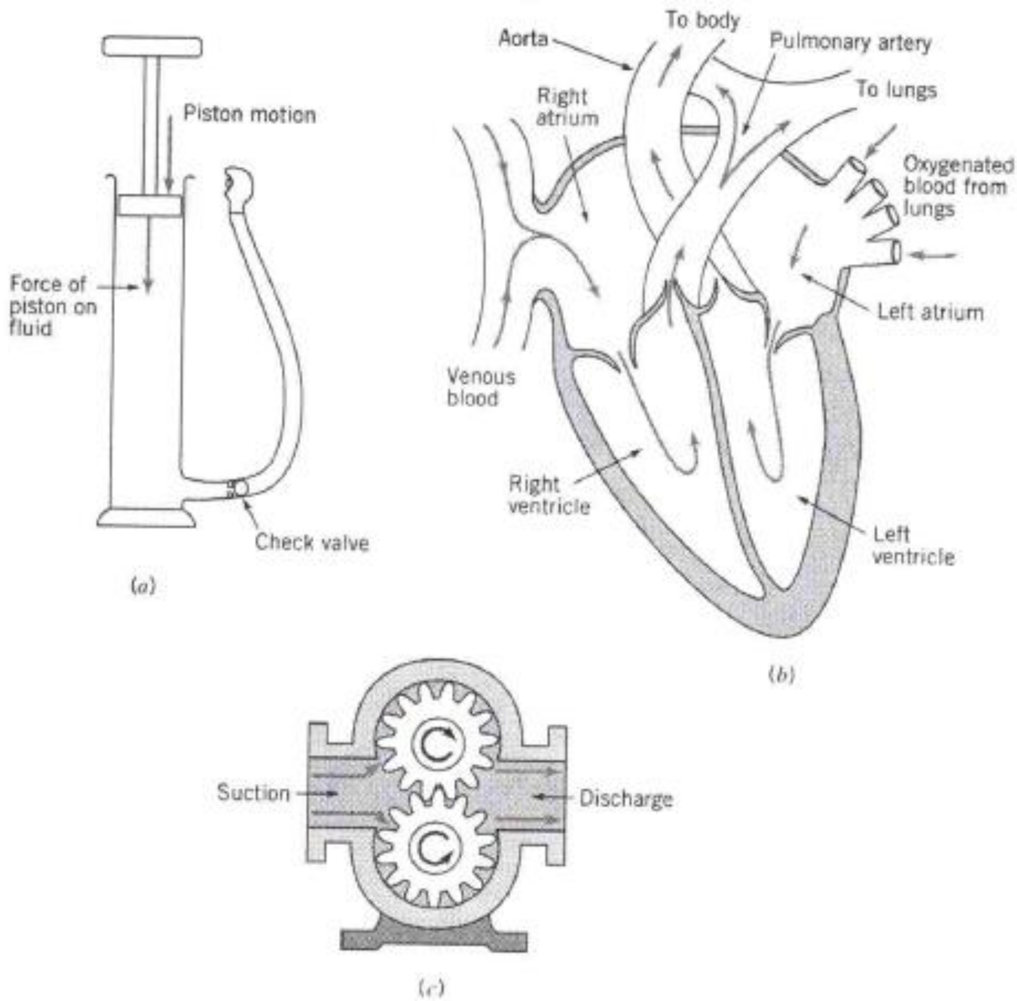
There are different designs of positive displacement pumps.

1) **Reciprocating pump**

A reciprocating piston draws in fluid on an intake stroke, and moves that fluid out on the discharge stroke. One-way valves in he flow lines control the flow direction.

2) **Rotary Gear Pump**

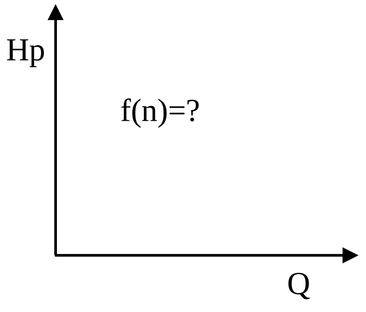
It consists of two matched gears that rotate within a housing. Fluid enters the region between the two gears, and as the gears rotate, fluid is drawn into the volumes between adjacent teeth and housing. The fluid is discharged on the other side of the housing.

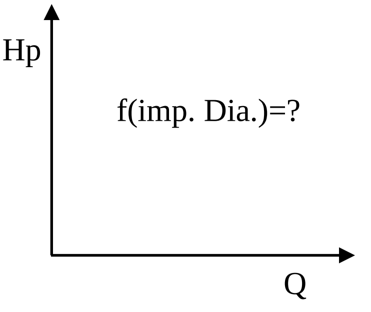


PUMP TESTING METHODS

Details regarding the design of pumps are responsibility of pump manufacturers. **Our purpose here is to examine how pumps are tested and sized (selected) for a given application.** To specify fluid machines for a flow system, the designer should know the pressure rise (or head), torque, power requirement and efficiency of the machine. For a given machine, each of these characteristics is a function of flow rate. The characteristics of similar machines depend on impeller diameter and operating speed.

Objective of pump tests is to obtain a performance map for pumps, i.e. pump head and efficiency as a function of volume flow rate, rotational speed and impeller diameter.





To calculate the above characteristics, pumps are tested by pump manufacturers and during tests the following data are obtained .

- Torque
- Rotational speed
- Inlet pressure
- Outlet pressure
- Volume flow rate

PUMP TEST SETUP

There are different methods to measure the above parameters. **However, all the tests must be carried out according to the related standards.** A centrifugal pump testing setup is given in the figure.

