PUMPS

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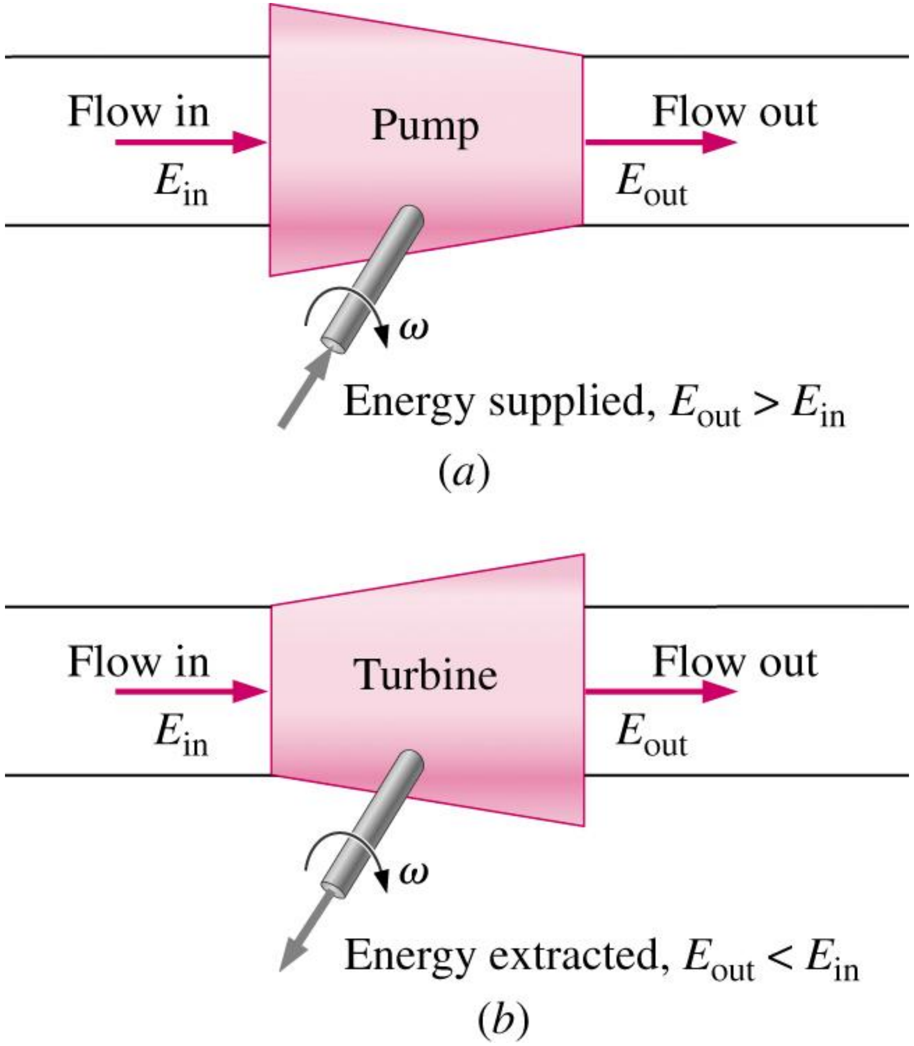
Contents

- Introduction
  - Positive displacement pumps
  - Turbomachine-type pumps
- Turbomachine
- Pump performance
- Net positive suction head
- System characteristics and pump selection
- Affinity law
- Pump specific speed

Introduction

- The **purpose** of this chapter is **to understand from a fluid mechanics standpoint how pumps work.**
- Pumps and turbines, or sometime called ***fluid machines***, occur in a wide variety of configuration.
- In general, **pumps add energy to fluid** (*they do work on the fluid*), whereas turbine extract energy from the fluid (the fluid does work on them).
- In this subject (SKTG 2343), we will only consider on pumps.
- The term “pump” will be used to generically refer to all pumping machines, including pumps, fans, blowers, and compressors.
- Pumps can be divided into two main categories: positive displacement pumps (denoted as the static type) and turbomachines or turbomachine-type pumps (denoted as the dynamic type).
- The majority of this chapter deals with **turbopumps.**

Introduction



- **Pump: adds** energy to a fluid, resulting in an increase in pressure across the pump
- **Turbine: extracts** energy from the fluid, resulting in a decrease in pressure across the turbine

Introduction

- As mentioned earlier, pumps add energy to fluid (*they do work on the fluid*)
- Pump is needed to move fluid
  - from $p\downarrow$ location to $p\uparrow$ location
  - from $z\downarrow$ location to $z\uparrow$ location

  or both
- Generally, **pumps are needed if the energy at upstream (beginning) of flow is not enough to overcome the energy at downstream** (ending) of flow
- Mathematically,

$$H_1 = H_2 + h_f + h_m$$

$$H = \frac{p}{\gamma} + \frac{v^2}{2g} + z$$

$$\text{if } H_1 < H_2 + h_f + h_m \quad \text{pump is needed}$$

$$\therefore\ H_1 + \Delta h = H_2 + h_f + h_m$$

where,

H_1 = total pressure per unit weight at point 1
H_2 = total pressure per unit weight at point 2
h_f = major losses
h_m = minor losses
Δh = energy added by the pump

Positive Displacement Pumps

- Positive displacement pumps force a fluid into or out a chamber by changing the volume of the chamber
- The pressures developed and the work done are a result of essentially static forces rather than dynamic effects
- Typical example shown in figure.
- **In these cases the devices does work on the fluid** (the container wall moves against the fluid pressure force on the moving wall)