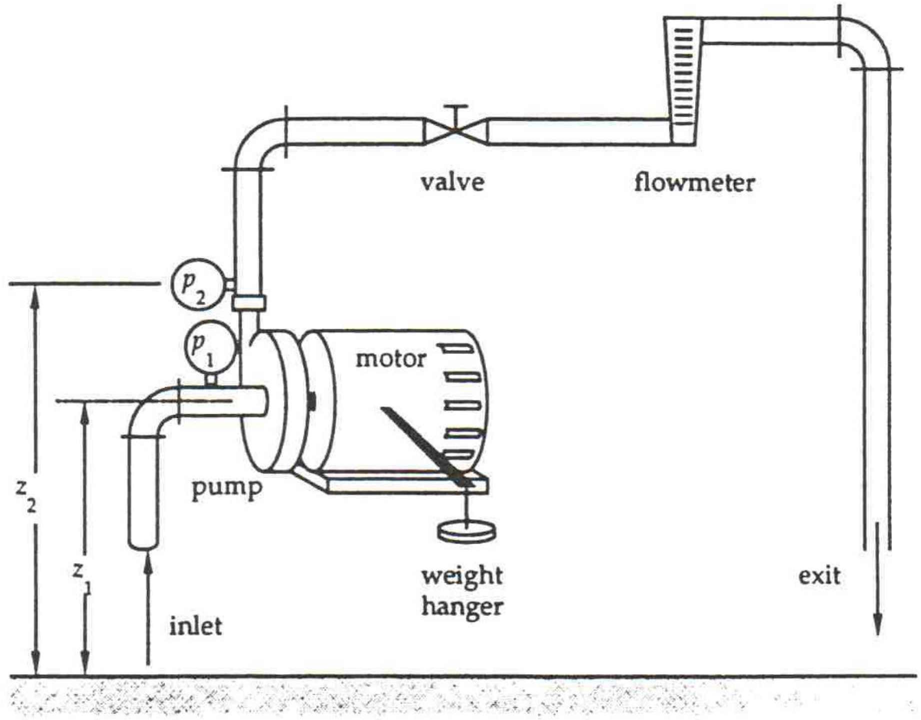


FIGURE 5.1. *Centrifugal pump testing setup.*

In this system, the impeller is rotated by the motor and the motor is mounted so that it is free to rotate within limits. The motor housing tends to rotate in the opposite direction from that of the impeller. Weights are placed on the weight hanger so that at any rotational speed, the motor is kept at an equilibrium position.

To calculate the torque exerted by the motor, the weight is multiplied by the distance from the motor axis to the weight hanger, i. e.

$T = W \times L$ [Nm] or [$lb_f ft$]

- Rotation of the motor (impeller) is obtained with any number of devices.
- Pressures at pump inlet and outlet are measured by pressure gages.
- A flow meter is used to measure the flow rate.
- A valve placed in the outlet line is used to control the flow rate.

Using the system described above, **for different valve positions (flow rates)**, the following parameters are measured:

Raw Data				
Parameter	**Symbol**	**Dimensions**	**Units**	
torque	T	F·L	J=N·m	lbf·ft
rotational speed	ω	1/T	rad/s	rad/s
inlet pressure	p_1	F/L^2	kPa	psi (lbf/ft^2)
outlet pressure	p_2	F/L^2	kPa	psi (lbf/ft^2)
volume flow rate	Q	L^3/T	m^3/s	ft^3/s (gpm)

PUMP CHARACTERIZATION PARAMETERS

TABLE 5.2. *Pump characterization parameters.*

Reduced Data				
Parameter	**Symbol**	**Dimensions**	**Units**	
input power	dW_a/dt	F·L/T	W = J/s	ft·lbf/s (hp)
total head diff	ΔH	L	m	ft
power to liquid	dW/dt	F·L/T	W	ft·lbf/s (hp)
efficiency	η	—	—	—

CALCULATION OF PUMP CHARACTERIZATION PARAMETERS

INPUT POWER

$$-\frac{dW_a}{dt} = T\omega \quad [W]$$

TOTAL HEAD DIFFERENCE

Total head difference is calculated as the difference between the total head at pump outlet (section 2) and total head at pump inlet (section 1)

$$\Delta H = H_2 - H_1 = \left(\frac{P_2 g_c}{\rho g} + \frac{V_2^2}{2g} + z_2 \right) - \left(\frac{P_1 g_c}{\rho g} + \frac{V_1^2}{2g} + z_1 \right) \qquad [m \;\; or \;\; ft]$$

POWER TO LIQUID

The power imparted o the liquid is calculated with the steady flow energy equation applied from section 1 to 2:

$$-\frac{dW}{dt} = \frac{\dot{m}g}{g_c}\left[\left(\frac{P_2 g_c}{\rho g} + \frac{V_2^2}{2g} + z_2 \right) - \left(\frac{P_1 g_c}{\rho g} + \frac{V_1^2}{2g} + z_1 \right) \right] \quad [W]$$

In terms of total head, we expres

$$-\frac{dW}{dt} = \frac{\dot{m}g}{g_c}\left(H_2 - H_1 \right) = \frac{\dot{m}g}{g_c}\Delta H \quad [W]$$

EFFICIENCY

$$\eta = \frac{Power \; imparted \; to \; liquid}{input \;\; power \; to \; impeller} = \frac{dW/dt}{dW_a/dt}$$

PERFORMANCE MAP

The experimental technique used in obtaining data depends on the desired method of expressing the performance characteristics.

The objective of the entire test is to locate the region of maximum efficiency for the pump which is being tested.

For example, data could be taken on only one impeller casing-motor combination (keeping the impeller diameter constant) at different rotational speeds with different flow rates. Then, total head difference versus flow rate and the iso-efficiency curves can be plotted as below.

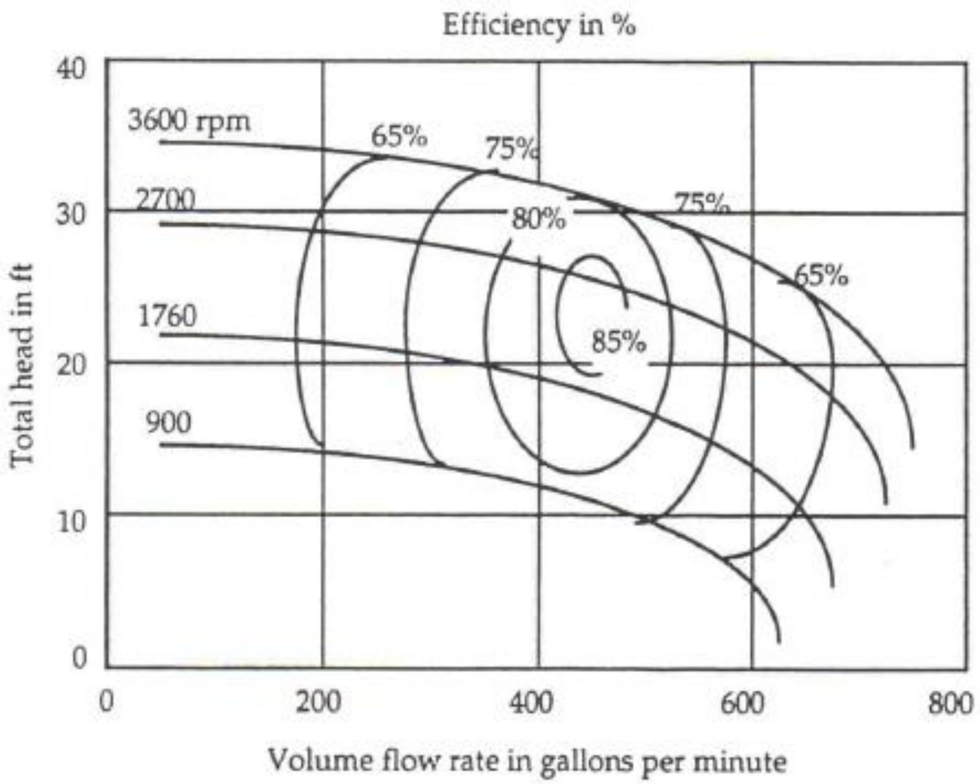


FIGURE 5.2. *Performance map of one impeller-casing-motor combination obtained at four different rotational speeds.*

Data could also be taken using one pump casing-motor combination with different impeller diameters keeping the rotational speed the same. Then, total head difference versus flow rate and iso-efficiency curves can be plotted as below.

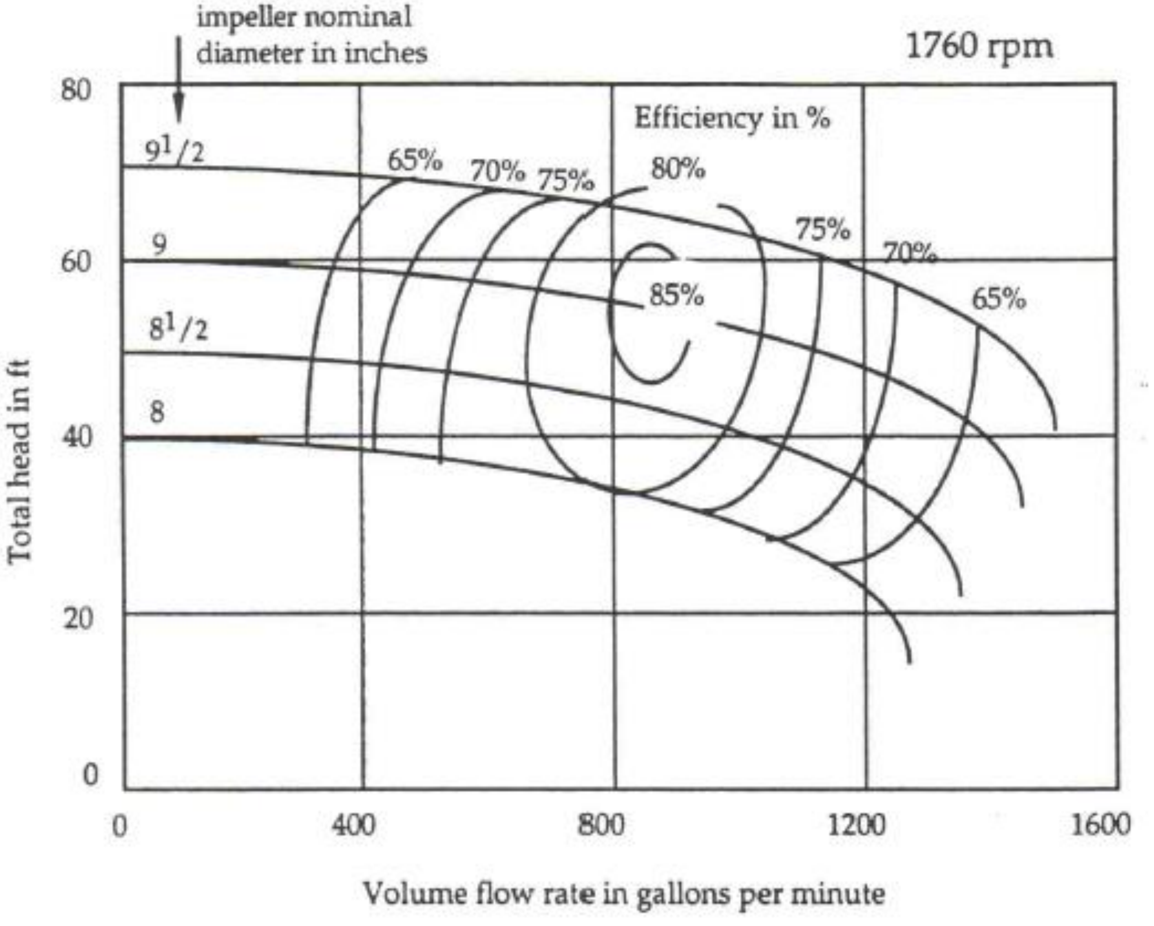


FIGURE 5.3. *Performance map of one motor-casing combination and four different impellers obtained at one rotational speed.*

To potential users, a manufacturer would need to supply a summary of the maximum efficiency region of all the pumps manufactured.

A graph of Δ**H versus Q** for a number of pumps showing only the maximum efficiency region for each pump is given below.

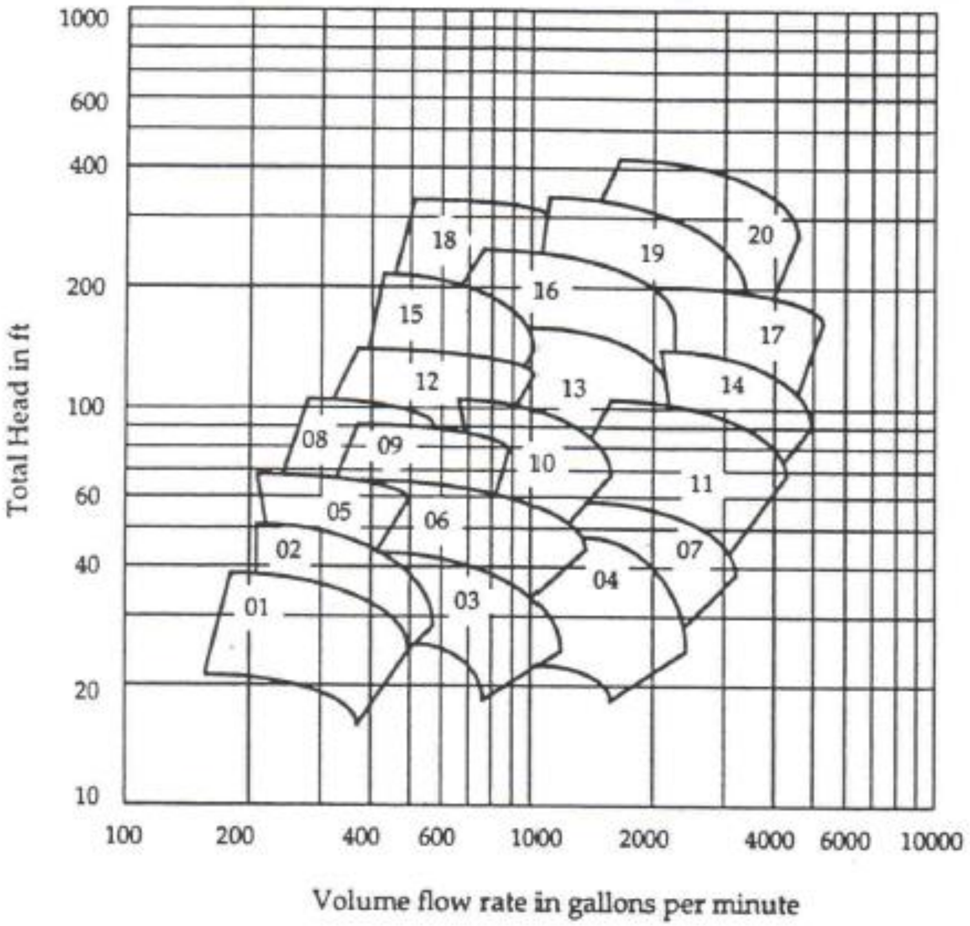


FIGURE 5.4. *Composite graph of total head difference versus volume flow rate showing maximum efficiency regions for 20 pumps.*