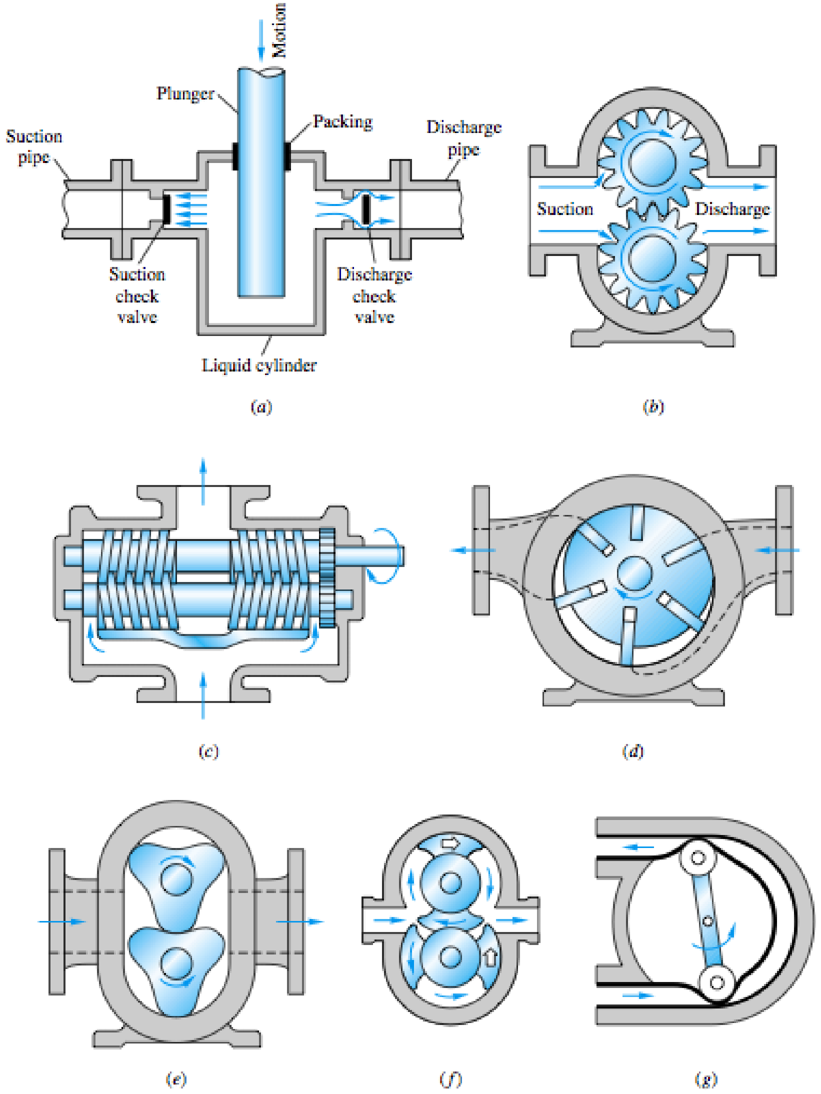


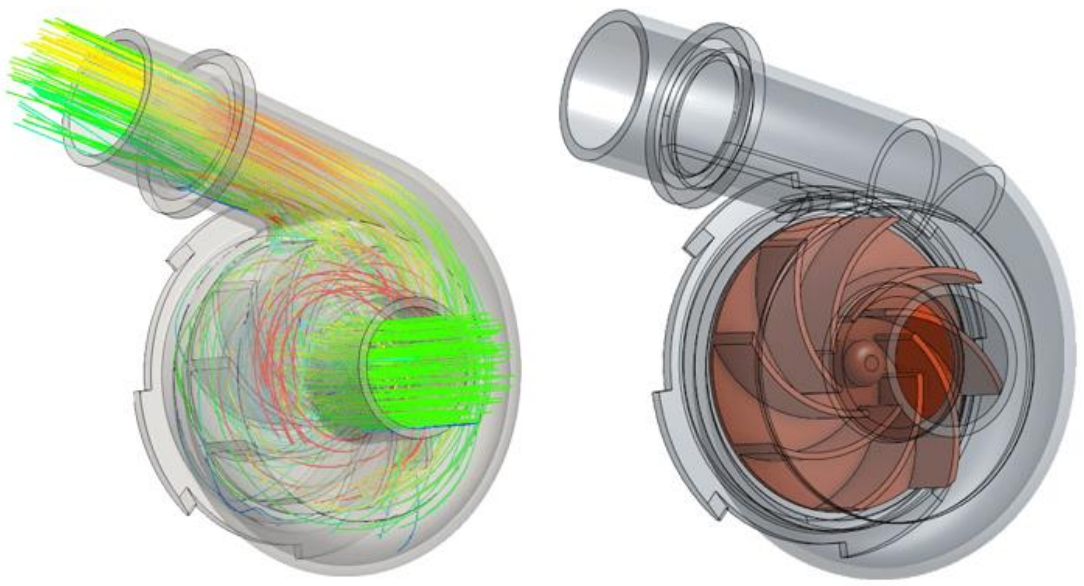
Figure Schematic design of positive displacement pumps:
(a) reciprocating piston or plunger
(b) external gear pump
(c) double-screw pump
(d) sliding vane
(e) three-lobe pump
(f) double circumferential piston
(g) flexible-tube squeegee

Turbomachines-type Pumps

- Turbomachine-type pumps **involve a collection of blades, buckets, flow channels, or passages arranged around axis of rotation to form rotor.**

- Rotation of the rotor produces dynamic effects that add energy to the fluid.

- Example of these **devices include simple windows fans, propellers on ships or airplanes, squirrels-cage fans on home furnaces, axial-flow water-pumps used in deep wells, and compressors in automobile turbochargers.**

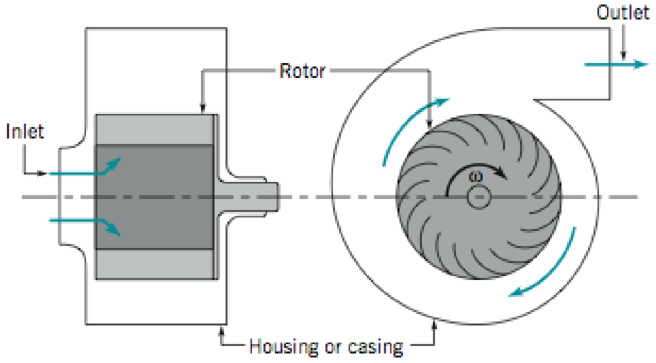


Turbomachines-type Pumps

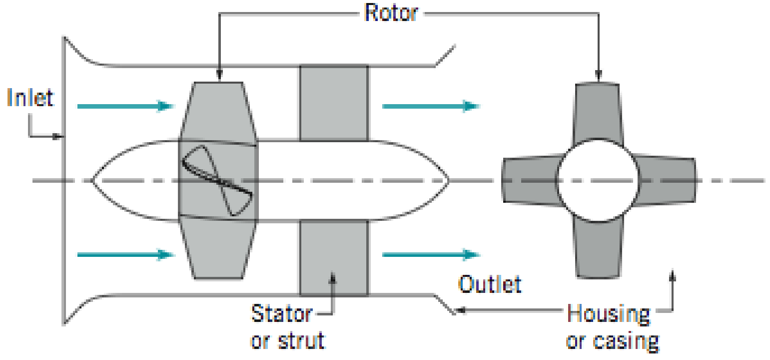
- Turbomachines **involve** the related parameters of **force, work, and power.**
- **Energy is supplied to the rotating shaft** by a motor and transferred to the fluid by the blades
- The fluid used can be either a gas (as with a window fan) or a liquid (as with the water pump)
- The basic operating principles are the same whether the fluid is a liquid or a gas, important differences in the fluid dynamics involved can occur
- For example, ***cavitation*** may be an important design consideration when liquids are involved if the pressure at any point within the flow is reduced to the vapor pressure. ***Compressibility*** effects may be important when gases are involved if the Mach number becomes large enough

Turbomachines-type Pumps

- Many turbomachines contain some type of housing or casing that surrounds the rotating blades or rotor, thus forming an internal flow passageway through which the fluid flows
- Turbomachines are classified as ***axial*** *flow*, ***radial*** *flow,* or ***mixed*** *flow* machines depending on the predominant direction of the fluid motion relative to the rotor's axis as the fluid passes the blades (Figure)



(a) Radial-flow fan



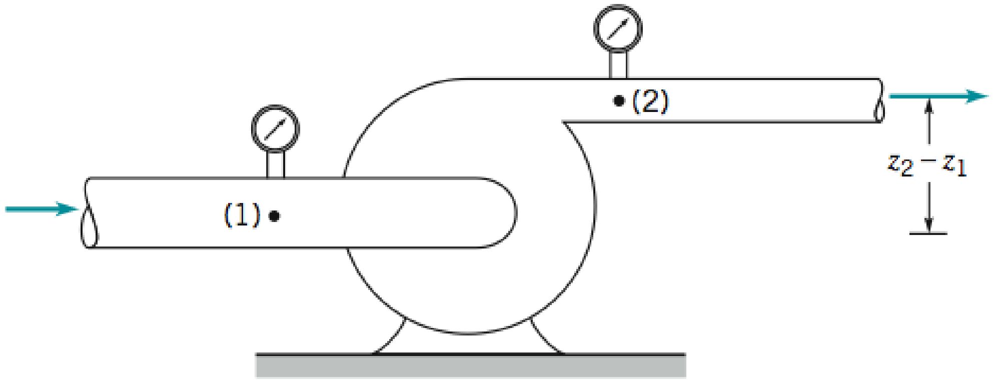
(b) Axial-flow fan

a. For a ***radial-flow machine*** the flow across the blades involves a substantial radial-flow component at the rotor inlet, exit, or both. (e.g. ***centrifugal pump***)
b. For an ***axial-flow machine*** the fluid maintains a significant axial-flow direction component from the inlet to outlet of the rotor. (e.g. ***propeller pump***)
c. For a ***mixed-flow machine*** is a combination of both machines, has significant radial- and axial flow velocity components for the flow through the rotor row.

➢ Each type of machine has **advantageous** and **disadvantageous** for different applications and in term of **fluid-mechanical performance**

Pump Performance

- The actual head rise, h_a, gained by fluid flow through a pump can be determined with an experimental arrangement as shown in figure



Typical experimental arrangement for determining the head rise gained by a fluid flowing through a pump.

- By using energy equation with $h_a = h_s - h_L$ (where **h_s is the shaft work head** and **h_L is the pump head loss**)

$$h_a = \frac{p_2 - p_1}{\gamma} + z_2 - z_1 + \frac{v_2^2 - v_1^2}{2g}$$

- The head, h_a, is the same as h_p used with Bernoulli's equation
- Typically, the differences in elevations and velocities are small, so that

$$h_a \approx \frac{p_2 - p_1}{\gamma}$$

Pump Performance

- The power, P_f, gained by the fluid is given by the equation

$$P_f = \gamma Q h_a$$

and this quantity, expressed in terms of horsepower is

$$P_f = \text{horsepower} = \frac{\gamma Q h_a}{550} \quad ----(1)$$

with γ →lb/ft^3, Q→ft^3/sec, and h_a→ft [In SI Unit: with γ →N/m^3, Q→m^3/sec, and h_a→m, $hp = \frac{\gamma Q h_a}{746}$

1 hp = 550 lb.ft = 746 watt

- If the pumped fluid is not water, the *g* appearing in Eqn. (1) must be the specific weight of the fluid moving through the pump
- The overall efficiency, η, is

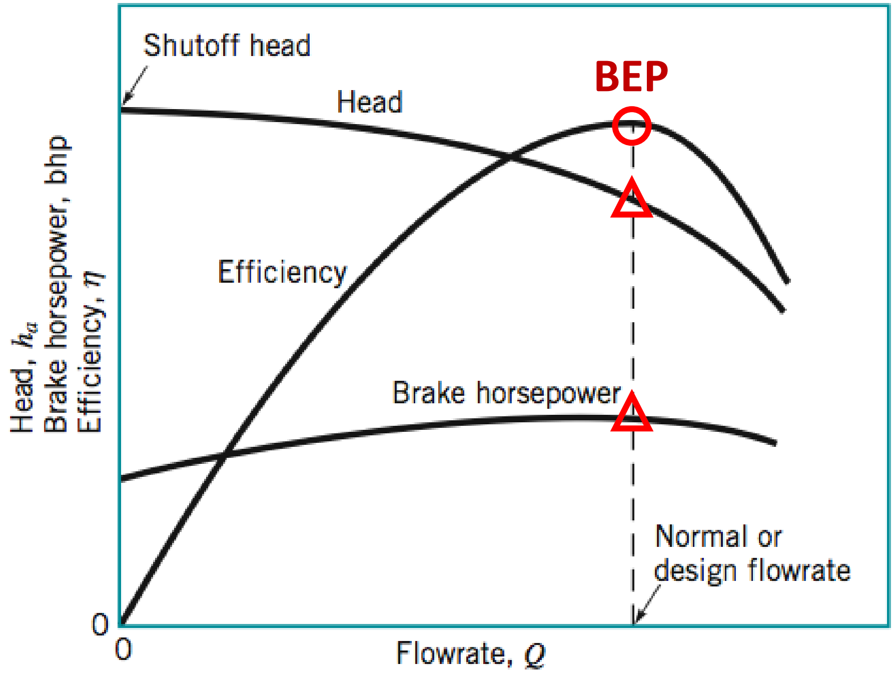
$$\eta = \frac{\text{power gained by the fluid}}{\text{power driving the pump}} = \frac{P_f}{W}$$

- The denominator of this relationship represents the total power applied to the pump and is often referred to as ***brake horsepower* (bhp).** Thus

$$\eta = \frac{\gamma Q h_a / 550}{\text{bhp}}$$

Pump Performance

- Performance characteristics for a given pump geometry and operating speed are usually given in the form of plots of h_a, η, and bhp versus Q (commonly referred to as ***capacity***)

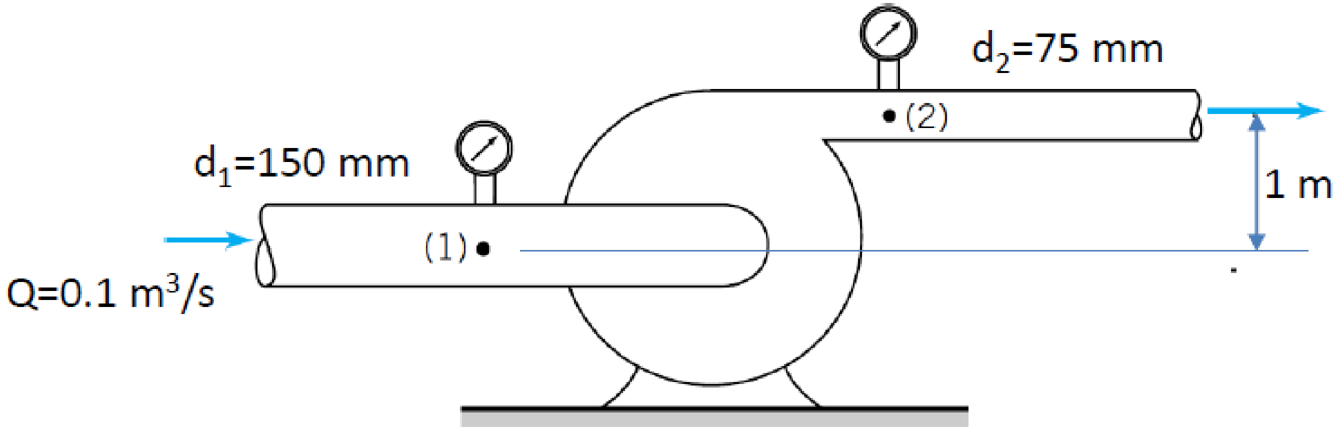


Typical performance characteristics for a centrifugal pump of a given size operating at a constant impeller speed

- The η is a function of Q and reaches a maximum value at some particular value of Q, commonly referred to as **the *normal*** or ***design Q*** or **capacity for the pump**
- The point on the various curves corresponding to the maximum efficiency are denoted as the ***best efficiency point*** **(BEP)**
- Performance curves of pumps are very important to engineer for selection of pumps for a particular flow system

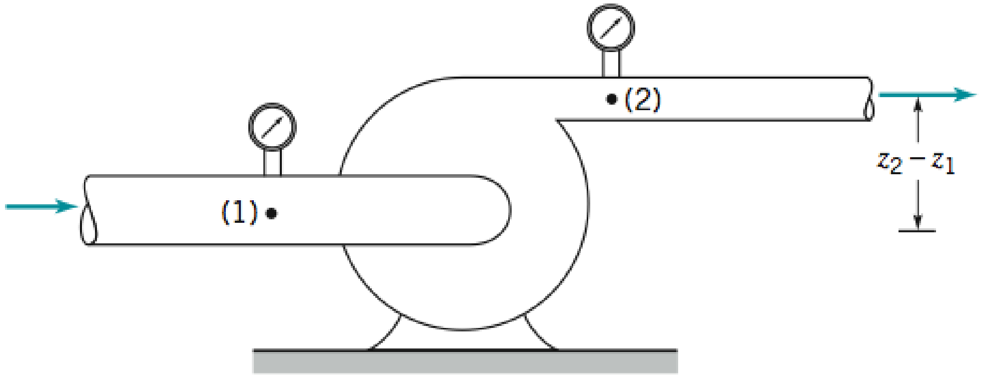
Example 6.1

A pump used to pump water between two reservoirs. If water is discharged at $0.1\ \mathrm{m^3/s}$ through diameter pipes of 150 mm (suction) and 75 mm (discharge), determine power gained by the water. Assume the height between suction and discharge point is 1 m, outlet pressure is 70 kPa above inlet pressure and water specific weight is $9810\ \mathrm{N/m^3}$.

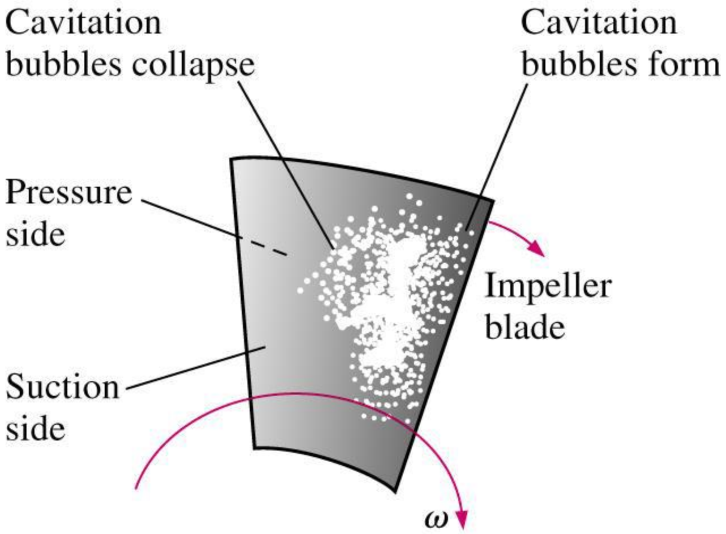


Net Positive Suction Head (NPSH)