Example: A pump is tested using the pump test setup given above. For one setting of the valve in the discharge line, the following data were obtained:

Torque= T= 0.5 ft lbf
Rotational speed = ω = 1800 rpm
Inlet pressure = p1 = 3 psig
Outlet pressure = p2 = 20 psig
Volume flow rate = Q = 6 gpm
Height to the inlet = z1 = 2 ft
Height to the outlet = z2 = 3 ft
Inlet flow line = 2-nominal schedule 40
Outlet line = $1^{1/2}$-nominal schedule 40
Fluid = water

Calculate the pump characterization parameters.

Solution:

For 2-nominal sch 40 pipe: D1=0.1723 ft A= 0.02330 ft2 (Table D1)
For $1^{1/2}$-nomi sch 40 pipe: D2=0.1342 ft A= 0.01414 ft2 (Table D1)

For water: ρ = 62.4 lbm/ft^3 = 1.94 slug/ft^3

Example: Figure shows a pipeline that conveys water to an elevated tank at a campsite. The elevated tank supplies water to people taking showers. The 40 ft long pipeline contains 3 elbows and one ball check valve, and is made of 6-nominal schedule 40 PVC pipe. The pump must deliver 250 gpm. Select a pump for the system, and calculate the pumping power.

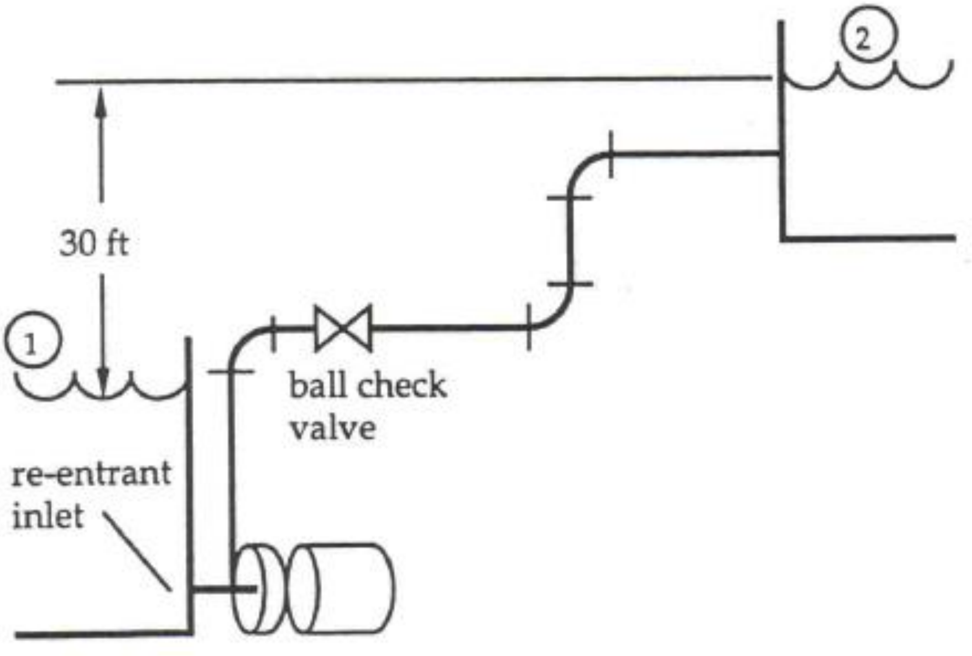


FIGURE 5.5. *The piping system of Example 5.2.*

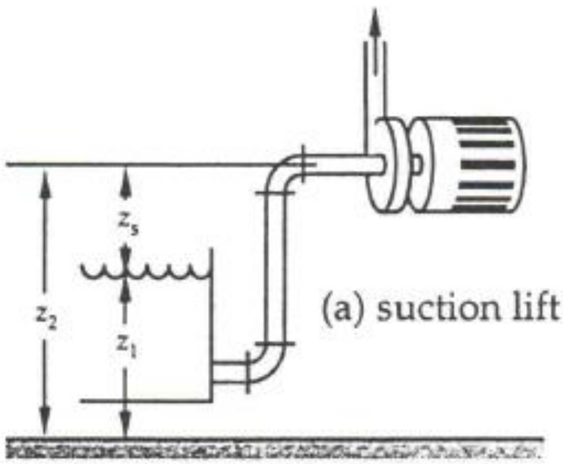
Solution:

For 6-nominal sch 40 pipe: D=0.5054 ft   A= 0.2006 ft^2 (Table D1)
For PVC ε/D= 0.0

For water: ρ = 62.4 lbm/ft^3 = 1.94 slug/ft^3, μ = 1.9 x 10-5 lbf s/ft^2

CAVITATION AND NET POSITIVE SUCTION HEAD

The suction line of a pump contains liquid at a pressure that is lower than the atmospheric p



If this suction **pressure is sufficiently low**, the liquid will **begin to boil** at the local temperature. **For example, water boils at 33 ^{o}C if the pressure is lowered to 5.1 kPa.**

Boiling involves vapor bubble formation, and this phenomenon when it occurs in a pump is **called cavitation**. In a cavitating pump, vapor bubbles usually form at the inlet of the impeller; and as they move through the impeller with the liquid, the bubbles encounter a high pressure region. Due to this high pressure, bubbles collapse and pressure waves form.

The pressure waves have an erosive effect on the impeller and housing, known **as cavitation erosion**. If the situation is not corrected, the pump may eventually fail due to **metal erosion and fatigue of shaft bearing and/or seals**.

When cavitation occurs, the impeller moves in the vapor bubble-liquid mixture. As a result, the **efficiency of the pump falls drastically.**

The cavitation is a result of incorrect installation of the system. The inception of the cavitation is predictable. During the design of the system, **the engineer should ensure that the cavitation will not occur.**

Pump manufacturers perform tests on pumps and provide information useful for predicting when cavitation will occur.

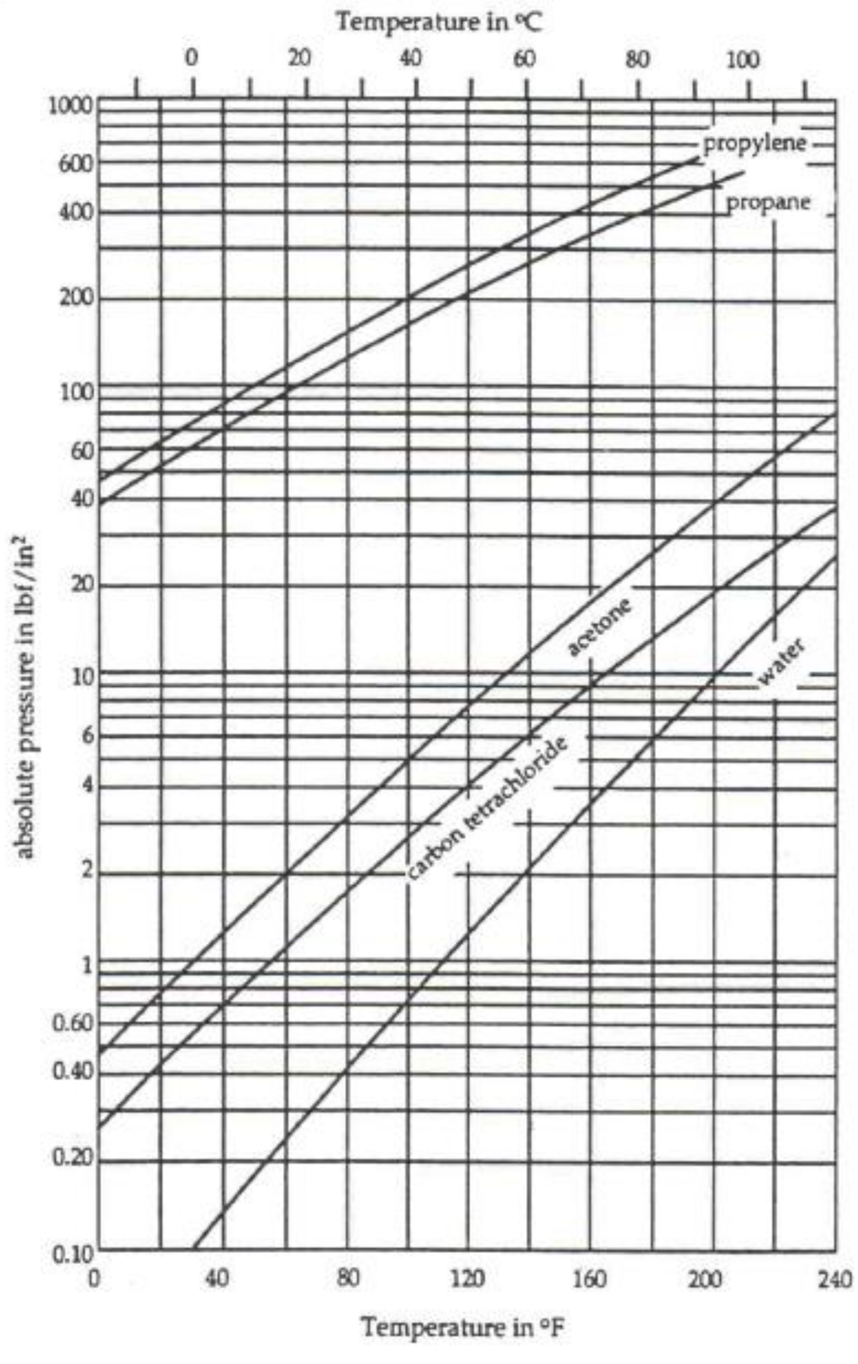


FIGURE 5.7. *Vapor pressure vs temperature for various liquids. (Data from several sources.)*

Net Positive Suction Head

Net positive suction head is an indication if cavitation will occur in a pump installed into a system.

Two configuration is shown.

- **Suction lift**: The liquid level in the tank is below the impeller centerline
- **Suction head**: The liquid level in the tank is above the pump impeller centerline.

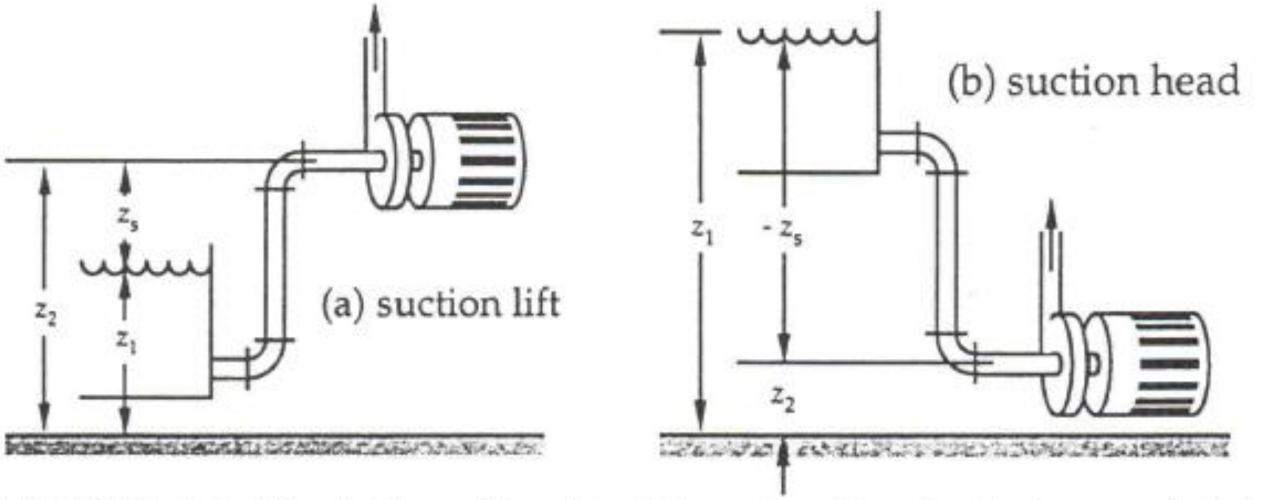


FIGURE 5.6. *Illustration of suction lift and suction head at pump inlet.*

Our objective is to determine the **pressure at the inlet of the pump** and compare it to the **vapor pressure** of the liquid at the local temperature. If the pressure at the pump inlet is **less than the vapor pressure** of the liquid at the local temperature, thus the pump will cavitate.