- On the suction side of a pump, low pressures are commonly encountered, which possibly ***cavitation*** will occurs within the pump.
- **Cavitation occurs when the liquid pressure at given location is reduced to the vapor pressure of the liquid**. When this occurs, vapor bubbles form, and this phenomenon can cause a loss in η as well as structural damage to the pump.
- To avoid cavitation, **we must ensure that the local pressure everywhere inside the pump stay above the vapor pressure.**
- In order to avoid cavitation from occurring, firstly, the difference between the total head on the suction head, $p_s/\gamma + v_s^2/2g$, and the liquid vapor pressure head, p_v/γ, has to be known.
- This difference is called the ***net positive suction head* (NPSH),** or mathematically



$$\boxed{\mathrm{NPSH} = \frac{p_s}{\gamma} + \frac{v_s^2}{2g} - \frac{p_v}{\gamma}}$$



- There are actually two values of NPSH
  - ***Required* NPSH**, denoted as NPSH_R
  - ***Available* NPSH**, denoted as NPSH_A

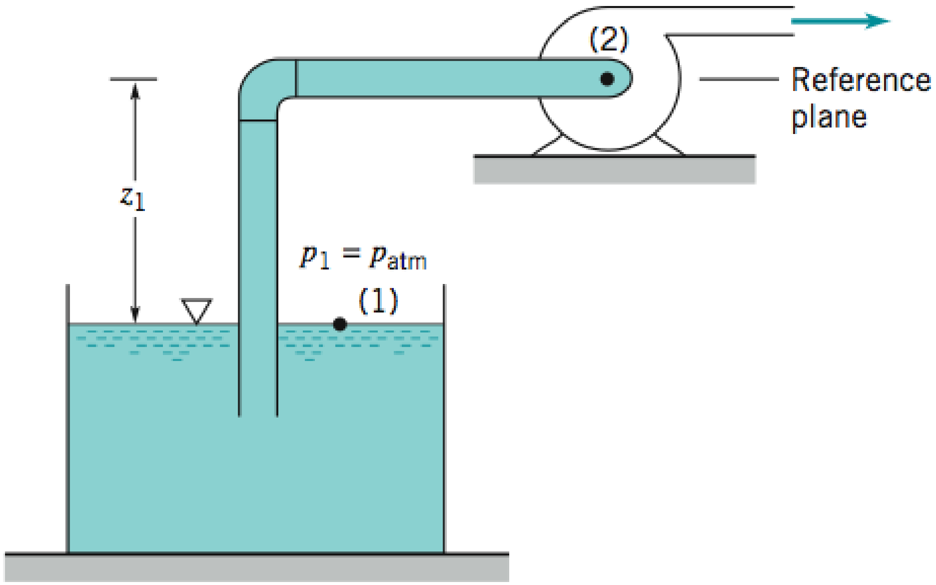
Net Positive Suction Head (NPSH)

- $NPSH_R$ must be maintained, or exceeded, so that cavitation will not occur.
- Pumps are tested experimentally (manufacturer) to determine the value for $NPSH_R$.
- $NPSH_A$ represents the head that actually occurs for the particular flow system
- This value can be determined experimentally, or calculated if the system parameters are known
- For example consider a typical flow system below

By applying Bernoulli's eqn. from 1 to 2, yields

$$\frac{p_{atm}}{\gamma} - z_1 = \frac{p_s}{\gamma} + \frac{v_s^2}{2g} + \sum h_L$$

where Σh_L represents head losses between the free surface and the pump impeller inlet



Net Positive Suction Head (NPSH)

- Thus, the head available at the pump impeller inlet is

$$\frac{p_s}{\gamma} + \frac{v_s^2}{2g} = \frac{p_{atm}}{\gamma} - z_1 - \sum h_L$$

so that,

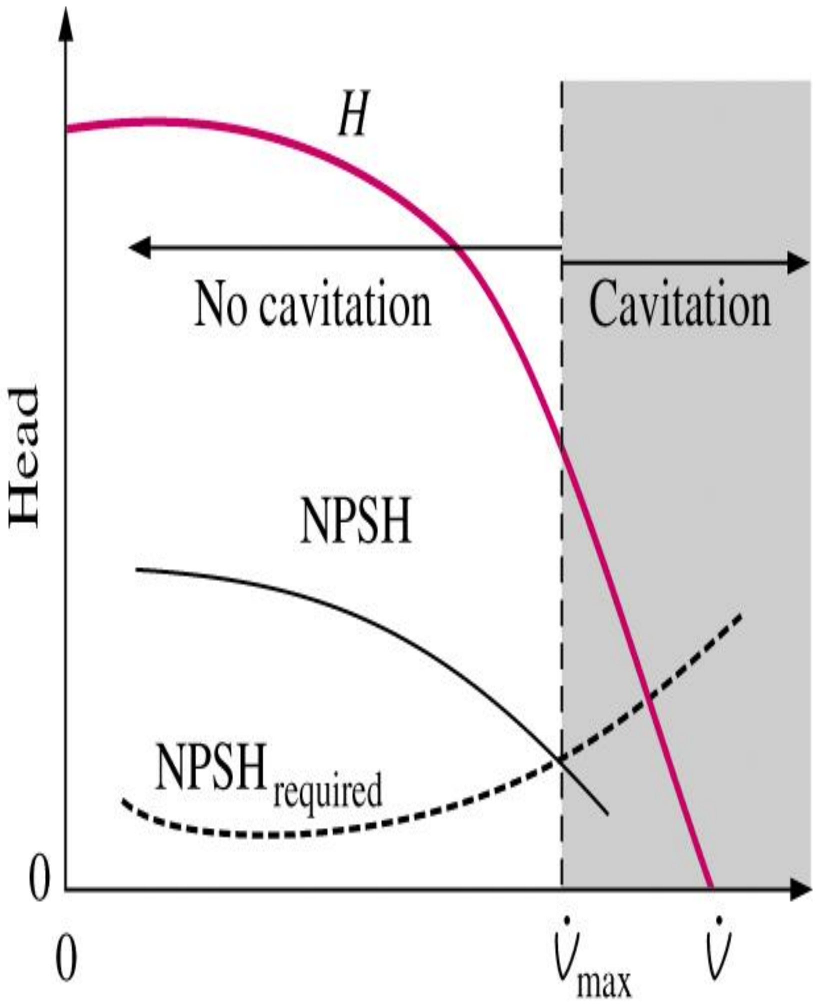
$$\mathrm{NPSH_A} = \frac{p_{atm}}{\gamma} - z_1 - \sum h_L - \frac{p_v}{\gamma} \qquad ----(1)$$

- For this calculation, absolute pressure are normally used, since the vapor pressure is usually specified as an absolute pressure
- For proper pump operation, it is necessary that

$$\boxed{\mathrm{NPSH_A} \geq \mathrm{NPSH_R}}$$

- From Eqn. (1), as the height of the pump above the fluid surface, z_1, is increased, the $\mathrm{NPSH_A}$ is decreased
- Therefore, there is some critical value for z_1 above which the pump cannot operate without cavitation
- The specific value depends on the head losses and the value of the vapor pressure.
- The physical properties of water is given in Table 6.1 (BG units) and Table 6.2 (SI units)

Net Positive Suction Head (NPSH)



Net Positive Suction Head (NPSH)

Table 6.1 Physical properties of water (BG units)