For the suction lift case (Figure a), apply the modified Bernoulli equation between a point at the free surface of the tank (point 1) and a point at the pump inlet (point 2).

$$\frac{P_1 g_c}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2 g_c}{\rho g} + \frac{V_2^2}{2g} + z_2 + \left(\sum f \frac{L}{D} \frac{V^2}{2g} + \sum K \frac{V^2}{2g} \right)$$

Although $\mathbf{p_1}$ is atmospheric, we will not set it equal to zero. This will allow our final equation to account for an overpressure on the liquid surface.

Evaluation of the variables yields

$V_1 = 0$ $z_2 - z_1 = +z_s$ $V_2 = V$ = velocity in the pipe.

Rearranging this equation, we get

$$\frac{P_2 g_c}{\rho g} = \frac{P_1 g_c}{\rho g} - z_s - \left(\sum f \frac{L}{D} + \sum K + 1\right)\frac{V^2}{2g}$$

Subtracting the vapor pressure from the both sides of the equation and rearranging, we obtain

$$\frac{P_2 g_c}{\rho g} - \frac{P_v g_c}{\rho g} = \frac{P_1 g_c}{\rho g} - z_s - \left(\sum f \frac{L}{D} + \sum K + 1\right)\frac{V^2}{2g} - \frac{P_v g_c}{\rho g}$$

or

$$NPSH_a = \frac{P_2 g_c}{\rho g} - \frac{P_v g_c}{\rho g} = \frac{P_1 g_c}{\rho g} - z_s - \left(\sum f \frac{L}{D} + \sum K + 1\right)\frac{V^2}{2g} - \frac{P_v g_c}{\rho g}$$

Left side of the preceding equation is defined as the net positive suction head available, NPSHa

<u>For figure b (suction head case), we obtain,</u>

$$NPSH_a = \frac{P_2 g_c}{\rho g} - \frac{P_v g_c}{\rho g} = \frac{P_1 g_c}{\rho g} + z_s - \left(\sum f \frac{L}{D} + \sum K + 1\right)\frac{V^2}{2g} - \frac{P_v g_c}{\rho g}$$

In some text books, net positive suction head available is written as

$$h_{p2} - h_{vp} = NPSH_a = h_{p1} - h_{z2} - h_f - h_{vp}$$

Pump manufacturers perform tests on pumps and report the values of net positive suction head required, NPSHr. Cavitation is prevented when the available net positive suction head is grater than the required net positive suction head, i. e.

$$NPSH_a \succ NPSH_r$$

NOTE: In practical applications, to prevent the cavitation, **NPSHa>NPSHr** by a margin of safety **3 ft or 1 m**.

Example: A pump delivers **900 gpm** of water from a tank at a head difference ΔH of **8 ft**. The net positive suction head required is **10 ft**. Determine where the pump inlet should be placed with respect to the level of water in the tank. The water surface is exposed to atmospheric pressure. Neglect the frictional effects and take the water temperature to be 90 ^oF.

Solution:

Water $\rho = 62.4 \text{ lbm/ft}^3$

$p_v = 0.55$ lbf/in2 at 90 ^oF

$P_{atm} = 14.7 \text{ lbf/in}^2$

DIMENSIONAL ANALYSIS OF PUMPS

A dimensional analysis can be performed for pumps, and the results ca be used to analyze the pumps under different conditions and as an aid in selecting the pump type for a specific application.

Using dimensional analysis, we wish to express **efficiency, η**, **energy transfer rate, g∆H**, **power, dW/dt** in terms of dimensionless variables.

$$\eta = f_1(\rho, \mu, Q, \omega, D, g_c)$$

$$g\Delta H = f_2(\rho, \mu, Q, \omega, D, g_c)$$

$$\frac{dW}{dt} = f_3(\rho, \mu, Q, \omega, D, g_c)$$

Using **Pi theorem** (see a fluid mechanics book), following dimensionless expressions are obtained.

$$\eta = g_1\left(\frac{\rho\omega D^2}{\mu g_c}, \frac{Q}{\omega D^3}\right)$$

$$\frac{g\Delta H}{\omega^2 D^2} = g_2\left(\frac{\rho\omega D^2}{\mu g_c}, \frac{Q}{\omega D^3}\right)$$

$$\frac{g_c(dW/dt)}{\rho\omega^3 D^5} = g_3\left(\frac{\rho\omega D^2}{\mu g_c}, \frac{Q}{\omega D^3}\right)$$

$$\frac{g\Delta H}{\omega^2 D^2} = energy\ transfer\ coefficient$$

$$\frac{Q}{\omega D^3} = volumetric\ \ flow\ coefficient$$

$$\frac{\rho\omega D^2}{\mu g_c} = rotational\ reynolds\ number$$

$$\frac{g_c (dW / dt)}{\rho\omega^3 D^5} = power\ coefficient$$

Experiments show that the **rotational Reynolds** number has a smaller effect on the dependent variables than does the **flow coefficient**. Therefore, for incompressible flow through the pumps, the above dimensionless equations can be written as

$$\eta \approx g_1\left(\frac{Q}{\omega D^3}\right)$$

Similarity laws or affinity laws

$$\frac{g\Delta H}{\omega^2 D^2} \approx g_2\left(\frac{Q}{\omega D^3}\right)$$

$$\frac{g_c (dW / dt)}{\rho\omega^3 D^5} \approx g_3\left(\frac{Q}{\omega D^3}\right)$$

The above relations are useful expressions and dimensionless groups. Suppose that performance data are available for a particular pump operating under certain conditions. Using the above expressions and the available data, the **performance of the pump** can be predicted when something has been changed, such as rotational speed, impeller diameter, volume flow rate or fluid density.

For **similar pumps**, pump affinity laws might be written as:

$$\left(\frac{Q}{\omega D^3}\right)_1 = \left(\frac{Q}{\omega D^3}\right)_2$$

$$\left(\frac{g\Delta H}{\omega^2 D^2}\right)_1 = \left(\frac{g\Delta H}{\omega^2 D^2}\right)_2$$

$$\left[\frac{(dW/dt)}{\rho\omega^3 D^5}\right]_1 = \left[\frac{(dW/dt)}{\rho\omega^3 D^5}\right]_2$$

Example: Actual performance data on a centrifugal pump are as follows:

Rotational speed = ω = 3500 rpm
Total head difference = ΔH = 80 ft
Volume flow rate = Q = 50 gpm
Impeller diameter = D = $5^{1/5}$ in.
fluid = water

It is desired to change the rotational speed to 1750 rpm and the impeller diameter to $4^{5/8}$ in. Determine how the new configuration will affect the pump performance with water as the working fluid.

SPECIFIC SPEED AND PUMP TYPES

It is necessary to have some criteria regarding determination of the type of the pump to use for a specific application.