Temperature (°F)	Density, ρ (slugs/ft³)	Specific Weight[b], γ (lb/ft³)	Dynamic Viscosity, μ (lb·s/ft²)	Kinematic Viscosity, ν (ft²/s)	Surface Tension[c], σ (lb/ft)	Vapor Pressure, p_v [lb/in.²(abs)]	Speed of Sound[d], c (ft/s)
32	1.940	62.42	3.732 E − 5	1.924 E − 5	5.18 E − 3	8.854 E − 2	4603
40	1.940	62.43	3.228 E − 5	1.664 E − 5	5.13 E − 3	1.217 E − 1	4672
50	1.940	62.41	2.730 E − 5	1.407 E − 5	5.09 E − 3	1.781 E − 1	4748
60	1.938	62.37	2.344 E − 5	1.210 E − 5	5.03 E − 3	2.563 E − 1	4814
70	1.936	62.30	2.037 E − 5	1.052 E − 5	4.97 E − 3	3.631 E − 1	4871
80	1.934	62.22	1.791 E − 5	9.262 E − 6	4.91 E − 3	5.069 E − 1	4819
90	1.931	62.11	1.500 E − 5	8.233 E − 6	4.86 E − 3	6.979 E − 1	4960
100	1.927	62.00	1.423 E − 5	7.383 E − 6	4.79 E − 3	9.493 E − 1	4995
120	1.918	61.71	1.164 E − 5	6.067 E − 6	4.67 E − 3	1.692 E + 0	5049
140	1.908	61.38	9.743 E − 6	5.106 E − 6	4.53 E − 3	2.888 E + 0	5091
160	1.896	61.00	8.315 E − 6	4.385 E − 6	4.40 E − 3	4.736 E + 0	5101
180	1.883	60.58	7.207 E − 6	3.827 E − 6	4.26 E − 3	7.507 E + 0	5195
200	1.869	60.12	6.342 E − 6	3.393 E − 6	4.12 E − 3	1.152 E + 1	5089
212	1.860	59.83	5.886 E − 6	3.165 E − 6	4.04 E − 3	1.469 E + 1	5062

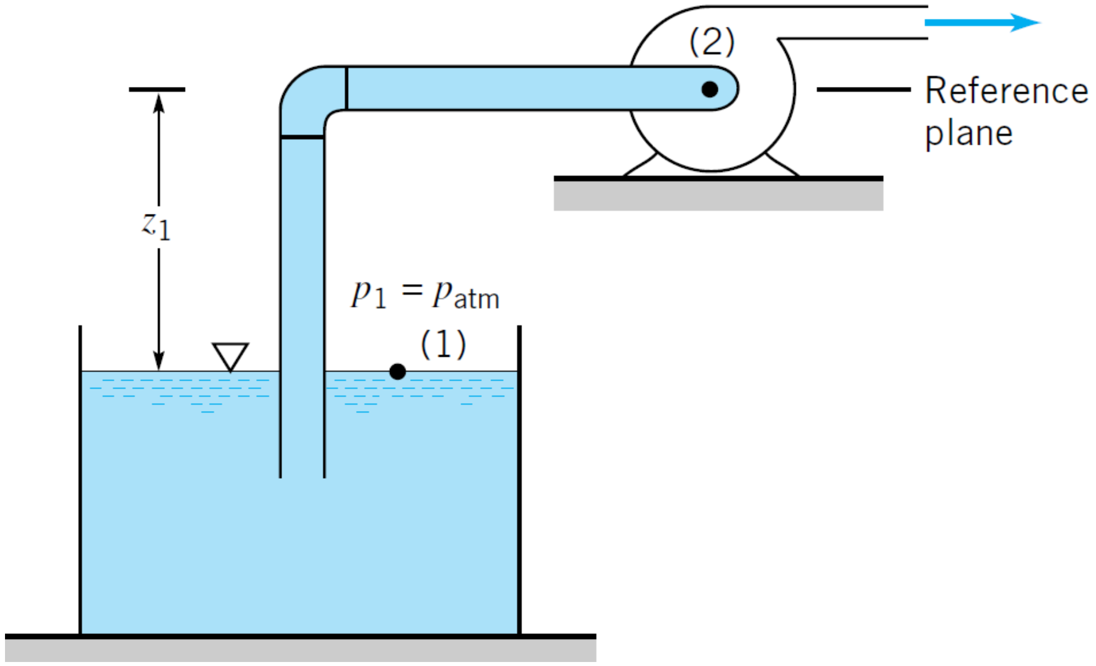
Net Positive Suction Head (NPSH)

Table 6.2 Physical properties of water (SI units)

Temperature (°C)	Density, ρ (kg/m^3)	Specific Weight[b], γ (kN/m^3)	Dynamic Viscosity, μ (N·s/m^2)	Kinematic Viscosity, ν (m^2/s)	Surface Tension[c], σ (N/m)	Vapor Pressure, p_v [N/m^2(abs)]	Speed of Sound[d], c (m/s)
0	999.9	9.806	1.787 E − 3	1.787 E − 6	7.56 E − 2	6.105 E + 2	1403
5	1000.0	9.807	1.519 E − 3	1.519 E − 6	7.49 E − 2	8.722 E + 2	1427
10	999.7	9.804	1.307 E − 3	1.307 E − 6	7.42 E − 2	1.228 E + 3	1447
20	998.2	9.789	1.002 E − 3	1.004 E − 6	7.28 E − 2	2.338 E + 3	1481
30	995.7	9.765	7.975 E − 4	8.009 E − 7	7.12 E − 2	4.243 E + 3	1507
40	992.2	9.731	6.529 E − 4	6.580 E − 7	6.96 E − 2	7.376 E + 3	1526
50	988.1	9.690	5.468 E − 4	5.534 E − 7	6.79 E − 2	1.233 E + 4	1541
60	983.2	9.642	4.665 E − 4	4.745 E − 7	6.62 E − 2	1.992 E + 4	1552
70	977.8	9.589	4.042 E − 4	4.134 E − 7	6.44 E − 2	3.116 E + 4	1555
80	971.8	9.530	3.547 E − 4	3.650 E − 7	6.26 E − 2	4.734 E + 4	1555
90	965.3	9.467	3.147 E − 4	3.260 E − 7	6.08 E − 2	7.010 E + 4	1550
100	958.4	9.399	2.818 E − 4	2.940 E − 7	5.89 E − 2	1.013 E + 5	1543

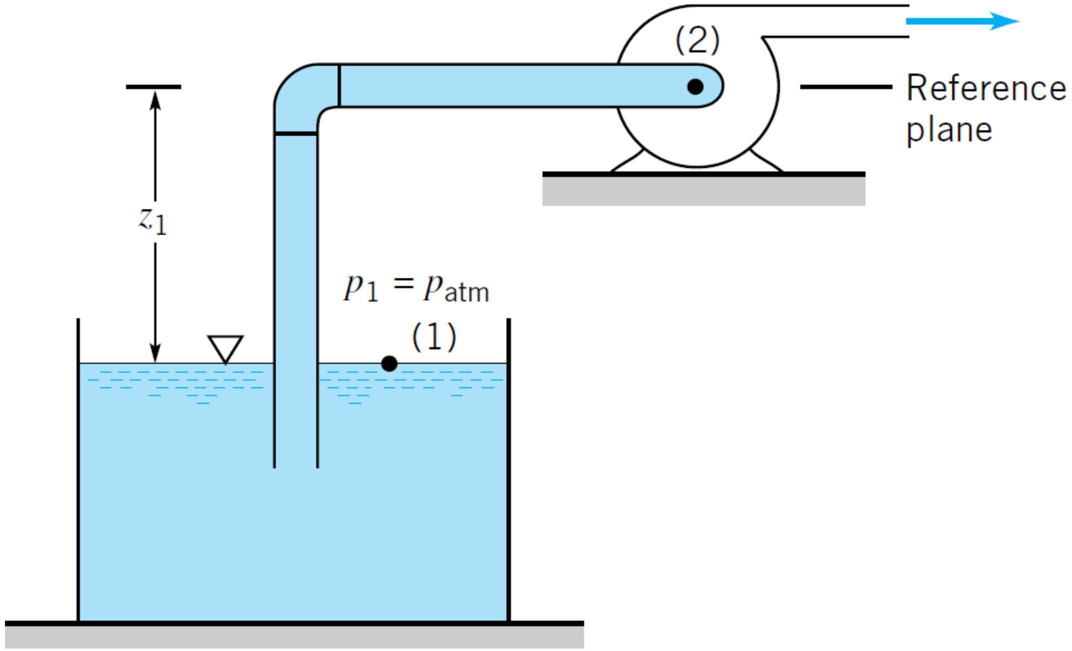
Example 6.2

A pump is placed so that its centerline is 1.5 m above the level in a reservoir that is open to the atmosphere. If water is being pumped at the rate of 1 m^3/s in the suction line and friction losses are estimated at 1.2 m, determine the NPSH if the water is at 40°C. Repeat the calculation if the pump centerline is 1.5 m below the level.



Example 6.3

A manufacturer states that the required NPSH for a centrifugal pump is 7 m. Water is at 20°C. If the head loss up to the pump inlet is 2 m, what should the placement of the pump be (z1) to avoid cavitation?