A dimensionless group known as **specific speed** is used in **decision making** process of **determining the type of the pump**.

Specific speed defined as the speed required for a machine to produce unit head at unit volume flow rate.

Specific speed is found by **combining head coefficient and volumetric flow coefficient** in order to eliminate characteristic length D.

$$\omega_{ss} = \left(\frac{Q}{\omega D^3}\right)^{1/2} \left(\frac{\omega^2 D^2}{g\Delta H}\right)^{3/4}$$

$$or \quad \omega_{ss} = \frac{\omega Q^{1/2}}{(g\Delta H)^{3/4}}$$

ω_{ss}: dimensionless

Q: ft^3 or m^3/s

ω: rad/s

ΔH: ft or m

Another definition for specific speed is given by

$$\omega_s = \frac{\omega Q^{1/2}}{\Delta H^{3/4}}$$

ω_s: rpm

Q: gpm

ω: rpm

ΔH: ft

Specific speed of a pump can be calculated at any operating point. However, It is advantageous to calculate specific speed for a pump only at maximum efficiency point.

With data obtained from tests on many types of pumps (including axial, mixed and radial or centrifugal), the data given in the table below have been produced. When this table is used, a machine that operates at or near its maximum efficiency is selected.

TABLE 5.3. *Pump selection chart for finding efficiency in %.*

ω_s rpm	Flow Rate in Gallons per Minute, gpm							ω_{ss}	Pump Type
	100	200	500	1000	3000	10000	>10000		
500	46	54						0.18	Centrifugal or radial flow type
600	52	58						0.22	
700	55	62	70					0.26	
800	58	64	72	76				0.29	
900	61	66	74	77	81			0.33	
1000	62	67	75	78	82			0.37	
1500	67	72	79	82	85	87	89	0.55	Mixed flow type
2000	70	75	81	83	87	89	92	0.73	
3000			81	83	87	89	92	1.10	
4000			79	81	85	88	90	1.46	
5000					84	87	90	1.83	
6000					83	85	87	2.19	
7000					82	84	86	2.56	
8000					80	84	86	2.92	
9000						83	85	3.29	
10000						82	84	3.65	Axial flow type
15000						77	80	5.48	

Equations:

$$\omega_{ss} = \frac{\omega Q^{1/2}}{(g\Delta H)^{3/4}} \qquad \text{[dimensionless]} \tag{5.22}$$

$$\omega_s = \frac{\omega\, Q^{1/2}}{\Delta H^{\,3/4}} \qquad \left[\text{rpm} = \frac{\text{rpm}(\text{gpm})^{1/2}}{\text{ft}^{3/4}}\right] \tag{5.23}$$

Example: Determine the type of pump best suited for pumping **250 gpm (=0.557 ft^3/s)** of water with a corresponding head of **6 ft**. The motor to be used has a rotational speed of **360 rpm**. Also calculate the power transmitted to the fluid and the power required to be transferred from the motor.

Piping System Design Practices

Piping system can be one of the greatest cost item in an installation. Hence, the piping should be designed to meet minimum cost requirements and still adequate for meeting the operational requirements.

Followings are design practices that help an engineer in the decision making process.

- At pressure drops greater than
  25-30 psi per 1000 ft (175-200 kPa per 300 m) of pipe line for liquids
  10-15 psi per 1000 ft (70-104 kPa per 300 m) of pipe line for gases,
  excessive and objectionable vibrations in the system will results.

- Usually when the economic pipe diameter is calculated, the results falls between two nominal sizes. Selecting the smaller size results in a lower initial capital investment. Selecting the larger size results in a lower operating cost. From the engineering standpoint, the larger size leads to more design flexibility.

- In all piping systems, it is possible that air will be trapped somewhere in the line. It is advisable to layout pipelines with a slight grade upward in the flow direction so that air will tend not to remain in the line. Where this is not possible, a small valve should be installed at places where air (or vapor) might tend to accumulate.

Piping System Design – Suggested Procedure

1. Determine the economic line size. Use the calculated economic diameter to find the optimum **economic velocity** from the table below. Use economic velocity to complete the details of the system design.

TABLE 5.4. *Reasonable velocities for various fluids, calculated by using optimum economic diameter equations.*

Fluid	Economic Velocity Range ft/s	Economic Velocity Range m/s
Acetone	4.9–9.8	1.5–3.0
Ethyl Alcohol	4.8–9.6	1.5–3.0
Methyl Alcohol	4.8–9.6	1.5–3.0
Propyl Alcohol	4.7–9.4	1.4–2.8
Benzene	4.6–9.2	1.4–2.8
Carbon Disulfide	4.2–8.4	1.3–2.6
Carbon Tetrachloride	3.9–7.8	1.2–2.4
Castor Oil	1.6–3.2	0.5–1.0
Chloroform	4.0–8.0	1.2–2.4
Decane	4.9–9.8	1.5–3.0
Ether	5.0–10.0	1.5–3.0
Ethylene Glycol	3.9–7.8	1.2–2.4
R-11	4.0–8.0	1.2–2.4
Glycerine	1.4–2.8	0.43–0.86
Heptane	5.1–10.2	1.5–3.0
Hexane	5.2–10.4	1.6–3.2
Kerosene	4.7–9.4	1.4–2.8
Linseed Oil	4.9–9.8	1.5–3.0
Mercury	2.1–4.2	0.64–1.3
Octane	5.0–10.0	1.5–3.0
Propane	5.6–11.2	1.7–3.4
Propylene	5.5–11.0	1.7–3.4
Propylene Glycol	4.5–9.0	1.4–2.8
Turpentine	4.6–9.2	1.4–2.8
Water	4.4–8.8	1.4–2.8

2. Calculate the pump power required for the system using the optimum economic line size. Check to ensure that the pressure drop is not excessive. Prepare a system curve of ΔH versus Q.

3. If pump is to be used, determine from the appropriate chart which pump should be selected. Refer to the pump performance map if available, and superimpose the system curve on it to find the exact operating point.

4. Use NPSH data to specify the exact location of the pump.

5. If tanks are present, specify the minimum and maximum liquid heights in them.

6. Prepare a drawing for the system and a summary of specification sheet that lists results of calculations only. Attach the calculations to the summary sheet.

Example: Piping system shown in the figure should convey **600 gpm** of propylene glycol from a tank open to atmospheric conditions. Follow the suggested design procedure and make recommendations about the piping system.