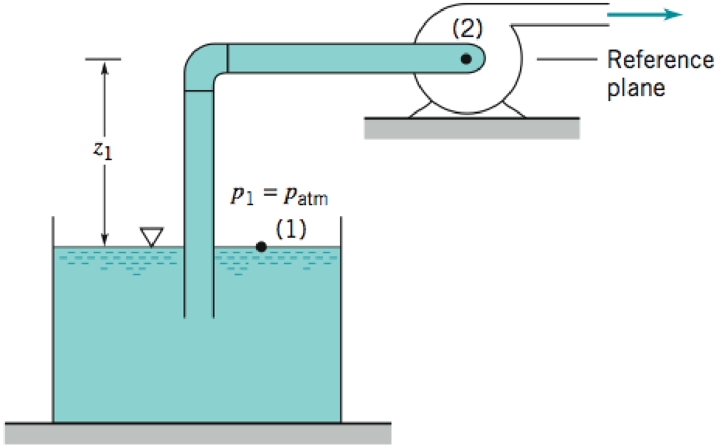


Example 6.4

A centrifugal pump is to be placed above a large, open water tank, as shown in Figure, and is to pump water at a rate of 0.5 ft^3/s. At this flow rate the required net positive suction head, $\mathrm{NPSH_R}$, is 15 ft, as specified by the pump manufacturer. If the water temperature is 80°F and atmospheric pressure is 14.7 psi, determine the maximum height, z_1, that the pump can be located above the water surface without cavitation.

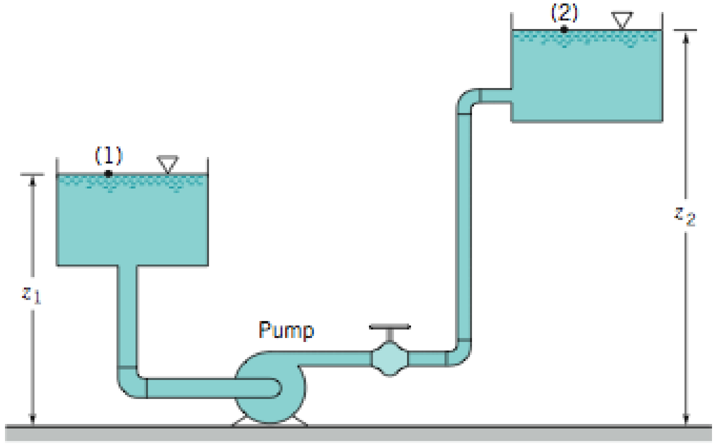
Assume that the major head loss between the tank and the pump inlet is due to filter at the pipe inlet having a minor loss coefficient K_L= 20. Other losses can be neglected.

The pipe on the suction side of the pump has a diameter of 4 inches.



System Characteristics and Pump Selection

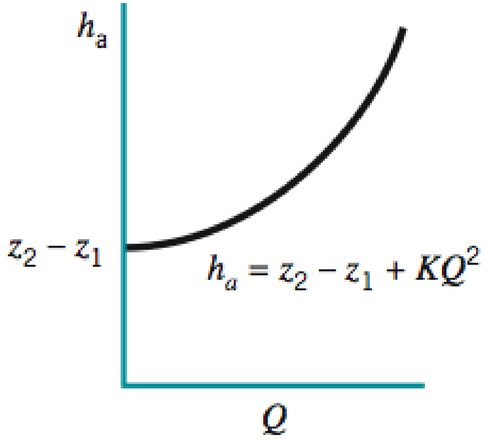
- A typical flow system in which a pump is used is shown below



- The Bernoulli's equation applied between points (1) and (2) indicates that

$$h_p = z_2 - z_1 + \sum h_L \quad ----(1)$$

where hp is the actual head gained by the fluid from the pump, and Σh_L represents all friction losses in the pipe and minor losses for pipe fittings and valves



- From the study of pipe flow, $h_L \infty Q^2$

System Characteristics and Pump Selection

- Thus, Eqn. (1) can be written in the form

$$h_p = z_1 - z_2 + KQ^2 \qquad ----(2)$$

where K depends on the pipe sizes and lengths, friction factors, and minor loss coefficients

- Eqn. (2) is the ***system equation*** and shows how the actual head gained by the fluid from the pump is related to the system parameters
- In this case the parameters include the change in elevation head, $z_2 - z_1$, and the losses due to friction as expressed by KQ^2
- Each flow system has its own specific system equation. If the **flow is laminar**, the **frictional losses** will be proportional to ***Q*** rather than Q^2
- There is also a unique relationship between the actual pump head gained by the fluid and the flow rate, which is governed by the pump design (as indicated by the pump performance curve)
- To select a pump for a particular application, it is necessary to utilize both the ***system curve***, as determined by the system equation, and the pump performance curve
- If both curves are plotted on the same graph, as illustrated in figure, their intersection (**point A**) represents the **operating point for the system**

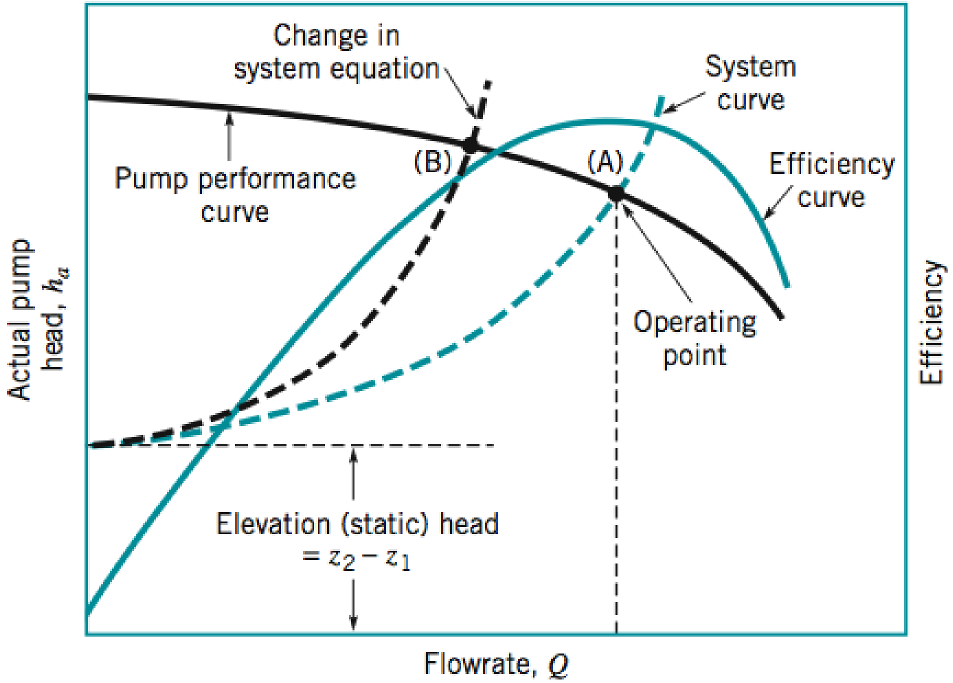


Figure Utilization of the system curve and the pump performance curve to obtain the operating point for the system

System Characteristics and Pump Selection

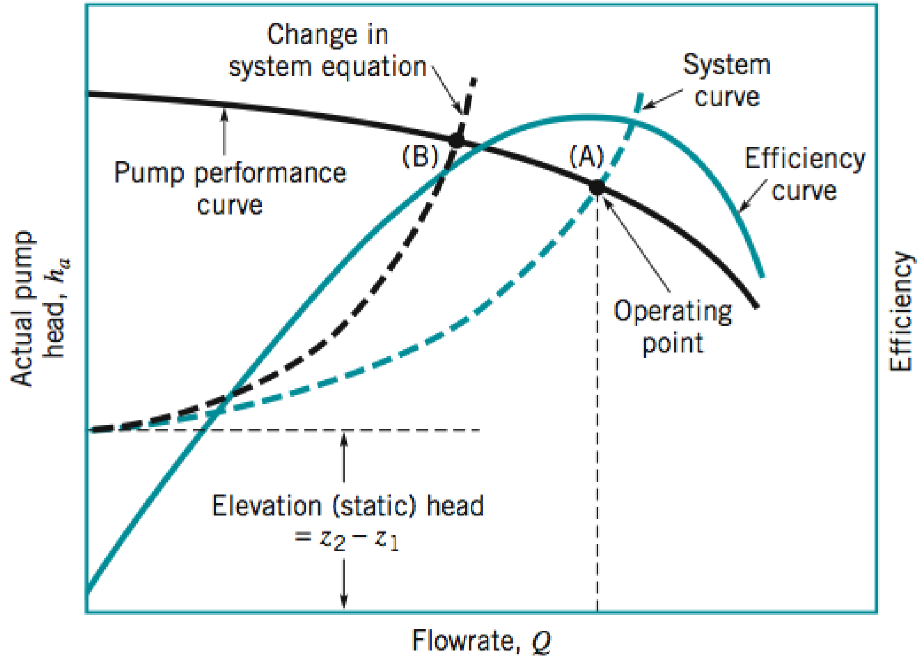
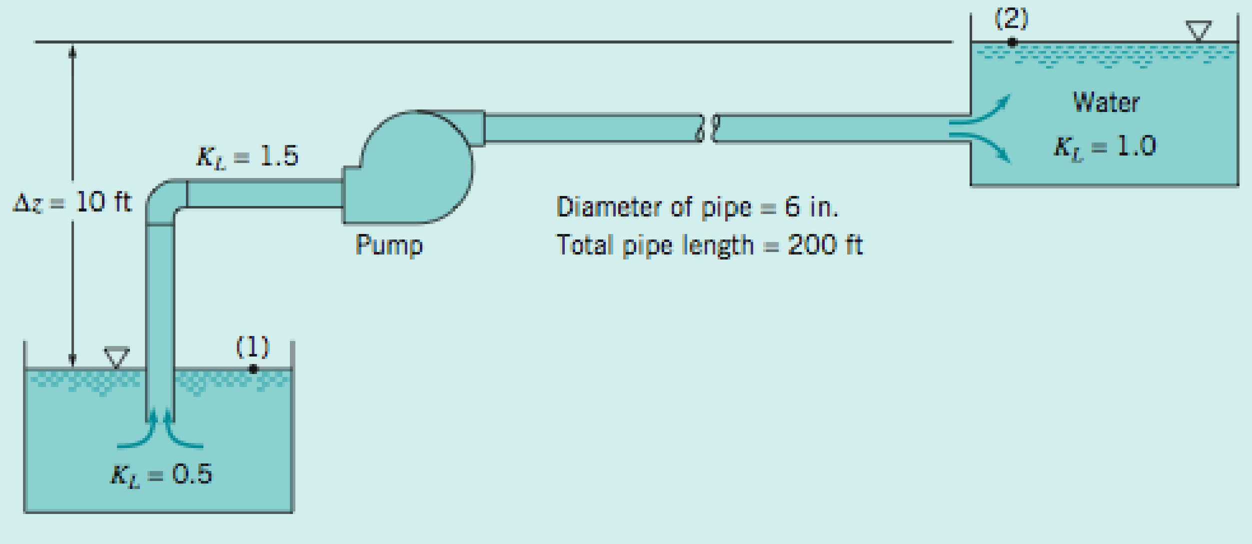


Figure Utilization of the system curve and the pump performance curve to obtain the operating point for the system

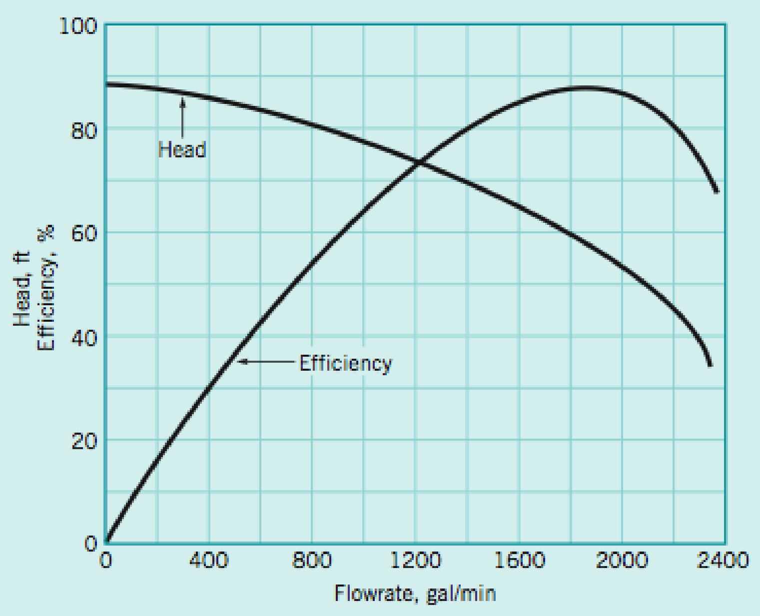
- That is, this point gives the head and flow rate that satisfies both the system equation and the pump equation
- On the same graph the pump efficiency is shown. Ideally, we want the operating point to be near the best efficiency point (BEP) for the pump
- For a given pump, it is clear that as the system equation changes, the operating point will shift. For example, if the pipe friction increases due to pipe wall fouling, the system curve changes, resulting in the operating point A shifting to point B in figure with a reduction in flow rate and efficiency

Example 6.5

Water is to be pumped from one large, open tank to a second large, open tank as shown in Figure (a) below. The pipe diameter throughout is 6 inches and the total length of the pipe between the pipe entrance and exit is 200 ft. Minor loss coefficients for the entrance, exit, and the elbow are shown on the figure, and the friction factor for the pipe can be assumed constant and equal to 0.02. A certain centrifugal pump having the performance characteristics shown in Figure (b) is suggested as a good pump for this flow system. With this pump, what would be the flow rate between the tanks? Do you think this pump would be a good choice?



(a)



(b)

Pump Scaling Law / Affinity Law

When **selecting** a new **pump**, an engineer will **review pump curves** to make sure the pump chosen will run at **the best efficiency point** for the given application. But what if one of the **variables** in the application **needs to change**, and the curve that came perfectly packaged with the pump no longer applies? If no curve exists, should the pump be replaced with another that has the curve you need? Of course not. Using the affinity laws, it is possible to **determine approximate curves** to estimate the pump's new best efficiency point.

Pump Scaling Law / Affinity Law

- To use scaled models in designing turbomachines, geometric similitude is required as well as geometrically similar velocity vector diagram at the entrance to or exist from the impeller

- Two geometrically similar unit having similar velocity vector diagram are **homologous**. They will also have geometrically similar streamlines

Pump Scaling Law / Affinity Law

1. Discharging

$$Q = Av$$

Since area A α impeller diameter D^2.

$$Q \propto D^2$$

Since velocity v α impeller rotation speed ND (N rpm of impeller), then:

$$Q \propto ND$$

$$\therefore \frac{Q}{ND^3} = constant \qquad \cdots (1)$$

Pump Scaling Law / Affinity Law

2. Pump Head

$$Q = Av$$

Since area A α impeller diameter D^2.

$$Q \propto kD^2$$

The discharge Q through homologous unit can be related to head H

$$Q \propto \sqrt{H}$$

$$\therefore \frac{Q}{D^2\sqrt{H}} = constant$$

Eliminating Q based on Equation (1), gives:

$$\frac{H}{N^2D^2} = constant \qquad \cdots (2)$$

Pump Scaling Law / Affinity Law

2. Power of Pump

Power is proportional to γQH. If we use equation (1) and (2) for Q and H, then:

$$\frac{P}{N^3 D^5} = constant$$

Pump Scaling Law / Affinity Law

For Constant Impeller Diameter D

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

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

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

For Constant Impeller Speed N

$$\frac{Q_1}{Q_2} = \left(\frac{D_1}{D_2}\right)^3$$

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^5$$

This equation is called Affinity Law

If the speed of a pump is increased with *10%*

- the volume flow increases with *10%*
- the head increases with *21%*
- the power increases with *33 %*

Specific Speed of Pump

- The specific speed of homologous unit is a constant widely used in selecting type of unit and preliminary design

- Pump specific speed is used to characterize the operation of a pump at its optimum conditions (best efficiency point) and is useful for preliminary pump selection

- It is defined as the speed of a unit of a series of such size that it delivers unit discharge at unit head