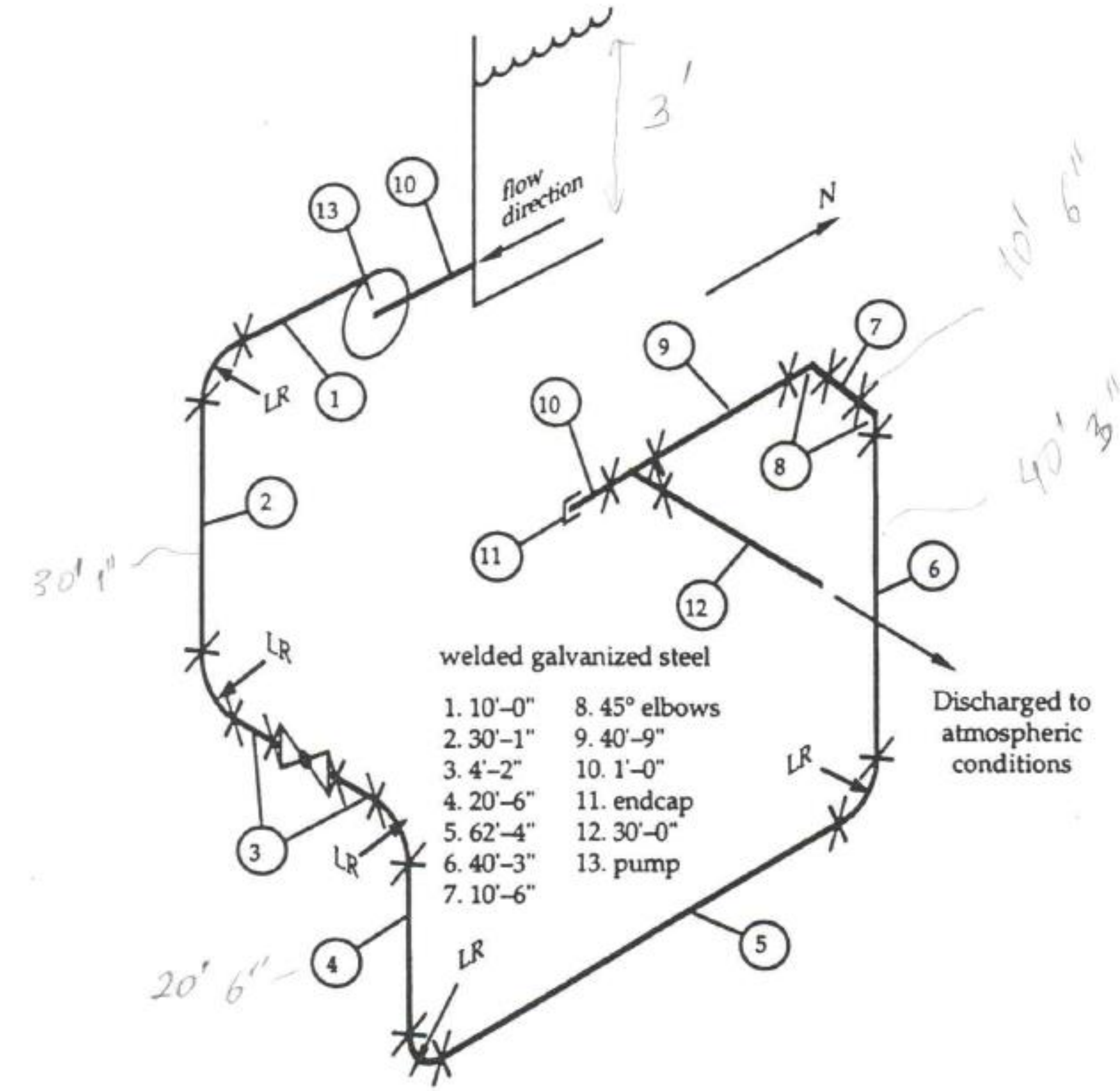


FIGURE 5.8. *The piping system of Example 5.6.*

Solution:

<u>Propylene glycol</u>

ρ = 0.968 (1.94) slug/ft^3 μ = 88 x 10-5 lbf s/ft^2 (Appendix Table B1)

<u>6-nominal schedule 40</u>

ID = D= 0.5054 ft A = 0.2006 ft^2 (Appendix Table D.1

<u>Galvanized surface</u> ε = 0.0005 (Table 3.1)

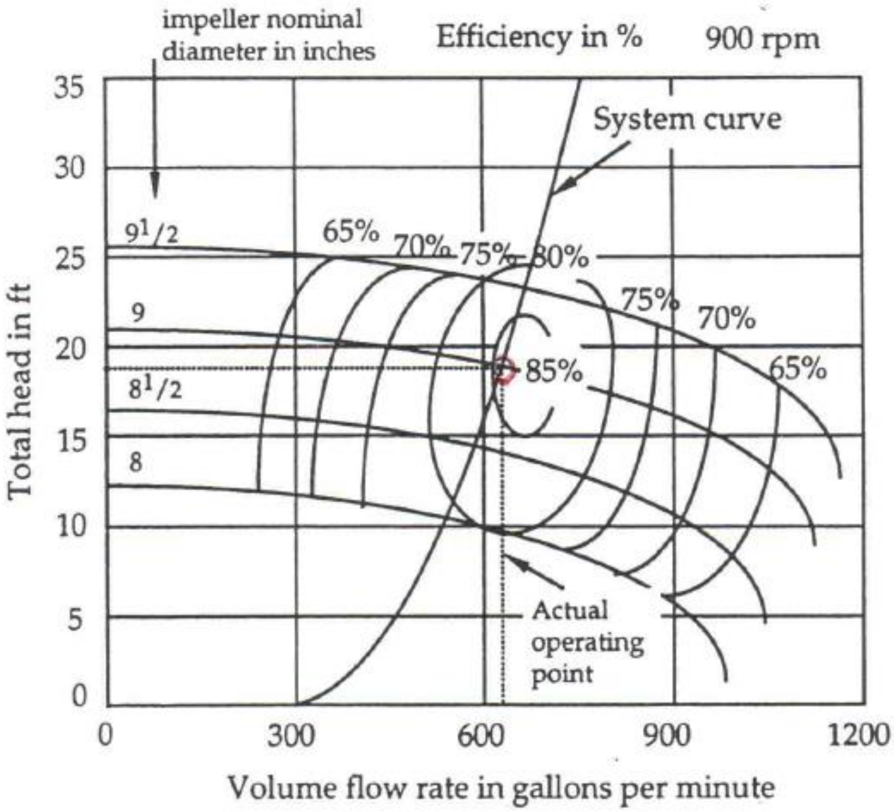


FIGURE 5.10. *System curve superimposed on performance map to find the actual operating point.*

6. Summary of Results

Economic line size	=	6-nominal schedule 40
Layout	=	Figure 5.8
System curve	=	Figure 5.9
Specific speed	=	2756 rpm at 600 gpm
Expected pump eff	=	~81%
Pump type	=	Mixed flow
Motor power	≈	3 hp