Specific Speed of Pump

- Pump specific speed is obtained by eliminating D in below equations and rearranging

$$\frac{Q}{ND^3} = constant \qquad \frac{H}{N^2D^2} = constant$$

$$\therefore \frac{N\sqrt{Q}}{H^{3/4}} = constant$$

- By the definition of specific speed of the pump, the constant Ns, the speed of a unit for Q=1 and H=1, is:

$$N_{sd} = \frac{N(rpm)\sqrt{Q(gal/min)}}{[h_a(ft)]^{3/4}} \quad \text{or} \quad N_s = \frac{N(rpm)\sqrt{Q(liter/hour)}}{g(m/s^2)[h_a(ft)]^{3/4}}$$

Specific Speed of Pump

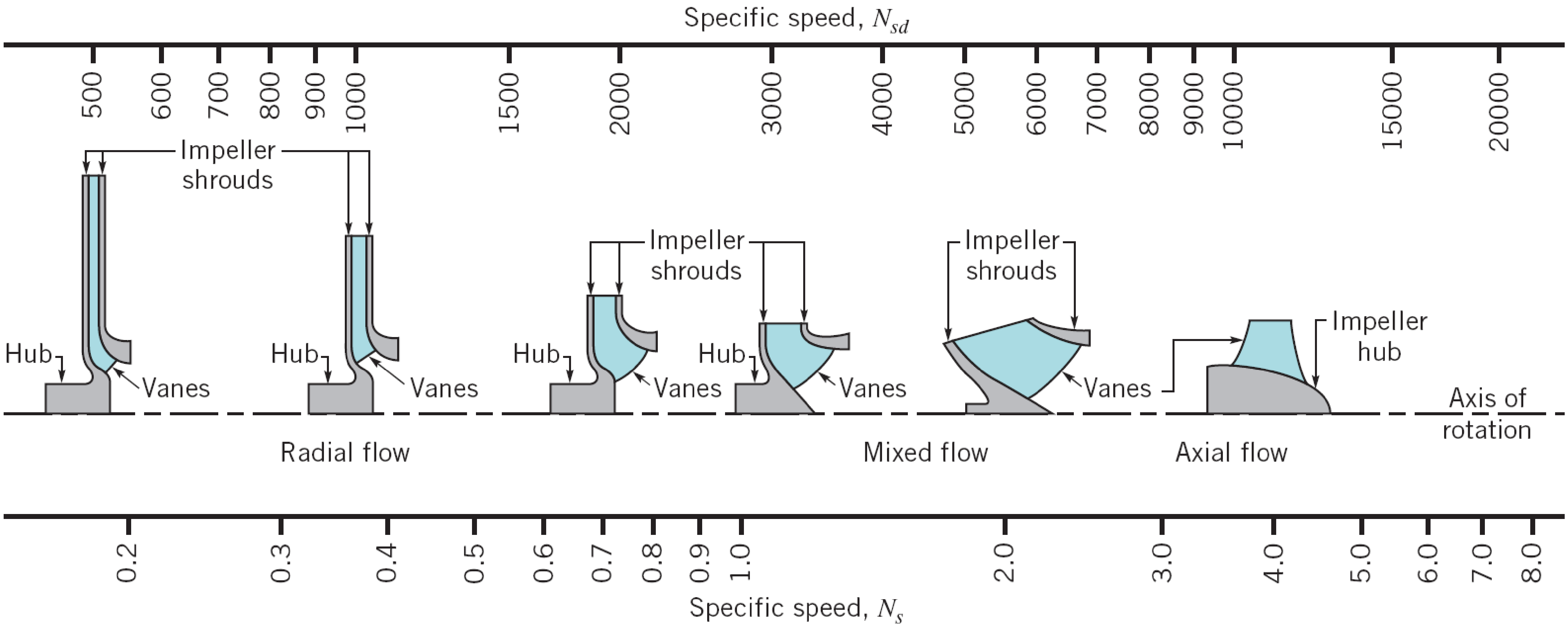
- For any pump it is customary to specify a value of specific speed at the flow coefficient corresponding to peak efficiency only.

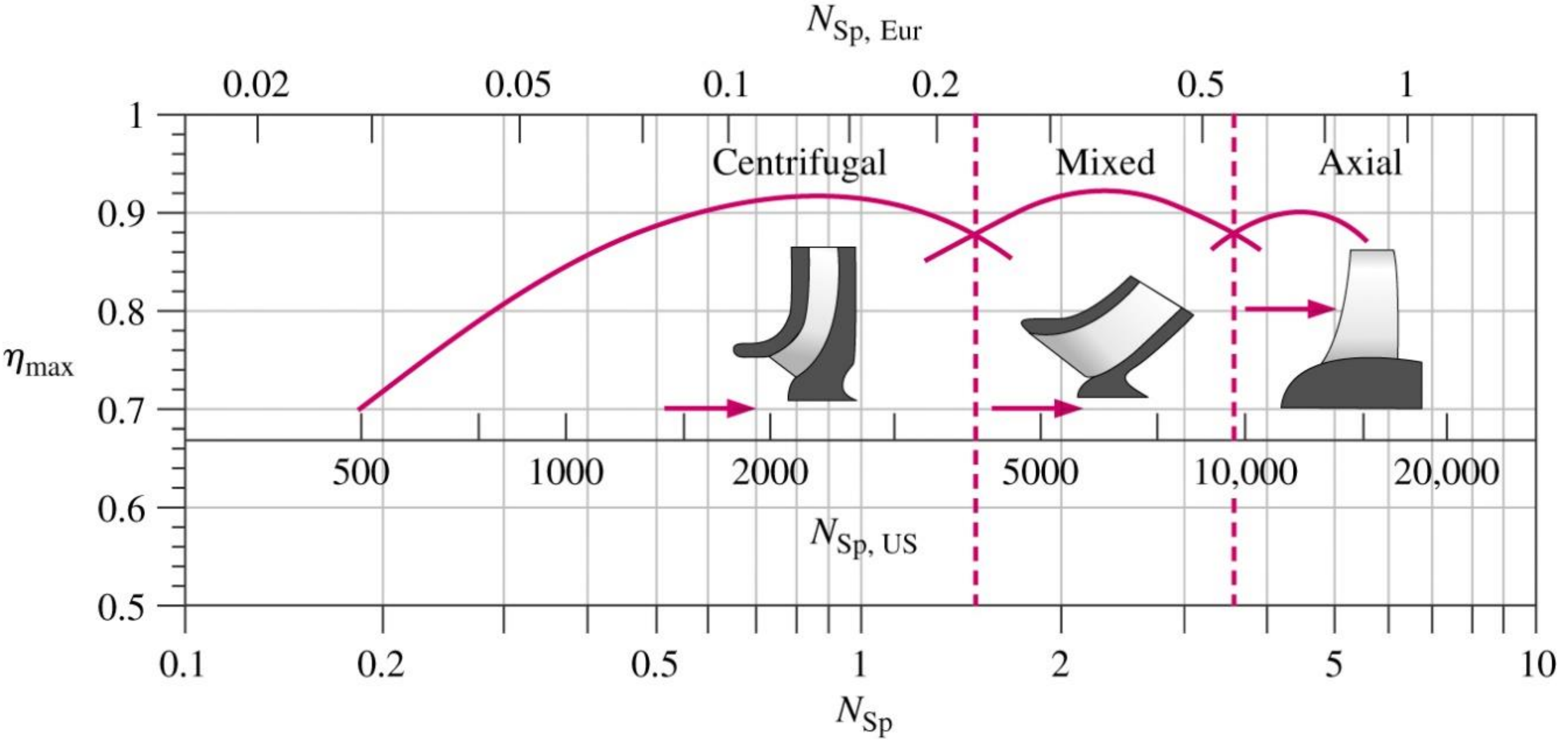
- Each family or class of pumps has a particular range of values of specific speed associated with it.

- Pumps that have low-capacity, high-head characteristics will have specific speed that are smaller than that have high-capacity, low-head characteristics.

- As N_{sd} increases beyond about 2000 the peak efficiency of the purely radial-flow centrifugal pump starts to fall off, and other type of more efficient pump design are preferred.

Specific Speed of Pump




$N_{Sp,\ Eur}$
0.02
0.05
0.1
0.2
0.5
1
Centrifugal
Mixed
Axial
η_{max}
1
0.9
0.8
0.7
0.6
0.5
500
1000
2000
5000
10,000
20,000
$N_{Sp,\ US}$
0.1
0.2
0.5
1
2
5
10
N_{Sp}

THANK YOU

Stay safe!